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handbook of cane sugar engineering

third, completely revised, edition

e. hugot



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HANDBOOK OF CANE SUGAR ENGINEERING

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- Vol. 7. Handbook of Cane Sugar Engineering (Hugot, 3rd edition)

HANDBOOK OF CANE SUGAR ENGINEERING

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THIRD, COMPLETELY REVISED, EDITION



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Preface to the third edition

The technology of sugar manufacture has evolved so much in the course of the last twelve years that the preparation of a third edition has necessitated not only the revision of the major portion of the text but also the addition of numerous, entirely new sections. As important revisions and new sections, there will be found in particular in this new edition, the following:

Powerful modern shredders

Pressure feeders to mills, Australian type

New formulae for maximal speed of mills

A new formula for mill capacity

A more complete table for calculation of power requirements in milling

New systems of electric drive for mills

Mill rollers

The Lotus roller

Calculation of weight of bagasse and weight of juices in the milling tandem

Calculation of material balance in the tandem

A new formula for reduced extraction

A new type of diffuser

Saccharate liming of juice

Rapid clarifiers

Modification of existing clarifiers for rapid operation

Dorrclones

New formulae for heat-transfer coefficient in heaters

Evaporator vessels with lateral or annular downtake

Heat balance for the factory

Falling-film evaporators

Formulae for estimating coefficient k for flow in vessel and pan calandria

Formulae for steam consumption of pans

The method of calculating the material balance for the boiling house

Continuous vacuum pans, including Langreny

Vertical-crystalliser coolers

Continuous centrifugals

Sugar dryers

Standard factory control

Liquid-annulus air pumps

Water requirements of the factory

Forced-draught cooling towers

Rain-type condensers and condenser heaters

Drying of bagasse

Steam balance of the factory for maximum economy

Pelletisation of bagasse.

On the other hand, sections discussing equipment or processes which are outdated or obsolete have been deleted. Readers who are still using such equipment or methods will find them described in the second edition. These include:

Crushers

Reciprocating steam engines

Compound clarification

Circulators in evaporators

Reciprocating air pumps.

For basic measures and quantities, we have adopted the metric system, which is becoming more and more general. Among countries which have adopted it recently, some have chosen the International System (S.I.). Accordingly, the table of equivalent units has been completed by adding these but, on account of the practice followed very generally and which retains the traditional units, such as kilogram force, kg/cm², kcal, these have been utilised for preference. Conversion within the metric system, moreover, is very simple and immediate. To avoid any difficulty for those engineers in English-speaking countries who find themselves more at ease with their British units, and to avoid involving our friends of the American hemisphere in conversions which are sometimes awkward, we have given in the great majority of cases, in formulae and calculations, values in British units alongside those in metric units.

Where given, *tons* and *gallons* in all cases are long tons (2,240 lb. or 1,016.047 kg) and Imperial gallons (4.546 l), respectively.

Tables of figures have in most cases been given in both systems of units.

We hope we have thus given sugar technologists a useful work of reference, particularly useful for calculations and design projects. The introduction of subjects of present-day interest, such as continuous pans and centrifugals, or steam economy to facilitate sale of power to outside authorities, should assist those sugar producers exposed to economic or financial difficulties. We shall be happy if the sugar world is able to benefit by it.

The chapter on information and automation in the sugar factory has been written entirely by George Windal of SNFS, Director of the IRIS, and his colleague J.C. Giorgi, both world authorities in that field. This chapter enhances the value and the interest of the book, and we express our deep gratitude to them.

We thank our friends of SASTA, QSSCT and MSIRI, as well as the directors of engineering firms (FCB, BMA, de Smet and many others), who have replied in clear and friendly fashion to our requests for information; and we express our deep gratitude to our friend George Jenkins who, in his usual friendly and cooperative way, has interrupted a well-deserved retirement to take on the heavy task of translation. To it he has brought all his competence and knowledge and his clear and pleasant style.

E. HUGOT

Translator's foreword

Once again it has been a pleasant task to cooperate with my friend Emile Hugot in translation of a revision of the Handbook. The extent of revision has been rather greater than expected, but this is not surprising in view of Mr. Hugot's thoroughness in treatment and revision of his work. In addition to keeping up with the literature very effectively, he has visited practically all the leading sugar industries of the world in recent years. I have indeed found that the revision includes much material new to me, covering new developments in the industry since I was last in contact with sugar technology some twelve or fourteen years ago.

Thanks again to Mrs. Joan Hodgson, who did the typing for the first edition and has again done an excellent job with the present revision.

G.H. JENKINS

This third edition of the Handbook of Cane Sugar Technology is dedicated to the memory of George Jenkins, a very good friend and a valued colleague, who died suddenly while the volume was in the final stages of production. He was the translator for all three editions of the Handbook, and it is fitting that this edition should be dedicated to him in acknowledgement of his important contribution to the work.

E. HUGOT

List of chapter headings

1. Delivery, unloading and handling of cane.
2. The cane carrier.
3. Cane knives.
4. Tramp-iron separators.
5. Crushers.
6. Shredders.
7. Combinations of cane preparators.
8. Feeding of mills and conveying of bagasse.
9. Roller grooving.
10. Pressures in milling.
11. Mill speeds.
12. Mill capacity.
13. Mill settings.
14. Power requirements of mills.
15. Electric drive of mills.
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18. Construction of mills.
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34. Crystallisation.
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37. Sugar.
38. Molasses.
39. Boiling-house control.
40. Vacuum equipment.
41. Steam production and usage.
42. Steam turbines.
43. Electricity.
44. Pumps.
45. Piping and fluid flow.
46. Automation and data processing (by G. Windal and J.C. Giorgi).

References

References are listed at the end of each chapter, following the usual method in English-language publications. The abbreviations used are generally self-evident, with the possible exception of the following:

- ISSCT - International Society of Sugar Cane Technologists
- QSSCT - Queensland Society of Sugar Cane Technologists
- SASFPI - South African Sugar Factory Plant Installations
- SASTA - South African Sugar Technologists' Association
- SMRI - Sugar Milling Research Institute
- TSJ - The Sugar Journal.

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1. Delivery, Unloading and Handling of Cane

The factory takes delivery of the cane, either directly at the factory weighbridge, or at auxiliary weighbridges serving certain important or remote points in the area from which the mill draws its supplies.

Transport is arranged by the factory, either by railway, or more often by lorries, or by tractors and trailers.

The bulk density, or weight per unit volume, of cane in the truck depends on the manner in which it is loaded. In Hawaii, for instance, it may be picked up in bulk in the field by a mobile crane mounted on caterpillar tracks and fitted with a grab. This simply drops the cane in a tangled mass into the trailer; in this case the bulk density may be taken as about 200 kg/m^3 (12 lb./cu. ft.). If the loading is done rather more carefully: 300 kg/m^3 (18 lb./cu. ft.). In Hawaii¹, a figure of $225 - 256 \text{ kg/m}^3$ (14- 16 lb./cu. ft.) is used. If the cane is loaded by hand, the stalks being placed parallel to each other in bundles or packages, as is often done with small suppliers, the density may be taken as about 350 kg/m^3 (22 lb./cu. ft.) and may reach 400 kg/m^3 (26 lb./cu. ft.). If the cane is cut by a chopper harvester and loaded in bulk, the density is approximately $300\text{-}380 \text{ kg/m}^3$ (18-24 lb./cu. ft.), averaging 340 kg/m^3 for pieces 28 cm (11 in.) in length. In Hawaii, figures of $321 - 335 \text{ kg/m}^3$ (20 - 21 lb./cu. ft.) are given; in Jamaica², $300\text{-}330 \text{ kg/m}^3$ (19-21 lb./cu. ft.).

This bulk density depends on the stand of the cane. Straight and erect stalks will give a more compact loading than curved or lodged canes.

ORGANISATION OF CANE SUPPLY THROUGHOUT THE DAY

A cane sugar factory generally operates continuously from Monday morning till Saturday evening. It shuts down for about 36 hours, including Sunday, for cleaning the multiple effects and for minor repairs. It operates then for about 132 hours per week.

During the day, cane transport is generally carried on for only 12 hours, from 6 a.m. to 6 p.m. So that the mill will not run out of cane, it is necessary that the factory should receive in 12 hours, during the day, the tonnage which it crushes in 24 hours.

Towards 6 p.m. then there will have accumulated a stock at least equal to half the daily tonnage:

$$\text{Overnight provision} = 12 A + a \quad (1.1)$$

A = quantity of cane crushed by the factory per hour

a = margin of safety, which should always be maintained to avoid being obliged to stop the mill for lack of cane.

This margin a is necessary to make provision for accidental fluctuations in supply: rain retarding the cutting and restricting transport, irregularities in supply by the farmers, etc. It should not be too small, nor too large. In the latter case, the cane yard will be uselessly encumbered, and the delay occurring between the cutting of the cane and its passage through the mills will be unnecessarily increased. A suitable quantity is that corresponding to three hours' crushing:

$$a = 3 A \quad (1.2)$$

If this value be adopted, the yard should be able to accommodate an overnight stock of:

$$P = 12 A + 3 A = 15 A \quad (1.3)$$



Fig. 1.1. Sling with automatic hook.



Fig. 1.2. Lifting a bundle of cane

UNLOADING OF CANE

As drays have practically disappeared from the scene, we discuss only cane transported by mechanical means: trailers, lorries and railway wagons.

Lorries and other vehicles are either unloaded onto a pile by means of a crane or are self-discharging.

A. Unloading by crane. This cane is often loaded in "packets" or bundles, bound by three chains or "slings" (Fig. 1.1). At one end these slings slide in a special hook fitted with a pawl, while a ring is attached to the other end.

The unloading device at the factory, generally a crane, lifts the bundle of cane with the aid of a "swingle-bar" or steel bar fitted with three hooks. The crane lifts the bundle and deposits it on the heap of cane which is called a "stock-pile". The catches of the sling-hooks are then released, and the crane lifts the swingle-bar with the slings hanging (Fig. 1.2). Release of the catch may be controlled electrically.

For a 3-tonne crane, each bundle is of the order of 1,500 – 3,000 kg (3,000 – 6,000 lb.) of cane, generally about 2,500 kg (5,000 lb.). A 4- or 5-tonne truck carries two such bundles.

When the swingle bar is replaced by a grab (Fig. 1.7), trucks carrying cane in bulk may also be unloaded. The grab operates mainly at night to transfer to the lateral tables the cane stacked during the day.

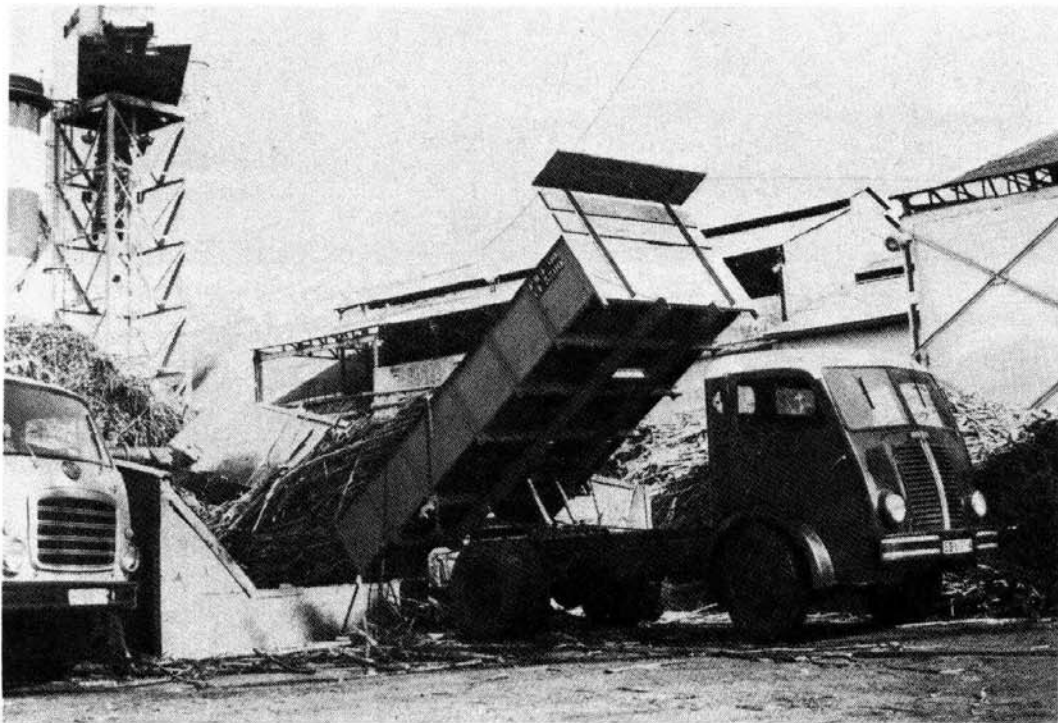


Fig 1.3. Unloading by tipping truck.

B. Self-discharging vehicles. Several types of these are in use, the main ones being:

(a) *Tipping trucks.* These are trucks fitted with a body which is arranged to pivot about its lower rear edge, and actuated by a hydraulic device which tilts the truck body towards the back till its load of cane slides into the conveyor arranged to receive it (Fig. 1.3). This conveyor is generally an inclined lateral table, rising towards the front so that it discharges the cane into the carrier from a height of about 1.5 m (5 ft.) above it; its rear end is located in a pit of approximately 2 m (6 ft.) in depth. The cane falls into this pit, which is defined at the back by a plate rising about 0.6 m (2 ft.) above ground level, low enough not to interfere with the tipping of the truck.

The slope of this lateral table is approximately 15° for the slat conveyor type, and 8° for the type with fixed plates and moving brackets. Its length accordingly depends on the level of the carrier relative to the ground, and is generally of the order 6 – 10 m (20 – 30 ft.). The cane slides from the truck body when it reaches an inclination of about $40\text{--}45^\circ$.

(b) *Trucks with detachable bodies.* This is a similar system, except that the truck body, called a "bin" or "basket", is separate from the chassis on which it rests, and may be placed on the ground or lifted onto the chassis by means of either mobile crane jibs carried on the chassis or a winch placed behind the cabin. In the former case, it can be tipped into the carrier in the same way as a tipping truck. If several bins are supplied for each truck, these can be filled in the field while the truck is returning with an empty bin; as soon as the empty bin is unloaded at a convenient point for the cane cutters, a full bin can be loaded on the truck immediately without loss of time. A similar delivery and pick-up can be effected at the fac-



Fig. 1.4. Cane transport by detachable container (loading in field)

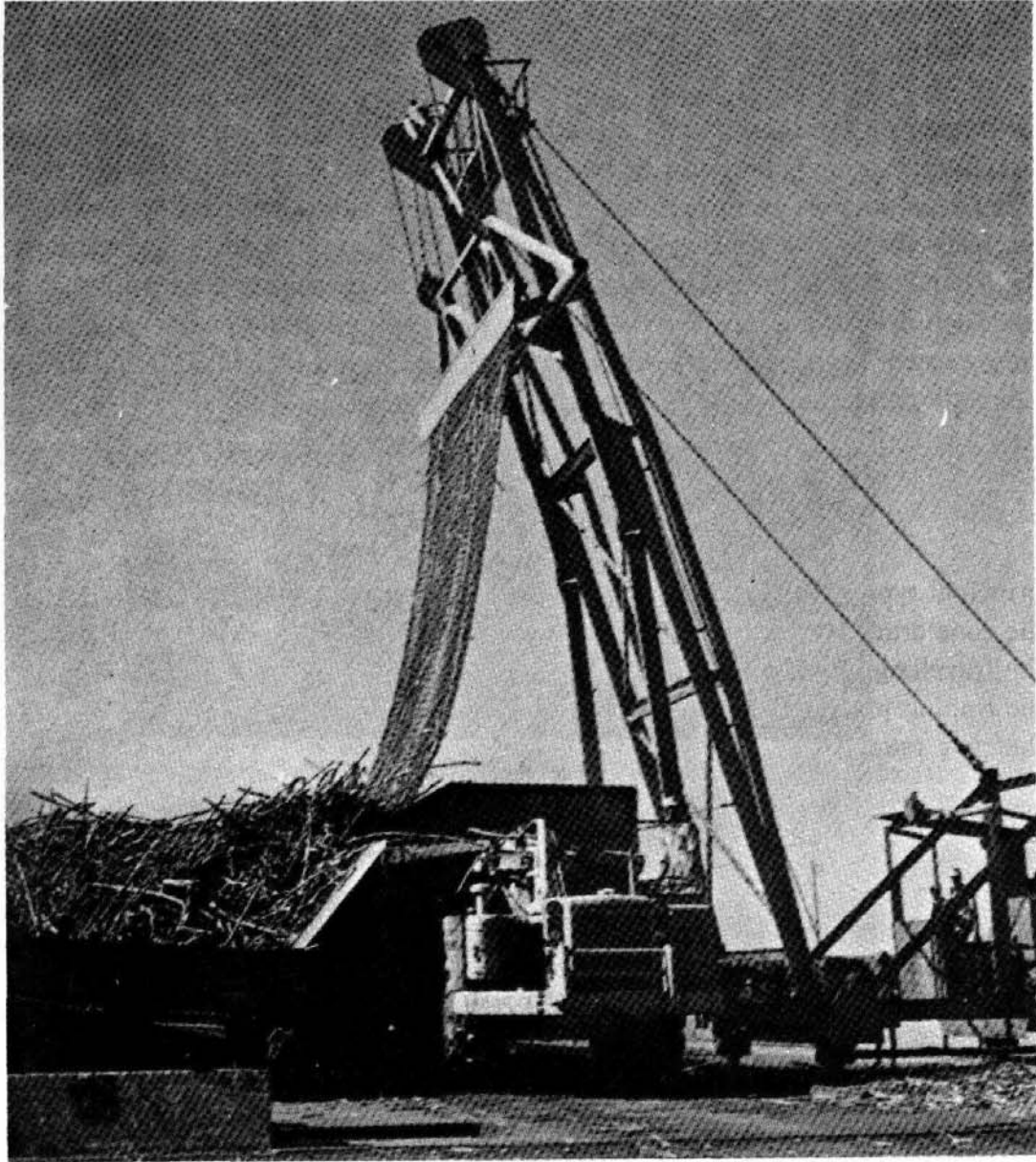


Fig. 1.5. Unloading by net.

tory; thus a rapid turnaround of the truck is achieved, and a good capacity in tonnes of cane transported per day (Fig. 1.4).

Numerous modifications of this system are in use.

(c) *Tractors and semi-trailers with nets.* Semi-trailers generally of large tonnage (25 - 40 tonnes) are fitted with a body of steel mesh (Fig. 1.5). On one of the sides of the body is fixed a steel net which drops down that side, passes across the bottom, rises on the opposite side, to which the end of the net is attached. In the field or at the cane loading station, the cane is loaded in a pile on this net. On arrival at the factory, the tractor stops alongside a deep, large-capacity auxiliary carrier, the side of the trailer body carrying the fixed end of the net being lined up alongside the carrier and directly above the side of it. A crane then

hooks onto the opposite end of the net and lifts it; the contents are thus tipped into the auxiliary carrier (Fig. 1.5).

(d) *Flat-topped bulk trailers.* Small trailers that can be attached behind light field tractors are often used, especially in South Africa. The cane is placed in piles on the ground with the stalks as far as possible lying parallel to each other. A cable is placed under the pile, and brought back above it; a winch placed on the tractor then hauls in the cable, thus hauling the pile of cane up on to the trailer. Many versions of this device are used, with rear or lateral loading. Such devices are inexpensive but, the canes being loosely packed, the width of the load may exceed the maximum permitted on the roads, and their use should be limited to fields directly connected to the factory by private roads (Fig. 1.6).

CANE-HANDLING EQUIPMENT

The principal types of equipment used at the factory cane yard are:

- (1) The cane crane, or "derrick"
- (2) The travelling crane
- (3) The tipping platform or truck tip
- (4) The cane rake.



Fig. 1.6. Loading cane in bulk by means of cable (Bell system).

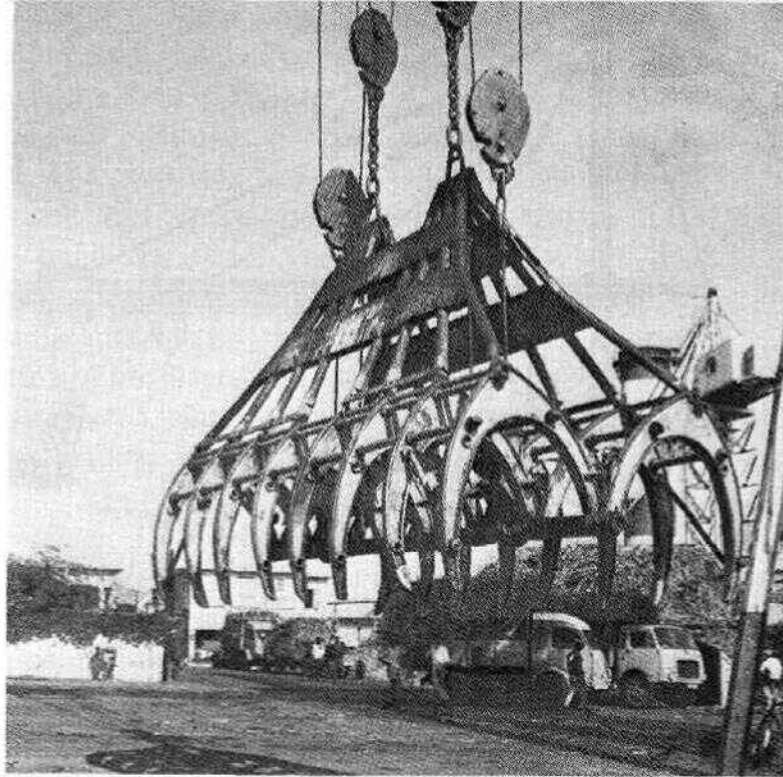


Fig. 1.7. Grab loader.

1. Cane crane

This is by far the most-used device. It is often called by its American name: "derrick". It consists (Fig. 1.8) of a mast of fabricated steel, mounted on a pivot, so that it can rotate through a full circle. At a suitable height this vertical mast carries a horizontal arm forming a track on which a trolley, carrying two pulleys, can run to and fro. A cable passes over these two pulleys and hangs between them, thus forming a loop which carries a snatch block, on which may be hung either the swingle-bar or the grab.

The driver works in a cabin, which is located at the base of the mast or below the arm and houses the motors for the various movements: rotation, traversing, raising or lowering of the hook. It also generally houses the winding drums.

These cranes are driven generally by electric motor. They are classified according to their mode of support, into:

- (a) Gayed cranes (Fig. 1.8), now used less and less
- (b) Self-supporting cranes (Fig. 1.9).

(a) Gayed cranes. This is the lighter form, the stability of the crane being ensured by retaining cables or guy ropes attached to a ring at the top of the mast (Fig. 1.8).

These guys must permit of rotation of the horizontal arm, and so must be fixed in the ground at a considerable distance from the axis of the crane.

Number of guys. Strictly speaking, 3 guy ropes, arranged at an angle of 120° to each other, would be sufficient. However, the number of guys is generally chosen so that breakage of

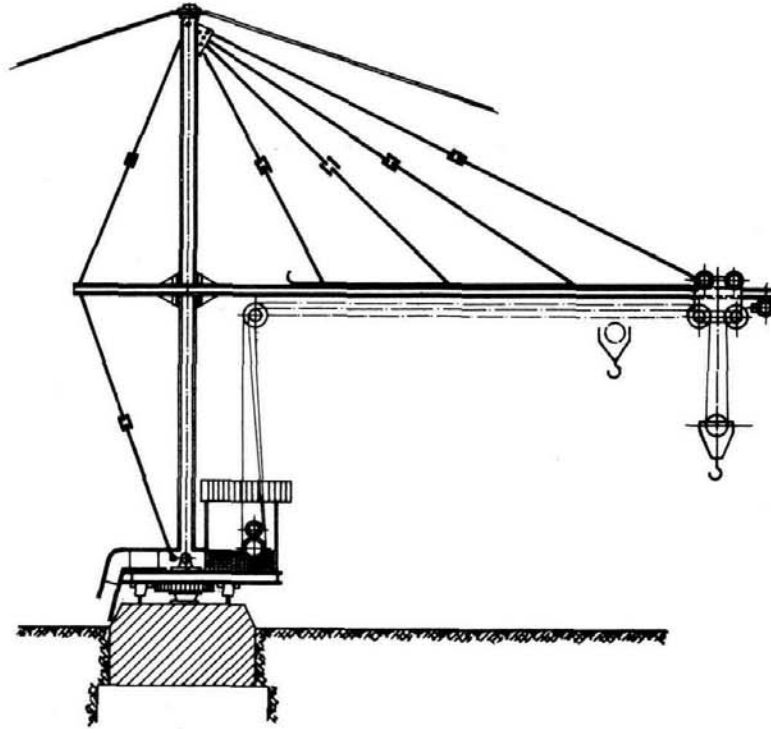


Fig. 1.8. Cane derrick with guys.

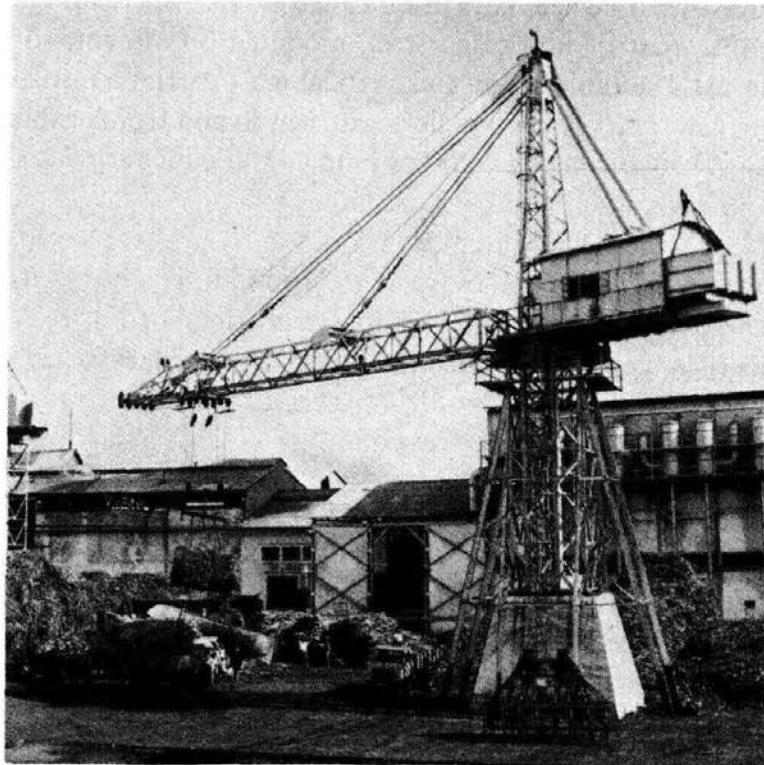


Fig. 1.9. Ten-tonne self-supporting derrick.

one of them does not cause the derrick to fall. This condition requires a minimum of 5, at 72°. For preference 6 or 8 should be used, if possible.

It would be necessary then to fix 5, 6, 7, or 8 anchor blocks, placed symmetrically on a circle of 55-75 m (180-240 ft.) radius with the axis of the crane as centre, to which the guys are attached. The presence of factory and office buildings makes the solution of this problem very difficult. If it is necessary to modify the spacing of two cables to avoid in particular the factory buildings, it is necessary to ensure that a breakage of one of these widely spaced cables will not leave a sector of more than 150°, which is the limit permissible for stability of two adjacent cables.

Obviously, in case of breakage, it will be necessary to stop the crane immediately until the damaged cable is replaced, since the remaining guys would not be able to ensure stability under the dynamic loading of the crane in operation.

Tension of the cables. To obtain maximum rigidity, the cables must be tightened so that the mast describes only a very small cone as the derrick is rotated, but without increasing unduly the tension of the cables themselves. This tension should be checked frequently. The cables should be painted or tarred every two or three years (to prevent corrosion).

Thickness of the cables. For a crane of 3 tonnes capacity, 5–8 cables of 25 mm (1 in.) diameter are provided.

(b) Self-supporting cranes. These are obviously heavier and more massive (Figs. 1.2 and 1.9), but avoid the nuisance and danger of guys, which restrict movement at the cane yard and which are sometimes very difficult to locate conveniently.

In countries subject to cyclones, these cranes should be designed for a wind pressure of at least 350 or preferably 400 kg/m² (80 lb./sq.ft.) (live load).

Capacity of cranes. The important characteristics of an unloading crane are:

- (1) The lifting force
- (2) The radius of operation
- (3) The maximum height of lift of the grab.

(1) *Lifting force.* This is the maximum weight which the crane can lift, at the end of the arm. It is the most important factor, which serves to specify its capacity. For instance, one speaks of a 3-tonne or 5-tonne crane. The usual values are 3, 5 and 10 tonnes.

(2) *Radius of operation.* This is the horizontal distance between the pivot of the crane and a vertical line through the pulley of the snatch block (corresponding to the centre of the grab) when the trolley is at its extreme position at the end of the crane arm. Typical values are: 18,25 and 30 m (60, 80, 100 ft.).

(3) *Maximum height of lift.* This is the height from the end of the tines of the grab above the platform level, when the grab is raised to its maximum height.

To increase this height, and at the same time to enable the driver to oversee the operations

and traffic of the platform, the base of a small (3 t.) derrick is generally raised by placing it on a truncated cone of masonry about 2 m (6 ft.) in height.

The height of lift is generally of the order of 6–8 m (20–25 ft.).

As a general rule, for a factory having only one derrick at its unloading station, the characteristics adopted are approximately as follows:

(1) Lifting force:

$$F = 0.1 A \quad (1.4)$$

F = lifting force, in tonnes

A = crushing rate of the factory, in t.c.h.

(2) Radius of operation:

$$R = 3 \sqrt{A} \quad (R = 10 \sqrt{A}) \quad (1.5)$$

R = radius of operation, in m (ft.).

(3) Maximum height of lift:

$$H = 8 \text{ m (25 ft.)} \quad (1.6)$$

When two derricks (or two unloading devices) are available, the total tonnage A is divided between them by allotting the tonnages A_1 and A_2 respectively, such that

$$A_1 + A_2 = A$$

Speeds and power requirements. The speeds adopted for the various movements of the crane, and the power required for the corresponding motors, are generally of the following order:

(1) *Lifting:*

Lifting speed 20–50 m/min (80–150 ft./min)

Power required 7.5 kW (10 h.p.) per tonne nominal capacity

(2) *Rotation:*

Rotational speed 1.5—2.5 r.p.m.

Power required 1.0 kW (1.5 h.p.) per tonne nominal capacity

(3) *Traversing:*

Speed of the trolley 30–60 m/min (100–200 ft./min)

Power required 0.75 kW (1 h.p.) per tonne nominal capacity

When the crane is driven by compressed air, the single motor requires approximately 7.5 kW (10 h.p.) per tonne nominal capacity.

Lateral tables. In large or medium factories, it is necessary to provide one or more lateral tables, on which the crane or cranes deposit the cane. In this case, the carrier does not receive the cane directly, but is fed by these lateral tables, controlled by an operator who regulates their speed as required. The feeding of a carrier supplied in this way is much more regular than in the case of direct feed.

Bulk density of cane in the pile. This density is similar to that of cane loaded mechanically, or about 200 kg/m^3 (12 lb./cu.ft.), if it refers to cane dumped at random. It increases to 300 kg/m^3 (18 lb./cu.ft.) if the pile is made up of bundles of parallel stalks, deposited by the swingle-bar. Tromp³ estimates 400 kg/m^3 (25 lb./cu.ft.).

For cane cut in small pieces 0.3-0.6 m (1-2 ft.) in length, as produced by certain mechanical harvesters, a figure of 350 kg/m^3 (22 lb./cu.ft.) may be taken⁴.

2. Transporter crane

Factories handling more than 150 or 200 t.c.h. may find the derrick barely adequate for storage and handling of the cane. They then resort to use of the transporter crane, which has greater capacity and is more easily accommodated.

The system is closely analogous to that of the derrick. Instead of a circle, it serves a rectangle, the length and breadth of which may be fixed at will. For this reason, it will be indicated for certain long and narrow cane yards which have not sufficient space for a crane to swing. The transporter crane must obviously span the cane carrier which it serves (Fig. 1.10).

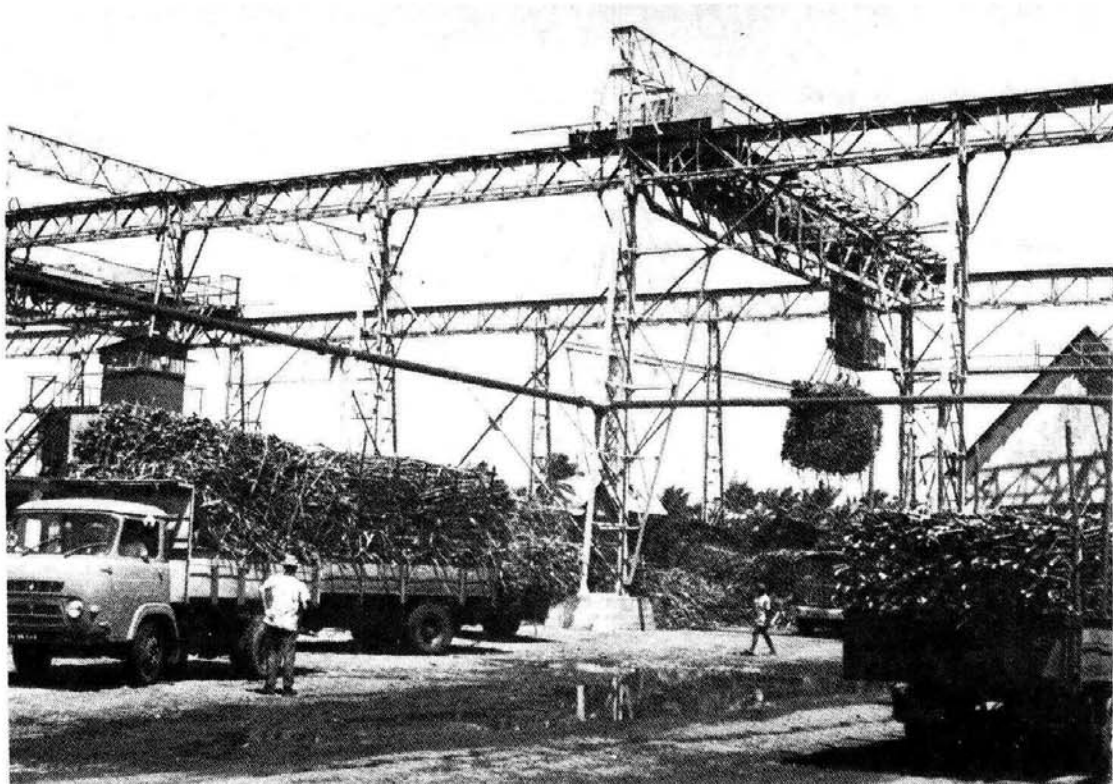


Fig. 1.10 Travelling crane (Fives Cai - Babcock)

Normal speeds are approximately:

Raising the hook or grab	15- 30 m/min (50 – 100 ft./min)
Longitudinal movement of the crane	50 – 100 m/min (150 – 300 ft./min)
Traversing movement of the trolley	30- 60 m/min (100-200 ft./min)

In addition, two minutes must be allowed for each movement for detaching the chains and returning for the next load⁵.

The corresponding power requirements of the motors are approximately:

Lifting:	4.5 kW (6 h.p.) per tonne nominal capacity
Longitudinal movement:	1.0 kW (1.5 h.p.) per tonne nominal capacity
Traversing movement:	0.25 kW (0.3 h.p.) per tonne nominal capacity

Often both the derrick and transporter crane are used, and this combination is readily adaptable to all types of unloading area. One advantage of the transporter crane is that the storage capacity can be readily increased, by increasing its length of travel. When it becomes very long, it may be of advantage to provide the length of track by means of two transporter cranes, which can then work simultaneously, each one serving one end of the carrier. The work is further facilitated if each one is provided with its own lateral table.

The width of transporter cranes is of the order of 20 – 30 m (60 – 100 ft.). Their cost increases much more rapidly with increasing width than with increasing length. The length may have almost any value up to 100 m (300 ft.) for a double transporter crane. The height of lift is 8- 13 m (25 – 40 ft.) and the capacity generally 6 – 10 tonnes.

3. Tip for trucks or trailers

When the cane arrives by railway trucks, it is more practical to empty the truck in one action,

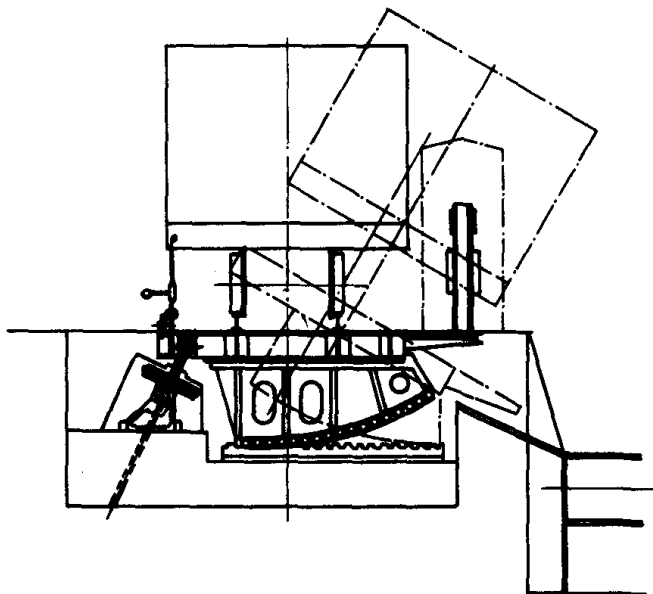


Fig. 1.11. Lateral tip for trucks.

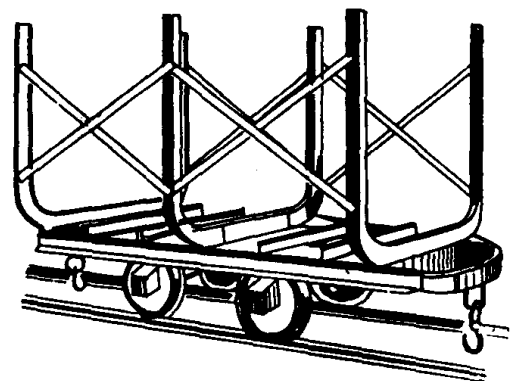


Fig. 1.12. Small truck for end-tipping.

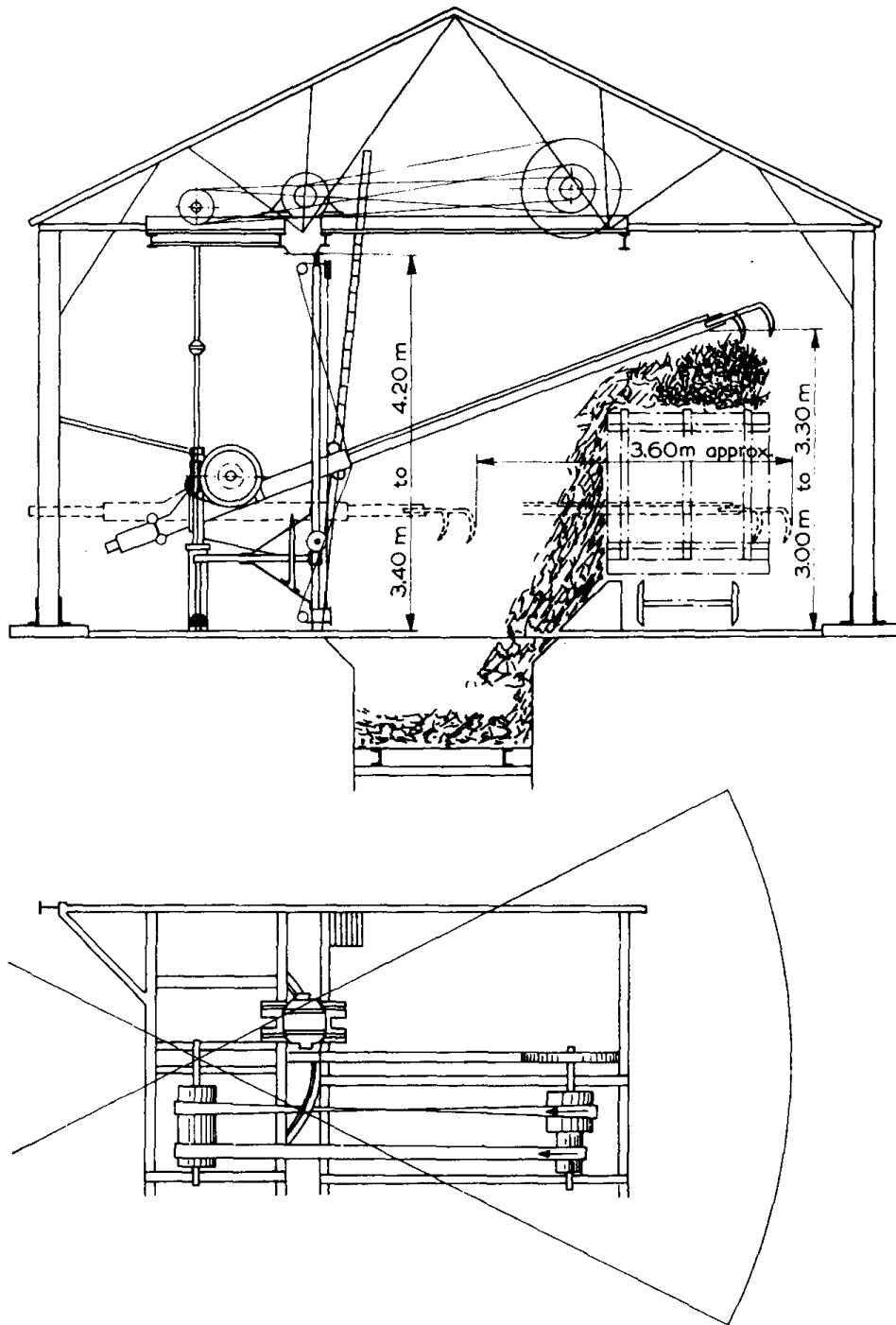


Fig. 1.13. "Wicks" rake (Mirrlees - Watson).

rather than to lift separately, with the crane, the bundles of cane piled in the truck.

This presents several advantages:

- (1) Saving in time
- (2) Saving in slings (this is by no means negligible)
- (3) Less handling, because the cane is emptied directly into the carrier, as with drays
- (4) Thanks to the absence of slings, the risk of passing pieces of chain or ratchet hooks

through the mill is avoided. With a crane and slings, it is very rarely that a season is completed, in the absence of a magnetic separator, without removing two or three such pieces from the mills; the roller grooving suffers severely as a result.

There are two types of tip:

- (1) The lateral tip, for large waggons (Fig. 1.11)
- (2) The endwise tip, for small trucks.

The trucks (Fig. 1.12) are provided with a U-shaped iron frame forming a cradle, in which the cane is placed lengthwise parallel to the track. The ends are free, and the cane, well compacted during the journey from the fields to the factory, falls out when the tip reaches the desired inclination (40-45°).

Tips are sometimes installed for lorries also. The lorry is generally tipped endwise, toward the rear.

The angle of discharge, for whole or chopped cane loaded in bulk, is approximately 42°. Tips are designed for a maximum inclination of approximately 48°.

4. Rakes

These form an auxiliary unloading device, which is used mostly in Java and in some British countries. Figure 1.13 gives a view which will make description superfluous.

It will be noted that the rake has not only a "picking" movement and a to-and-fro movement in a mean plane, but that it can also make the same movement in a great number of planes, forming a dihedral angle of some 30 – 40° to one side or the other of this mean plane.



Fig. 1.14. Loading canes in billets Legras trailer.

Chopper-harvested cane

Mechanical harvesting of cane is generally carried out by chopper harvesters such as the Massey-Ferguson, Toft, Claas, Cameco, Thomson, J. & L. (Fig. 1.14). These are machines which cut the cane, chopping it into pieces 25 - 30 cm (10 - 12 in.) in length, and discharging it into a truck or trailer moving alongside the harvester. When the vehicle is filled it is removed and replaced by another. These trucks or trailers circulate in the field, holding 5 - 10 tonnes of cane, and then transfer their load to a road (or rail) vehicle which runs the length of the field, holding 25 - 30 tonnes of cane and transporting its load to the factory (Fig. 1.15). One such

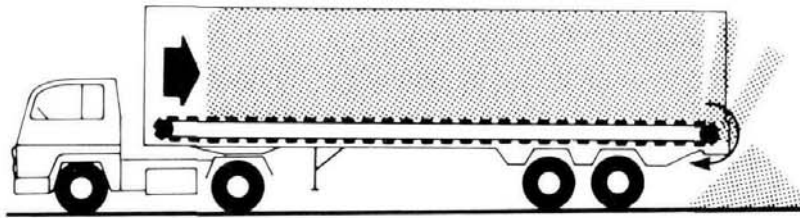


Fig. 1.15. Trailer with moving bottom (Legras).



Fig. 1.16. Transferring cane in the field (Legras).

transport widely used in French countries, and elsewhere, is the Legras trailer with conveyor bottom. This trailer has a slat conveyor which forms the floor, and a door at the back. On arrival at the factory, it stops above a pit, the door is opened and the conveyor set in motion; the chopped cane falls onto a table at the bottom of the pit (Fig. 1.16).

Storage in containers

Among newly established sugar enterprises, some produce themselves the greater part of the cane which they crush. These have adopted the following system: the cut cane is stacked in containers of large volume, for example, 12 m (40 ft.) in length, 2.50 m (8 ft.) in width, 2.50 m (8 ft.) in height and of 75 m³ (255 cu.ft.) in volume, holding about 22 – 25 tonnes of cane. These containers are transported from the field to the factory on flat-top trucks and deposited in rows on the unloading area at the factory. They are unloaded in the order in which they arrive, onto the lateral tables, either by tips or by discharging nets.

Pusher - stacker transfer

Other establishments, equally new, are content with discharging the cane on a reserved section of the cane yard which is concreted and on which one or two specialised machines, equipped with a pusher blade or large scoop, push the cane into a regular heap 2 or 3 m (6 or 9 ft.) in height. These same machines later bring the cane back to feed the cane carrier tables. Generally, there are two such machines, one receiving the cane and stacking it, the other bringing it back and feeding the conveyor. This arrangement permits the formation of two stacks, on opposite sides of the yard; thus it is possible to finish one stack before commencing on the other and so to avoid leaving one portion of the canes longer in the yard than the other. Delivery of the cane may be carried out around the clock or over 12 hours per day. In the latter case, the area of the unloading yard should consequently be calculated as a function of the height of the stack, arranging the space necessary for manoeuvring the two mobile machines. These are generally provided with four-wheel steering, to give a very small turning circle.

LATERAL FEEDING TABLES AND CARRIERS

When the number of trucks is insufficient (which may occur at certain fixed times of day, or accidentally at any moment), the crane makes up the cane supply to the carrier with the aid of the grab.

However, variations occur in the quantity of cane so deposited per unit length of the carrier; there are gaps which cannot be filled in time. On the other hand, a bundle of cane from a lorry, deposited almost intact on the carrier, can cause an overload at the knives, in the case of electric drive, opening the circuit breaker.

When such dense bundles of parallel canes occur, it is necessary to slow down the carrier and practically stop it, otherwise there is a risk of choking the knives. Even with close attention, the attempt is not always successful.

These difficulties, due to the direct feeding of the carrier, have led to the adoption of auxiliary carriers. Many of these disadvantages are indeed overcome by providing a second or

auxiliary carrier at right angles to the main carrier. Such a carrier may take the form of:

- (1) A wide and relatively short platform or "lateral table"
- (2) A carrier identical with the main carrier, but at right-angles to it. This is often called an "auxiliary carrier".

1. Lateral table

This is in effect a very wide and short carrier, driven by an independent motor. In plan it is rectangular or approximately square in shape. The upper run of the lateral table should end at approximately 2 m (6 ft.) above that of the main carrier and its discharge end almost vertically above the side plate of the carrier.

The crane deposits the cane on this table, and keeps up the supply to it as it discharges. An attendant, conveniently located to watch the whole of the cane yard, and handy to the speed control of the motor which drives the lateral table, starts it up whenever he sees that the main carrier would otherwise be insufficiently loaded. The cane falls from the lateral table into the carrier, and one advantage of this system is that the cane falls in a more or less tangled condition, greatly facilitating the work of the cane knives. As soon as the desired quantity of cane has fallen onto the carrier, the lateral table is slowed down or stopped. The movement of the latter is thus very irregular, interrupted and jerky, and it is advantageous to drive it by means of a very flexible variable-speed device, such as the French "Elcotron" or the English "Heenan and Froude" drive. These provide a drive without solid contact: a steel flywheel coupled to the motor transmits the drive, by induced Foucault currents, to a sleeve surrounding it and connected to the driven shaft.

The lateral table is particularly useful in countries where payment for cane is made according to sugar content or recoverable sugar. The cane to be tested from a particular supplier may then be accumulated on a special lateral table, without interfering with the feeding of the mills. When the table is filled, its load of cane is fed to the mills without interrupting

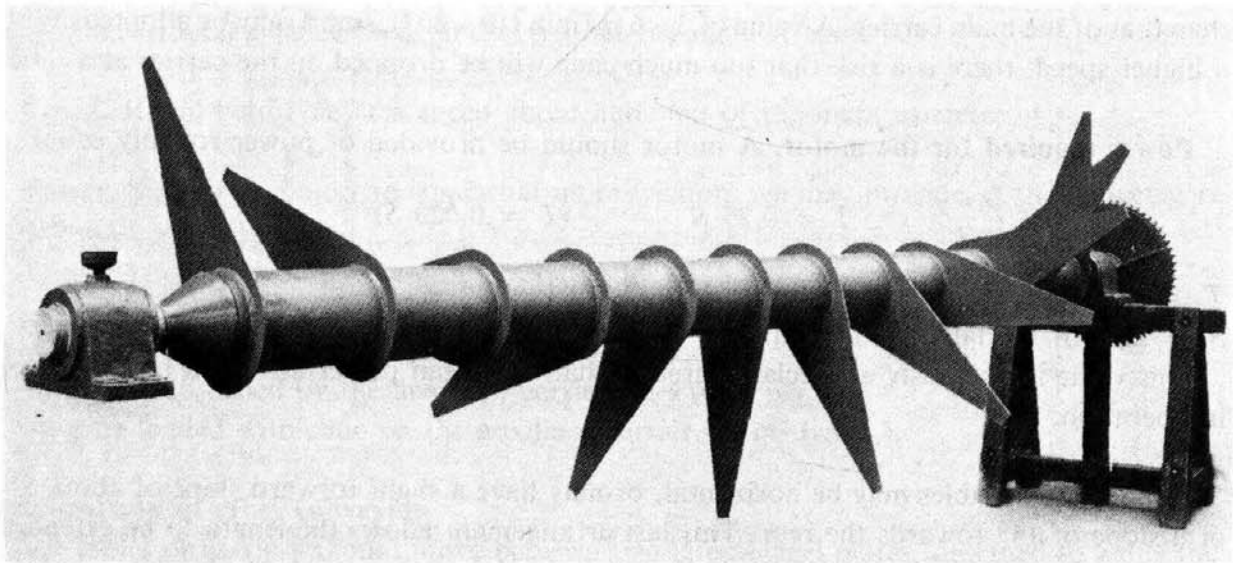


Fig. 1.17. Cane tumbler (Fives Cail - Babcock).

the movement of the main carrier; the beginning and end of the batch to be analysed are simply marked with lime, in such a way that the juice sample boy can see clearly the beginning and end of the batch.

Tumbler. The lateral table is much more effective if fitted with a "tumbler". This is a horizontal shaft (Fig. 1.17), placed above the axis of the front drum of the table, and which rotates slowly in the reverse direction. It is provided with arms arranged in a helix along its length, which ensure that the cane falls into the carrier in small lots, avoiding a heavy fall of large masses which would be liable to provoke chokes at the knives.

High-speed tumblers may also be used, running at 80 – 100 r.p.m. and furnished with small knives at the end of the arms; these give a more regular feed and cut some of the cane stalks, thus assisting the work of the knives.

Area of the lateral table. The dimensions of the table vary considerably according to individual cases. A good dimension, designing for ample capacity, is:

$$S = 2 A/3 \quad (S = 6 A) \quad (1.7)$$

S = area of the lateral table, in m² (sq. ft.)

A = crushing rate of the factory, in t.c.h.

For example, a table of 5 x 7 m (16 x 20 ft.) for 50 t.c.h.

For crushing rates greater than 70 t.c.h., it is of advantage to provide two lateral tables rather than one large one. The dimensions of these will be arranged to follow the same rule:

$$S_1 + S_2 = 2 A/3 \quad (S_1 + S_2 = 6 A) \quad (1.8)$$

Three or four tables may be used for one cane carrier.

Speed of the table. By reason of its width, the speed of the lateral table is made much lower than that of the main carrier. A value of 3 — 6 m/min (10 — 20 ft./min) may be adopted. With a higher speed, there is a risk that too much cane will be dropped on the carrier at a time.

Power required for the motor. A motor should be provided of power roughly equal to

$$T = 0.25 S \quad (T = 0.023 S) \quad (1.9)$$

T = power of the driving motor, in h.p. (h.p.)

S = area of the table, in m² (sq.ft.).

This value is obviously appreciably greater than the mean power absorbed by the motor in operation.

Slope. Lateral tables may be horizontal, or may have a slight forward slope of about 5°, or a slope of 15° towards the rear. This last arrangement allows the length to be extended to the point where tipping trucks can deliver their load directly onto the table, thus avoiding extra handling and loss of time.

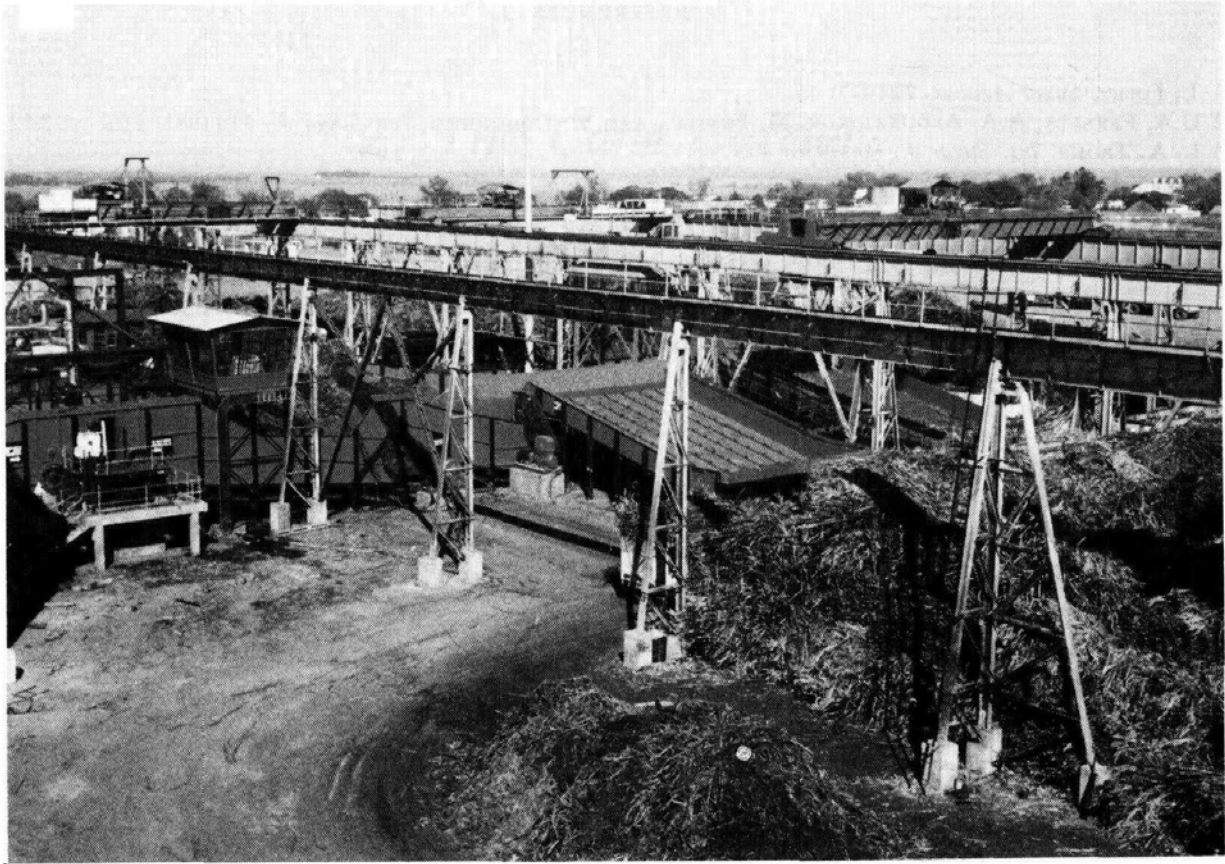


Fig. 1.18. Unloading cane - travelling crane and lateral feed tables (BMA)

2. Auxiliary carrier (cross carrier)

This is a carrier of the same dimensions as the main carrier, but with its axis at right-angles to that of the latter, and discharging to it. It supplies the main carrier in the manner of a regulating tributary which feeds and evens out the main stream.

Speed. It will be driven at a speed about half that of the main carrier.

Power. Since it is subject to somewhat more friction, we may provide, if the conveyor is horizontal:

$$T = 0.34 S \quad (T = 0.05 S) \quad (1.10)$$

T = power absorbed by the auxiliary carrier, in kW (h.p.)

S = area loaded with cane on the auxiliary carrier, in m^2 (sq.ft.).

Construction of cross conveyors

These tables or carriers should move between two lateral steel plates, designed to guide the cane and prevent it from falling beside the carrier. These side plates can conveniently be given a 10° batter.

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2. The Cane Carrier

The cane carrier is the moving apron which conveys the cane into the factory and which assures the feed to the mills by transporting the cane from the yard to the crusher.

Since effective feeding of the crusher requires an elevated hopper, and the cane must be raised to this high level from the level of the yard, where the carrier is generally in a pit, the carrier always includes a sloping portion (Fig. 2.1). We have:

- (a) The horizontal portion
- (b) The inclined portion
- (c) The head where the cane arrives above the crusher.

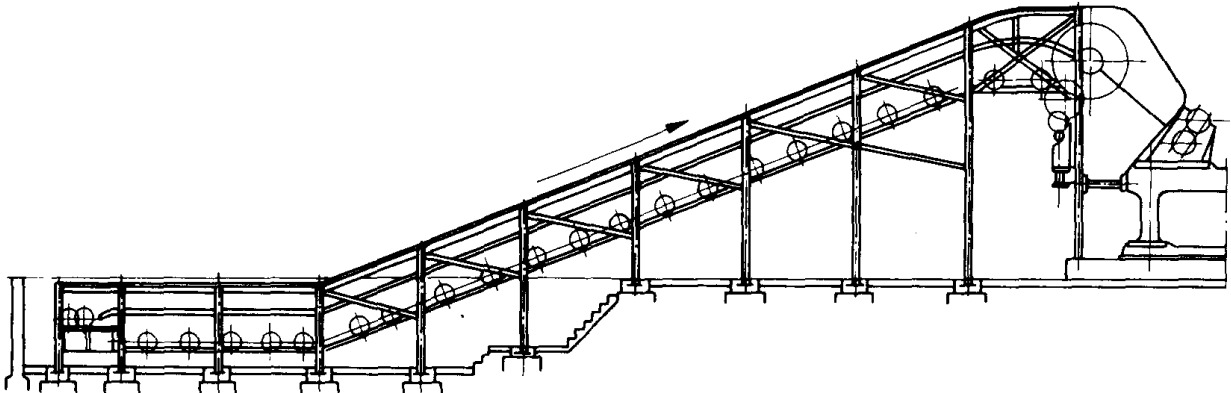


Fig. 2.1. Cane carrier (Fives Cail-Babcock).

Slope

The slope of the inclined portion varies from 27% (15°) to a maximum of 40% (22°). Generally, a value below 27% is not used. If too low a slope is adopted, the cost of the installation and the space occupied are increased. A value of 40% is not exceeded, otherwise the cane would be liable to slip and the conveyor would move forward without picking it up. The best and commonest values are 30% (17°) to 38% (21°).

When no knives are installed, or when the knife set is at the top of the carrier (which amounts to the same thing), it is advisable to keep the slope at 33 or 36%. When one or two sets of knives precede the inclined portion, or are located at the beginning of the latter, a slope of 38% (21°) may safely be adopted, and may even attain 40% (22°) if necessary.

Length of inclined portion

This length will be that necessary to attain, with the inclination adopted, the desired level above the crusher. Generally, the height from the ground level at the mills to the highest point of the carrier will be about:

6-7 m (20–23 ft.) in the case of an ordinary (2-roller) crusher, located above the first mill 4–5 m (13–16 ft.) in the case of a mill-crusher (3 rollers).

Assuming the floor-level of the horizontal portion of the carrier, in the yard, to be 1 m (3 ft.) below the ground level at the mills, and basing our figures on a slope of 36%, the inclined portion of the carrier will have a length of:

(a) With an ordinary crusher:

$$L = \frac{(7-8)}{0.36} = 20-22 \text{ m (64-72 ft.)} \quad (2.1)$$

(b) With a 3-roller crusher:

$$L = \frac{(4-5)}{0.36} = 14-17 \text{ m (45-56 ft.)} \quad (2.2)$$

Length of horizontal portion

The length of the horizontal portion is determined by the width of lateral tables or the space required for other unloading devices.

Width

The width of the carrier is always made equal to the length of the mill rollers. A wider carrier would not feed the crusher uniformly over its width, since it would necessitate a feed hopper in the form of a portion of a pyramid, converging towards the crusher; the ends of the rollers will thus receive a heavier feed than the centre. A narrower carrier would have the opposite disadvantage, which, however, would be less marked on account of the spreading of the cut canes in the chute.

Speed

The speed of the carrier is not fixed in any absolute fashion. However, it is advisable that it be related to the peripheral speed of the mills, for example, half of the latter:

$$u \ll \frac{v}{2} \quad (2.3)$$

u = mean speed of the carrier

v = mean peripheral speed of the mills, in the same unit.

We shall consider how the thickness of cane on the carrier will vary, under this condition. The capacity of the carrier has the value:

$$1,000 A = 60 uLhd \quad (2,240 A = 60 uLhd) \quad (2.4)$$

A = crushing rate, in t.c.h.

M = speed of the carrier, in m/min (ft./min)

L = width of the carrier, in m (ft.)

h = mean depth of the layer of cane on the carrier, in m (ft.)

d = bulk density of the cane on the carrier:

$$\begin{cases} d = 150 \text{ kg/m}^3 \text{ (9 Ib./cu.ft.) (tangled canes)} \\ d = 175 \text{ kg/m}^3 \text{ (11 Ib./cu.ft.) (parallel canes)} \\ d = 300 \text{ kg/m}^3 \text{ (18 Ib./cu.ft.) (cane after knife preparation).} \end{cases}$$

We shall assume the case of two sets of knives and a shredder, the knives located before the sloping portion. We have then:

$$A = 18 L u h \quad (A = 0.536 L u h) \quad (2.5)$$

We shall see later (eqn. (12.6)) that the normal capacity of the mills, A , has a value:

$$A = \frac{0.9 \text{ } c n L D^2 (1 - 0.06 \text{ } n D) \sqrt{N}}{f} \quad \left(A = 0.3277 \frac{c n L D^2 (1 - 0.018 \text{ } n D) \sqrt{N}}{f} \right)$$

A = capacity of mills with knife preparation, in t.c.h.

n = speed of rotation of the mills, in r.p.m.

L = length of rollers, in m (ft.)

D = diameter of rollers, in m (ft.)

N = number of rollers in the tandem

f = fibre per unit weight of cane.

But:

$$v = \pi D n \text{ or: } D n = \frac{v}{\pi} \quad (2.6)$$

v = peripheral speed of the mill rollers, in m/min (ft./min).

Whence:

$$A = \frac{0.285 \text{ } c L v D (1 - 0.02 \text{ } v) \sqrt{N}}{f} \quad \left(A = \frac{0.008 \text{ } c L v D (1 - 0.06 \text{ } v) \sqrt{N}}{f} \right)$$

Substituting this value in eqn. (2.5) and taking $u = v/2$, we have:

$$h = \frac{0.032 \text{ } c D \sqrt{N} (1 - 0.02 \text{ } v)}{f} \quad \left(h = \frac{0.003108 \sqrt{N} (1 - 0.06 \text{ } v)}{f} \right) \quad (2.7)$$

h = being in m (ft.)

c = coefficient for preparatory plant.

If we neglect the correction term $(1 - 0.06 v)$, which varies only slightly with v and causes a slight decrease in the cane layer with increasing mill speeds, we see that the thickness of cane, for normal operation of the mills, will vary:

Inversely as fibre in cane
 Directly proportional to roller diameter
 Proportional to the square root of the number of rollers.

If, instead of a carrier speed proportional to that of the mills, we had adopted a constant speed, independent of that of the mills, for example, 5 m/min (15 ft./min), we would have found that h would vary as:

$$\frac{nD^2 \sqrt{N}}{f}$$

that is, the thickness of the cane layer would be proportional to the square of the roller diameter, and we would arrive at excessive thicknesses in the case of large mills and long tandems, working at high speeds.

For this reason it is preferable to relate the speed of the cane carrier to the mean peripheral speed v of the mill rollers. The thickness of the cane layer is then proportional to D , which is logical since it varies proportionally to a linear dimension of the mill.

In general, we adopt:

$$u = 0.3 - 0.5 v \quad (2.8)$$

Carrier slats

Cane carrier slats were previously made of wood. This material is no longer found except in certain lateral tables; it is unsuitable for use with knives, on account of the stresses imposed on the carrier apron in its passage below the knives, and especially because pieces of cane would fall through the spaces between the boards.

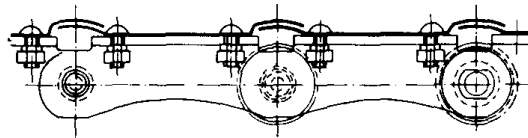


Fig. 2.2. Steel carrier slats (Fives Cail - Babcock).

Carrier aprons (Fig. 2.2) are made up of steel slats which overlap by a rounded edge in the form of a cylinder concentric with the rollers which carry the chain and the pins which join successive links of the chain together. This rounded edge permits a passage of the slats over the sprockets at the head and foot of the carrier, while maintaining the overlap, and gives the apron the necessary flexibility.

The apron is generally supported by two chains, sometimes three in very wide mills.

Power

The mean power absorbed by the cane carrier consists of two terms:

(a) The power necessary to overcome friction. It has the value:

$$P_f = \frac{(Q + K)f + Kf'}{60 \times 75} u\lambda \quad \left(P_f = \frac{(Q + K)f + Kf'}{60 \times 550} u\lambda \right) \quad (2.9)$$

P_f = power necessary to overcome friction, in h.p.

Q = weight of cane on the carrier, in kg (lb.)

K = weight of the upper run of the conveyor, in kg (lb.)

= weight of the lower run

= half the total weight of the moving part of the carrier

f = coefficient of friction of the upper run, of the order of 0.30

f' = coefficient of rolling friction of the lower run, of the order of 0.15

u = speed of the conveyor, in m/min (ft./min)

X = coefficient of the order of 1.4- 1.5, allowing for the efficiency of the gearing transmitting the drive.

The values given for f and f' take into account all sources of friction, as well as allowing for a certain proportion of rollers which do not rotate, or do not rotate freely. They also allow for the difference between the rolling friction of the upper run, on all the rollers, and the lower run, on loose pulleys of large diameter. If the upper run slides, instead of working on rollers, it would be necessary to take $f = 0.60$. The theoretical coefficient for rolling friction is 0.15.

The coefficients 0.30 and 0.15 take into account the inclined portion of the carrier, which theoretically should be calculated separately, taking into account the cosine and sine of the angle of inclination. However, the margin of error in the coefficients of friction makes it superfluous to introduce this refinement.

The term:

$$\frac{(6 + K) fu}{60 X 75} \quad \left(\frac{(Q + K) fu}{60 x 550} \right)$$

represents the friction of the upper run under load.

The term:

$$\frac{Kf'u}{60 x 75} \quad \left(\frac{Kf'u}{60 x 550} \right)$$

that of the lower run, empty.

Also, we have:

$$Q = Z_c L h d \quad (2.10)$$

Z_c = loaded length of the carrier, in m (ft.)

(for safety we take the full length of the apron)

L = width of the carrier, in m (ft.)

h = thickness of the cane layer on the carrier, in m (ft.)

d = bulk density of the cane on the carrier (cf. eqn. (2.4))

h = is given by eqn. (2.5).

We have also more simply:

$$Q = \frac{1,000 AZ_r}{60 u} \quad \left(Q = \frac{2,240 AZ_c}{60 u} \right) \quad (2.11)$$

On the other hand:

$$K = \frac{Z_t}{2} (2p + p') \quad (2.12)$$

Z_t = total length of the carrier apron, in m (ft.)

p = weight per unit length of the actual chain, in kg/m (lb./ft.) (we assume the slats are carried by 2 chains)

p' = weight of the apron (or slats) per unit length of the carrier, in kg/m (lb./ft.)

We shall take:

p = 18 – 30 kg/m (12-20 lb./ft.) according to the type of chain

p' = 32 – 40 kg/m per m of width (*i.e.* 8 – 10 kg/m²) (8 - 10 lb./sq.ft.) for steel slats. Hence, (48 – 60 lb./ft. length) for a 2 m (6 ft.) carrier.

(b) The power P_e necessary to elevate the cane:

$$P_e = \frac{1,000 AH}{3,600 \times 75} \quad \left(P_e = \frac{2,240 AH}{3,600 \times 550} \lambda \right) \quad (2.13)$$

P_e = power necessary to elevate the cane, in h.p.

A = crushing rate of the mills, in t.c.h.

H = difference in height between the highest part of the carrier, above the crusher, and the horizontal part of the carrier in the yard, in m (ft.)

X = coefficient depending on the efficiency of the gearing, varying from 1.4 to 1.5.

Finally:

$$P = P_e + P_f$$

P = mean total power absorbed by the carrier, in h.p.

Roughly, we may reckon, as a first approximation,

$$P = \frac{3 Z_t + A}{20} \text{ h.p.} \quad \left(P = \frac{Z_t + A}{16} \text{ h.p.} \right) \quad (2.14)$$

Z_t = total length of the carrier, in m (ft.)

A = crushing rate of the mills, in t.c.h.

The installed power should be appreciably higher, say:

$$P_i = \frac{3 Z_t + A}{10} \text{ h.p.} \quad \left(P_i = \frac{Z_t + A}{10} \text{ h.p.} \right) \quad (2.15)$$

Carrier drive

The cane carrier is always driven independently, by electric motor. As this is essentially a constant-speed machine, and the carrier speed must be frequently varied, the drive is effected through a variable-speed coupler.

Variable-speed coupling using Foucault current. Some organisations, particularly the firm of "Heenan and Froude" in England, supply couplings which may be interposed between the motor and the driven gears, permitting a very progressive and flexible speed variation of the driven shaft from 0 to 100%. This drive has already been mentioned in connection with the drive for lateral tables (p. 17). A flywheel, driven by the electric motor, turns freely inside a sleeve. The latter, by means of gears or belts, drives the shaft at the head of the carrier. By means of a suitable control, actuated by a knurled knob or a hand wheel, Foucault currents can be induced in the flywheel, and their intensity varied at will. As these currents are increased, the drive between flywheel and sleeve becomes more and more positive. Thus any desired degree of slip can be obtained between these two elements, and consequently any desired speed of the driven shaft.

This method is neat and flexible; its cost is not excessive. It may be employed, not only for driving the cane carrier, but also for lateral tables and auxiliary carriers. In addition to hand control, the coupling may be controlled by an electric motor, *e.g.*, by one phase of the knife motor. In this way, the speed of the carrier is automatically reduced when the knives are loaded, and increased in the opposite case; this serves to avoid chokes at the knives and to assure a regular feed to the mills.

In place of electrical control, speed variation may be effected by hydraulic motor (see p. 37).

Tension

The free end of the carrier is fitted with an adjustment permitting the tension of the conveyor to be adjusted. Since the drive is located at the head of the carrier, and the resistance to the movement is produced by the loaded upper run of the carrier, it is the lower run which will be slack. Care is necessary to ensure that this slack is neither too much, nor too little. The

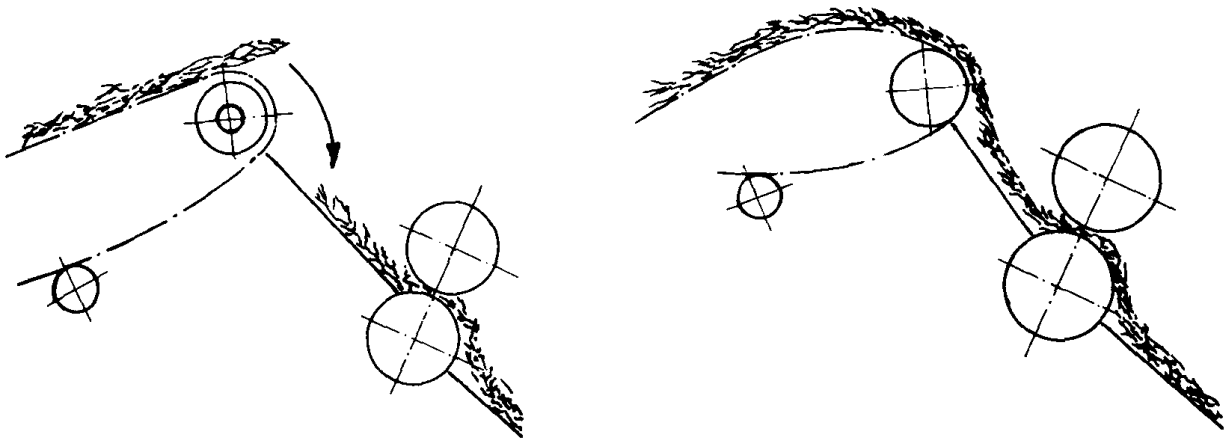


Fig. 2.3. Effect of shape of carrier head. Straight head. Fig. 2.4. Effect of shape of carrier head. Curved head.

tension moreover need not be great; the lower run is allowed to form fairly pronounced curves between the idler pulleys.

Form of head of the carrier

The knives sometimes leave a certain proportion of uncut sticks at the bottom of the cane layer. These sticks form a kind of grating at the bottom of the layer, which supports the chips and pieces of cut cane and prevents them from falling continuously from the top of the carrier.

If the carrier were to terminate abruptly, the inclined portion finishing right at the top sprockets, the cane reaching the upper end of the carrier in this way would first jut out beyond the carrier and then fall in batches into the feed hopper of the crusher (Fig. 2.3) or mill. The feed of the crusher would be irregular, jerky and defective.

To avoid this disadvantage, the head of the carrier is designed with a gradual curve, tangential to the sloping portion, so that the surface of the carrier becomes first horizontal, then slopes downwards until it almost reaches the angle of repose of cane (or cut pieces of cane). The cane will continue to fall in batches, but these batches are smaller and more numerous, and the intervals between them more uniform. The cane may even form a continuous blanket from the carrier to the crusher (Fig. 2.4) or the mill.

It is necessary therefore to give careful attention to the form of the head of the carrier.

Cane elevators

When space is restricted, the inclined portion of the cane carrier may be replaced by a cane elevator. This is a similar piece of equipment, but designed to lift the cane at a much steeper angle (Fig. 2.5). It operates on cut canes, since the elevator is always located after at least the first set of knives, and generally after both sets of knives.

The slope may thus be increased up to 60° . In general it is maintained between 35 and 40° . However, above 22° , the conveyor is no longer an ordinary carrier and it is desirable to make provision for preventing slip of the pieces of cane: angle irons, spikes, or hooks placed on the slats to give a positive grip on the cane.

Such elevators have, almost inevitably, the disadvantage described above; they end abruptly, and it would be difficult to give them a suitable curved form at the top.

Speed. The elevators are generally given a speed of 6- 10 m/min (18- 30 ft./min). For preference, a value will be chosen:

$$u = 0.3-0.5 v \quad (2.16)$$

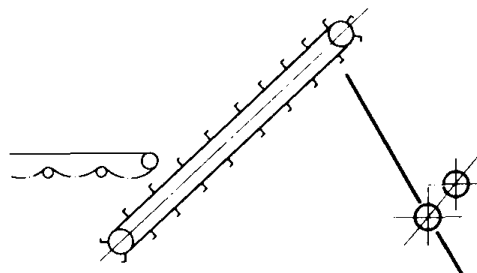


Fig. 2.5. Cane elevator.

u = speed of the elevator
 v = mean peripheral speed of the mills.

Power. The power required is calculated for elevators as for carriers. Since the inclination is no longer negligible, the term P_f in eqn. (2.9) should, however, be multiplied by the cosine of the angle of elevation and will consequently be reduced proportionately.

Equalisers

The term "equaliser" is used to describe an apparatus designed to even out the distribution of cane on the carrier, and to level the layer of cane to a certain extent (Fig. 2.6). This must not be confused with the "leveller", which is a set of knives, and which evens out the layer of cane while cutting it.

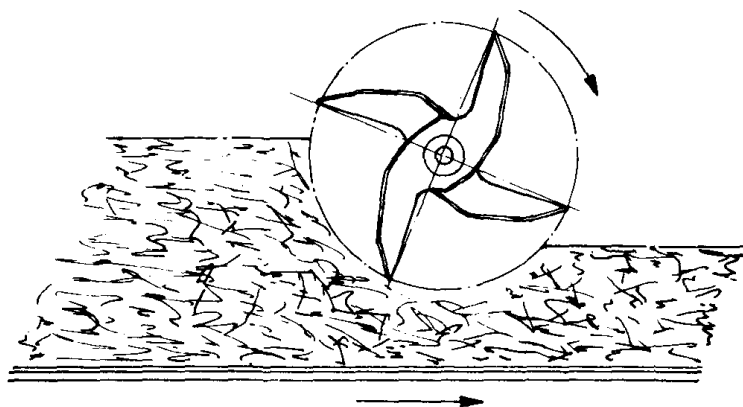


Fig. 2.6. Equaliser.

Description. An equaliser consists of a shaft placed across the carrier, carrying curved arms, and rotating in such a direction that the arms passing nearest the carrier boards move in the opposite direction to the latter. Thus the direction of rotation of the equaliser is such as to throw the cane backwards (Fig. 2.6), whereas the cane knives generally rotate in such a direction as to throw the pieces of cane forward (Fig. 3.1).

Use. An equaliser is installed when the cane is very tangled on the carrier and when the levelling knives, set fairly low, would themselves be unable to level the cane adequately. The equaliser is thus a secondary and inexpensive piece of equipment, placed after the knives, and designed to lighten their duty. It is not often found since the use of a second set of knives has become general.

Speed. An equaliser should rotate at about 40-50 r.p.m.

Clearance. The shaft will be set at such a height that the distance between the ends of the arms and the carrier slats is slightly less than the height h given by eqn. (2.5), in which d may be taken as 130 kg/m^3 (8 lb./cu.ft.).

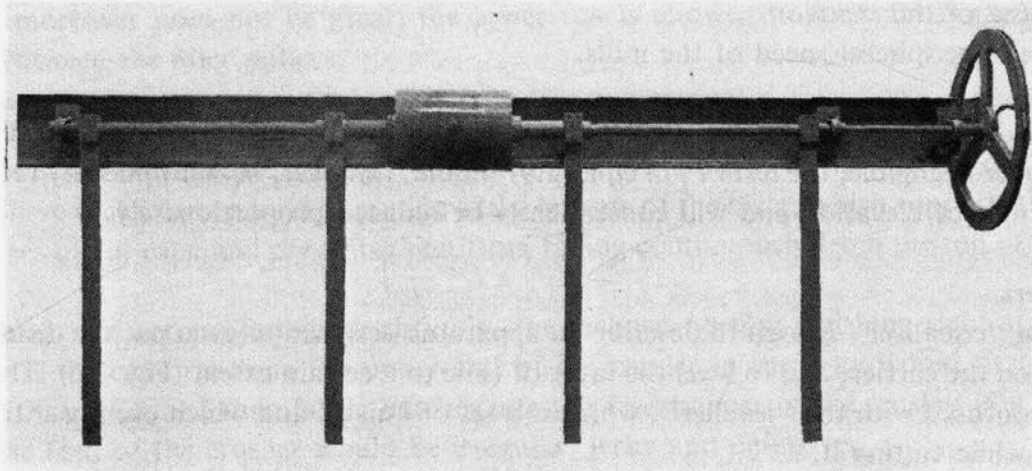


Fig. 2.7. "Auto-cane". Feeler (Edwards Eng. Corp.).

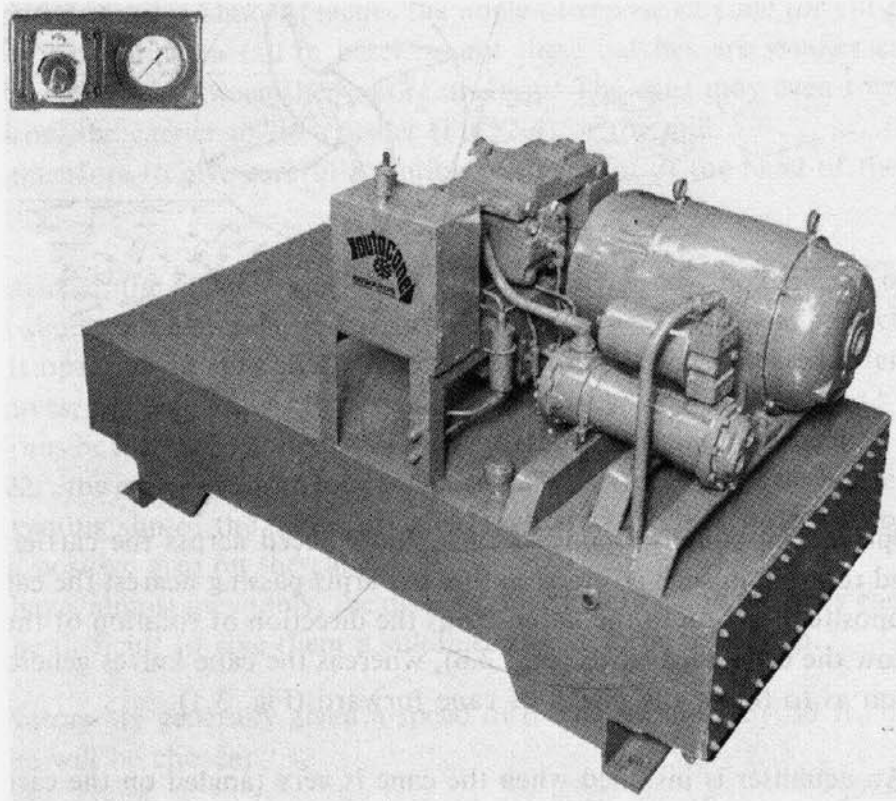


Fig. 2.8. "Auto-cane" Drive assembly, with motors, filter, hydraulic transmission, speed control and oil reservoir (Edwards Eng. Corp.).

Power. A motor of power

$$P = 0.15 A \quad (P = 0.2 A) \quad (2.17)$$

p = power of the driving motor, in kW (h.p.)

A = crushing rate, in t.c.h.

will generally suffice to cope with the frequent overloads caused by the passage of compact bundles of cane deposited on the carrier by the crane.

Automatic control of speed

Most factories in recent years have installed their own system for ensuring the regular feed of cane into the milling plant, or have adopted a system offered with the same object by a machinery manufacturer.

The best known system for such regularisation of cane feed is the “**Autocane**”, made by the Edwards Engineering Corporation.

It comprises a detector, a hydraulic motor, and control equipment.

The detector (Fig. 2.7) consists of a number of bars, one end of which rests on the layer of cut cane after the knives, while the other end is pivoted on a shaft and actuates a sleeve mounted concentrically on that shaft. Variations in the height of the cane layer are thus registered in and totalised, and furnish a signal which is transmitted to a motor. The latter drives an oil pump the output of which drives a hydraulic motor which in turn drives the shaft at the head of the carrier. The speed of the latter is thus maintained inversely proportional to the thickness of the layer of cut cane, in such a way that the volume delivered by the carrier remains constant.

The control equipment allows of adjustment of speed to suit the desired tonnage. This can be regulated from 25 to 100% of its maximum capacity.

The detector can be used only on the cut cane after passing through the knives; if a shredder is in use, the result can indeed be better. The power of the electric motor is 8- 16 kW (12 – 25 h.p.) depending on the capacity of the tandem.

The "Autocane" and similar devices assure a very uniform feed, eliminating the necessity for an operator placed at the carrier, and leading to a gain in crushing rate, thanks to the regularity of feed and the elimination of chokes.

3. Cane Knives

Object and uses

Knives are not an indispensable item of equipment in the sugar factory, in the sense that it would be possible to operate without them. Before 1920, many factories did not possess knives. However, this equipment gives such an improvement in feeding that today it is no longer a question of doing without them. A knife set, for a factory which does not possess one, would very soon pay for itself.

With whole cane, the attempt at obtaining a completely even feed to the mill is never really successful. By placing an "equaliser" (see p. 29) over the carrier, a cane layer of nearly uniform thickness may be obtained. However, this layer will reach the mill only in successive lumps, and its small bulk density will render difficult the work of the crusher, which consists, in short, of "swallowing" as much cane as possible in a given time.

Furthermore, the metal of the mill rollers often slips on the smooth, waxy and polished rind of the cane; hence chokes occur, with short or long interruptions to the feed or a decrease in crushing rate. The knives, on the other hand, supply the cane in very short and small pieces. Whereas whole canes lean one against the other, arching over and leaving voids between them, these small pieces settle together into a compact mass, which drops easily into the feed hopper, and which the crusher will grip without trouble and will absorb in a continuous manner.

Between whole canes and canes which have passed through the knives, there is the same difference as between a handful of matches and a handful of sawdust. The difference existing between these two states of the same material can be readily realised. It is shown in their respective bulk densities:

Whole canes more or less tangled:	125 – 150 kg/m ³ (8 – 9 lb./cu.ft.)
Canes passed through knives:	250 – 300 kg/m ³ (15 – 18 lb./cu.ft.)

The cane knives then perform two functions and have two advantages:

- (a) They favour the *capacity* of the mills
- (b) They assist the *extraction* of the mills by breaking the rind of the cane and so facilitating its disintegration and the extraction of its juice.

From the point of view of its resistance to pressure, cane can in fact be compared with a long cylinder reinforced with transverse dividing walls (the nodes). Accordingly, it presents to the mill a resistance similar to that of a stick of bamboo, the internodes of which have been filled with a sugar-bearing pulp (cane is moreover very closely related to bamboo and successful attempts have been made to cross these two plants and so obtain hybrids). The pulp in the interior of these cylinders, which lie end to end, will be more rapidly recovered as the cylinders are opened and the dividing walls broken.

However, the value of knives for the two functions described above is very unequal:

(a) From the point of view of *capacity*, nothing can replace them. If there are no knives the crusher can only take what it succeeds in gripping out of the light and bulky mass of whole canes.

(b) From the point of view of *extraction*, on the contrary, the crusher and the mills have precisely the object of breaking up the cane, and crushing the casing of its rind and the hard structure of its nodes. It is possible then to carry out this function of the knives by good work at crusher and mills. In particular, the gain obtained from a knife installation from this point of view is appreciably inferior to the gain which one would obtain by the addition of a further mill.

It is then predominantly to *improve the capacity* of the mills, that knives are generally installed.

Chopped cane. The preceding remarks apply to the general case of cane delivered as whole stalks. If the factory receives all or nearly all of its cane as cut pieces from chopper-harvesters, the case is very different and the main reason for using knives disappears. In Australia, for example, the cane supply is 100% chopper-harvested; the pieces moreover are clean-cut, straight and free of ragged ends. Australian factories take advantage of this and have completely abandoned knives. The carrier thus delivers the cane direct to the shredder, which becomes the only preparation device. It is a high-power, heavy-duty machine, since the chopped cane offers more resistance than normal knife-prepared cane.

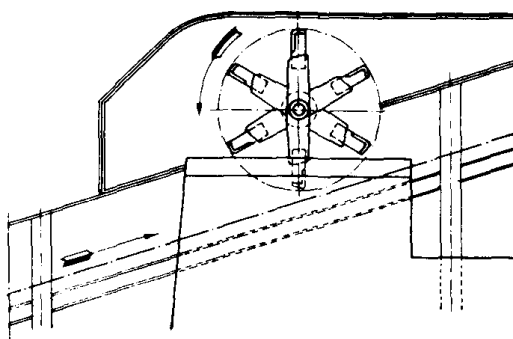


Fig. 3.1. Cane knives.

Description

Standard type. We show (Fig. 3.1) a standard type of knife installation. It consists of a heavy shaft of hexagonal or octagonal section mounted on roller bearings, and on which are threaded or fixed arms each carrying two blades symmetrically placed with relation to the shaft.

The second arm is offset 60° relative to the first, if the shaft is hexagonal; the following one is offset a further 60° and so on. In this way, if there are 36 arms, for example, the knife installation will consist of 72 blades distributed in rows of 12 in 6 different axial half-planes (or 3 planes). In order to reduce the pitch and increase the number of cutting planes, the arms

carrying the knives are so fabricated that the two blades on each arm rotate in two different planes separated by half a pitch. This doubles the number of cutting planes.

Viewed from the front, the 72 blades then rotate in 72 planes perpendicular to the axis, or so-called "cutting planes", consisting of so many circles of rotation.

The term "pitch" of knives is the mean distance, measured parallel to the axis, separating two successive circles of rotation.

The knife blades are removable, so that they can be easily taken off for resharpening or for replacement by another set of blades.

It is preferable to fix a knife to the arm in such a way that the edge affected by shocks is supported by a flange or a wide stirrup rather than fixing it so that shocks are absorbed by the bolts or pins securing the knife. In the latter case, the bolt hole becomes ovalised and some play develops, accentuating the effect of shocks; the hole thus forms a section of weakness in the knife, leading to breakage.

Other designs. Apart from this standard type, certain manufacturers have introduced various modifications of detail, the principal ones of which have the objects of:

- (1) Prolonging the working life of the blades by diminishing wear of the cutting edge.
- (2) Producing a self-sharpening effect of the knives on the trailing edge, and so rendering them reversible (Figs. 3.2 and 3.3).
- (3) Avoiding reduction of the length of the blades with wear.

With the Mirrlees – Watson bent or hoe knife, not only does the radius of action remain constant with wear, but one also obtains the effect of a transverse cut due to the bent end of the knife (Fig. 3.4); and this transverse cut is superimposed on the longitudinal cut made by the radial part of the knife.

- (4) Diminishing or reducing the consequences of shocks, by mounting the knives so as to swing on a pin, obtaining at the same time a slight displacement of the cutting points (Fig. 3.5).

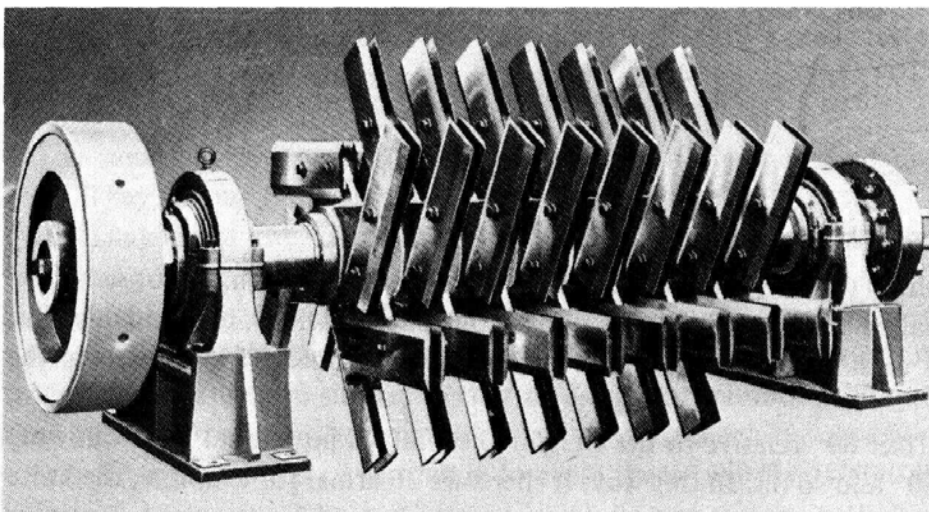


Fig. 3.2. Knife set with twin reversible blades (Fives Cail – Babcock).

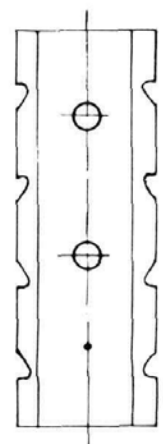


Fig. 3.3. Reversible blade.

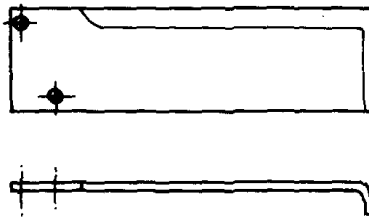


Fig. 3.4. Mirrless "Hoe" knife.

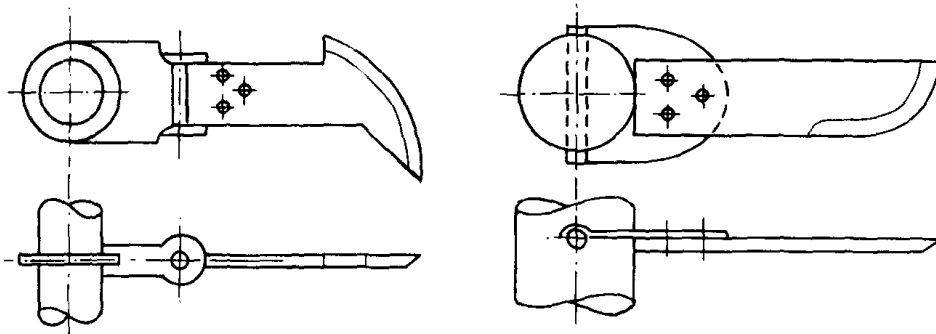


Fig. 3.5. Swinging knives.

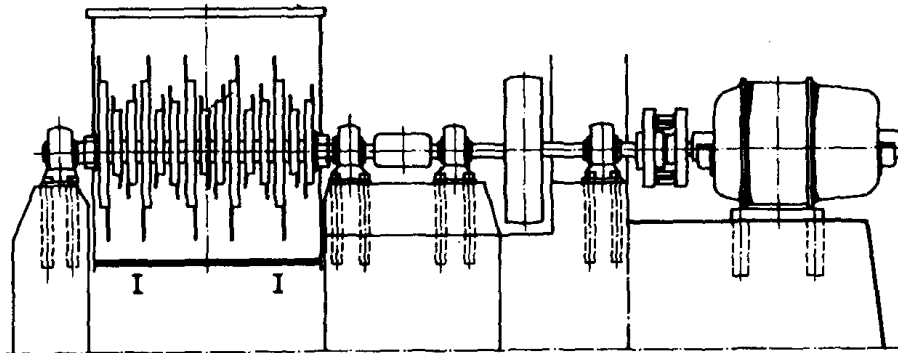


Fig. 3.6. Knife drive by electric motor (Fives Cail - Babcock).

Effect on cane carrier

Cane carriers were previously furnished with wooden slats. It is the adoption of knives which has necessitated the provision of steel slats, since the wooden boards would be damaged under the knives, and chips of cane would pass through the spaces between the carrier boards.

Even with steel slats, it is necessary to reinforce the carrier where it passes under the knives; a reinforcing beam is placed under each rail supporting the carrier chains, and an angle iron support under the ends of the carrier slats.

Methods of drive

Knives may be driven by two different sources of power:

- (1) Electric motor
- (2) Steam turbine.

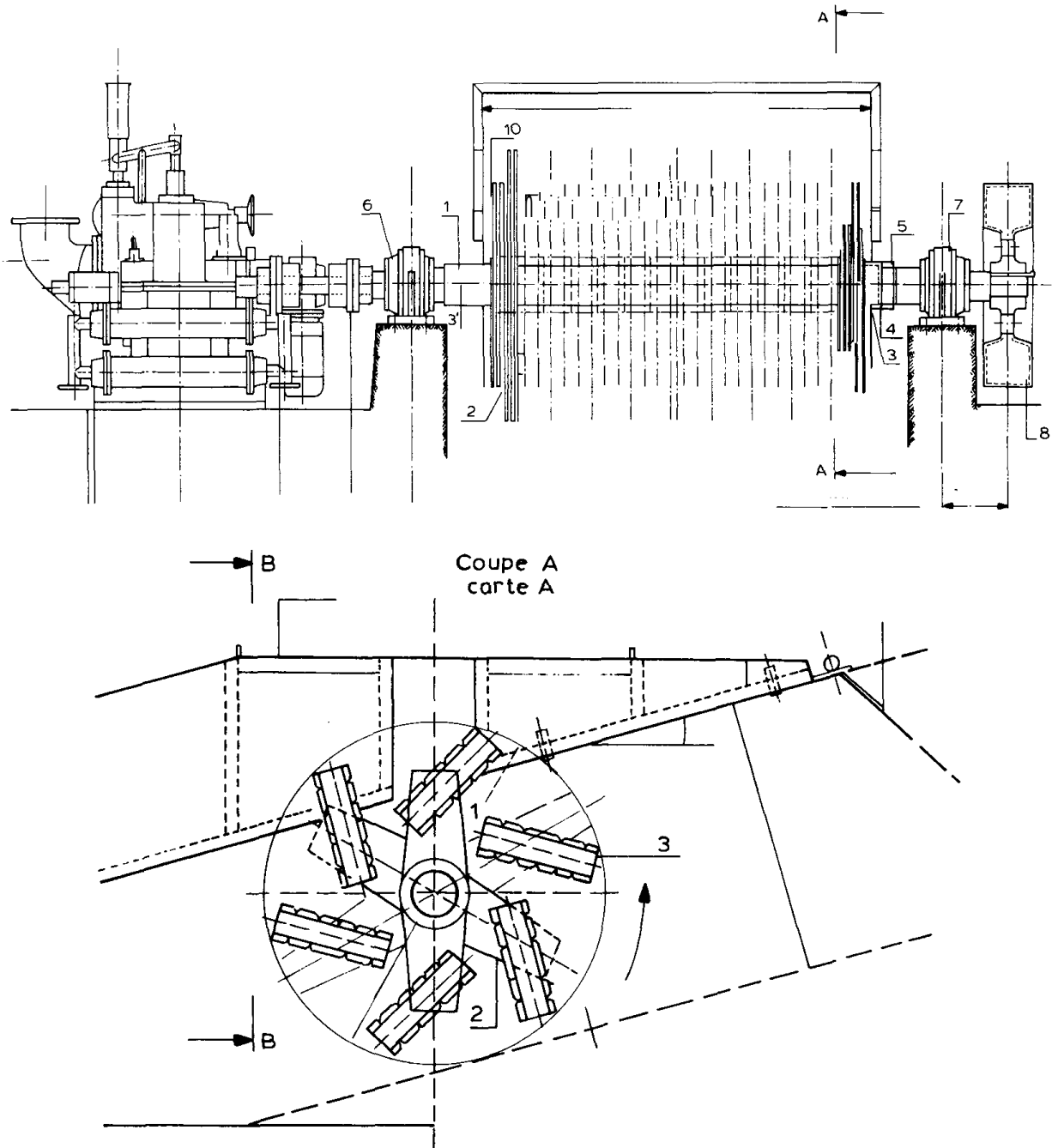


Fig. 3.7. Knife drive by steam turbine.

(1) Electric drive is effected by a wound rotor induction motor with slip-rings and with or without provision for lifting the brushes (Fig. 3.6).

(2) Drive by steam turbine is the most attractive (Fig. 3.7). It permits some degree of speed variation, but is expensive, as the cost of the turbine sometimes exceeds twice that of the electric motor. Moreover, it necessitates either a special attendant or at least partial supervision, and it does not warrant consideration except for large tandems above 150 t.c.h.

As regards transmission of the drive, we may distinguish: (a) belt drive, (b) direct drive through a flexible coupling.

(a) Belt drive originally had the prime object of avoiding too drastic shocks on the knives; in case of an overload, the belt would come off. Actually this advantage is very slight and it is sometimes just as much trouble to have to replace the belt which has come off, as to replace a broken knife blade. In any case, if it is a matter of a piece of tramp iron passing through with the cane the belt comes off but, generally, the knife is broken just the same. The only advantage is that, having stopped the mill, the knife and the piece of iron can be more easily recovered; or indeed the driver of the crusher is alerted and is more likely to see them drop into the feed hopper.

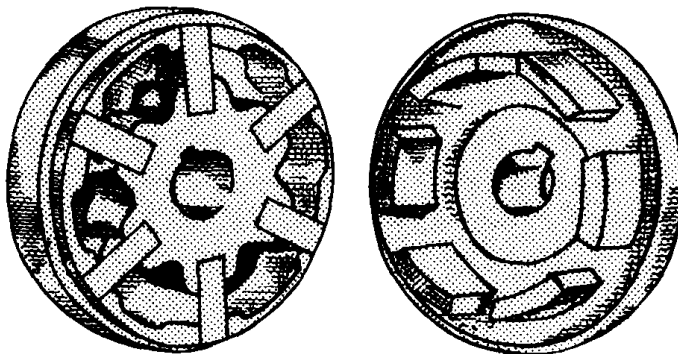


Fig. 3.8. Flexible coupling.

(b) In modern practice the preference is generally for direct drive through a flexible coupling (Fig. 3.8). The knife assembly is connected directly to the motor shaft by the simple interposition of a device which is relatively flexible and consists of a cord or rope laced alternately between pins or studs fixed in two discs, the driving disc being attached to the motor, the driven disc to the shaft assembly of the knives. This arrangement is much simpler and more economical, and gives every satisfaction, provided that the motor is sufficiently powerful.

Belt drive is now retained only for connecting two adjacent sets of knives.

Pitch of knives

We have already seen that the pitch of the knives is the interval separating the circles of rotation or the cutting planes of two adjacent blades.

Generally, 2 or 3 sets of knives are installed, the first of 50 mm (2 in.) pitch, the second (and third) of 20—22 mm (just below 1 in.).

Number of blades

The pitch fixes the number of blades. In the case where each knife turns in a different plane we have:

$$N = \frac{L}{p} - 1 \quad (3.1)$$

N = number of knives

L = width of carrier, in cm (in.)

p = pitch of knives, in cm (in.).

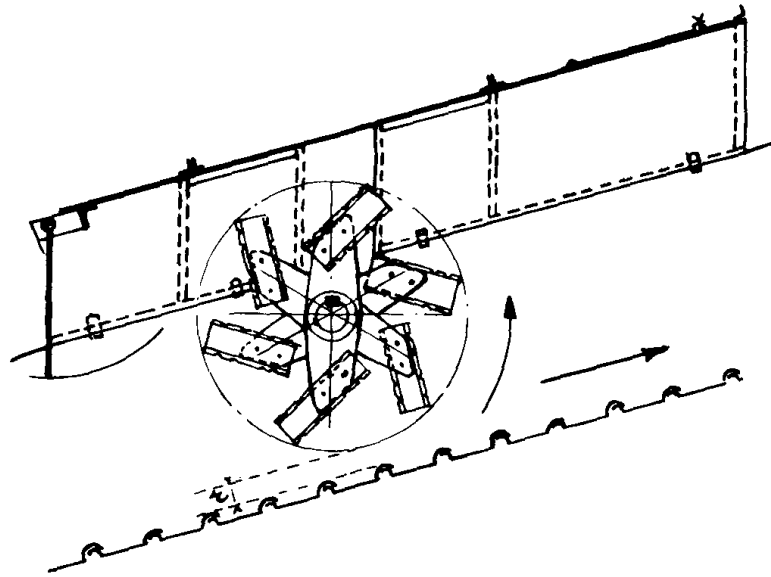


Fig. 3.9. Clearance of knives.

When eqn. (3.1) gives an odd number for N , the even number immediately lower is always adopted; the interval between the extreme blades and the sides of the carrier will then be greater than the pitch.

Clearance

The clearance of a knife set is the distance r between the circle described by the extremity of the knives and the plane passing through the highest portions of the carrier apron (Fig. 3.9).

The clearance is an important factor in the operation of a knife set. On it depends the proportion of cane cut and consequently, to some extent, the efficiency of the knives.

The clearance may be adjusted by raising or lowering bodily the steel frame supporting the carrier below the knives. However, it is preferable to adjust it by moving the knives and their driving motor on horizontal slides. Since the knives are placed (precisely with this object) on the inclined portion of the carrier, the clearance can thus be set at will to the desired value.

Clearance may also be adjusted by placing packing pieces under the knives and motor.

Proportion of cane not cut

This is immediately obtained:

$$i = \frac{r}{h} \times 100 \quad (3.2)$$

i = weight of uncut cane, % of weight treated

r = clearance of knives, in mm (in.)

h = height of cane layer, in mm (in.).

Remember that h is given in m (ft.) by eqn. (2.4):

$$h = 60 uLd \quad \left(h = \frac{60 uLd}{100} \right) \quad (3.3)$$

A = crushing rate of mills, in t.c.h.

u = speed of carrier, in m/min (ft./min)

L = width of carrier, in m (ft.)

d = bulk density of cane fed to the knives, in kg/m³ (lb./cu.ft.):

$d = 150 \text{ kg/m}^3$ (9 lb./cu.ft.) (tangled cane)

$d = 175 \text{ kg/m}^3$ (11 lb./cu.ft.) (parallel cane)

$d = 300 \text{ kg/m}^3$ (18 lb./cu.ft.) (cut cane, in the case of 2nd knife set).

These values of d should be somewhat higher than those indicated by eqn. (2.4) since it is actually a question of the density of cane as it passes through the knives, and since the knives, as they strike the cane, pack the cane to some extent on the apron of the carrier.

The proportion of cane cut is obviously:

$$K = \frac{100 - i}{100} \quad (3.4)$$

It is not easy to determine accurately the mean speed u for the cane carrier. The best way is to measure the total length of the carrier apron, to mark with paint one link or one slat, and to determine the time for this link to return to a given point, for example the head of the carrier, opposite a mark made for this purpose. If the total length of the apron is Z_t , and if the marked slat takes t minutes to complete its course, the average speed will be:

$$u = \frac{Z_t}{t} \text{ m/min (ft./min)} \quad (3.5)$$

It will be advisable to take the mean of 3, 5 or 10 complete cycles of the carrier.

Direction of rotation

The almost universal practice is to arrange the knives to rotate in such a manner that the knives at the lowest part of their rotation will move in the direction corresponding to the movement of the carrier (Figs. 3.1 and 3.9). However, some factories arrange for one or sometimes both sets of knives to rotate in the inverse direction. This is not always easily done; it augments the cutting effect on the cane but at the same time increases the power requirements. The knives should always be installed either at a high clearance or in such a way as to throw the pieces of cut cane upwards.

Knife blades

Knife blades should be in special steel. FCB supplies them in steel of type 55 SCD 8 with C-Si-Mn-Cr-Mo, heated, tempered and hardened, with Brinell hardness of 360/400

HB. When the knives are subject to wear due to sand and stone, carbon steel XC 32 may be used; this can be resurfaced in place by application of tungsten or stellite.

Balancing. A set of knives must be well balanced. In order to avoid lack of balance when the knives are reassembled, at the beginning of a season or after re-sharpening, it is necessary to weigh the blades and replace them in pairs such that on each arm carrying two opposed blades, the weight of these blades is practically equal. In the same way, when a knife has to be replaced, two opposite blades should be removed, and replaced by two new or used blades, of equal weight.

Maintenance. A mechanic should be detailed and charged with the job of inspecting the knife set every week at the time of the weekly shut-down, in order to check whether there are any loose bolts, or loose, ovalised or cracked knives which should be changed.

Method of utilising knives

Maxwell¹ distinguishes two types of knives, according to the role which is demanded of them:

(a) The leveller knives which are required mainly to even out the layer of cane. They are arranged to work with a high clearance and in consequence leave a large proportion of uncut cane.

(b) The cutting knives. This second effect is sought only when two sets of knives are installed: the first, at the bottom of the sloping portion of the carrier, would play the role of levellers, the second, immediately after the first or at the head of the carrier, would then have the function of finishing the complete disintegration of the cane, and would be used at a very small clearance (Fig. 3.10).

This distinction, we fear, is somewhat artificial. A single set of knives performs a task considerably more important than simple levelling of the cane layer. It "minces" the cane into small pieces which cover up and fill in the gaps in the feed and so improve the feeding of the crusher. Many single sets of knives are employed, with such a clearance that they are in reality *cutting* knives. However, the general practice is to install at least two sets of knives.

Finally, we consider that the above distinction has no significance except when it is a matter of describing the effective roles of two sets of cane knives working on the same carrier. It is then rather appropriate.

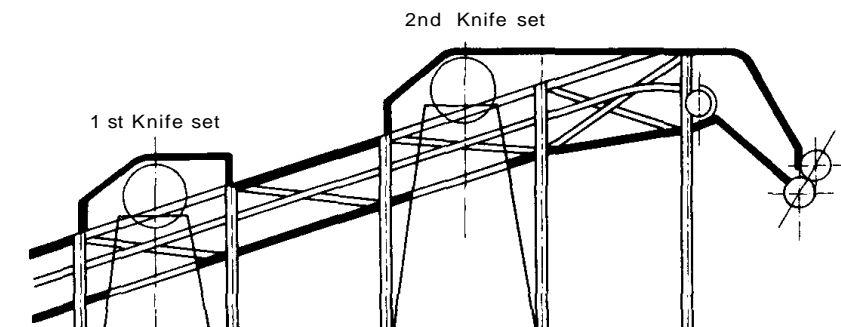


Fig. 3.10. Installation of two sets of knives.

The second knife set. We have seen the great value of a set of knives, on account of its favourable influence on the feeding and the capacity of the mills. To obtain the full effect desired, it is necessary to have two sets of knives at least. One set cannot reduce the thickness of the cane layer sufficiently nor can it cut the cane sufficiently fine. This action is achieved by the second set of knives, which cuts the lower layer which the first set has not touched, and delivers the cut cane in smaller and thinner pieces. The second set thus facilitating the feeding of the mills:

(7) From the point of view of extraction, according to several tests, the gain obtained by a second set is found to be small and difficult to detect.

(2) From the point of view of capacity, its effect is much more definite, although obviously appreciably less than that of the first set. We can reckon approximately:

	<i>Without knives</i>	<i>With 1 set</i>	<i>With 2 sets</i>
Relative capacities	1	1.15	1.20

In other words, the first set of knives gives a gain of approximately 15%, the second adds another 4 or 5%. This is indeed much less but, taking into account the costs of purchase and upkeep of a knife set, it is still very profitable. The second set is particularly useful with a short tandem, since it is then desirable to present to the mills pieces of cane as finely divided as possible, in such a way as to enable the earlier mills to obtain the maximum extraction. It is with tandems of 5 mills (say 15 rollers) or more that Maxwell² advised against a second set; he considered on the contrary that with 4 mills (12 rollers) and less, the second set could be useful. We think that if his comments, dating from 1932, had been written 30 years later, his conclusion would have been different and two sets of knives would have been recommended in all cases.

The third knife set. It is difficult to assess the gain in capacity achieved by a third set of knives. It could perhaps be expressed by inserting a coefficient of 1.22 in the above formula for capacity. It is certainly a small gain only, but a third knife set is fairly widely used. It completes the work of preparation and permits attaining a better index of preparation (*cf. p. 61*). Cane preparation today is considered so important that the installation of a third set of knives does not completely rule out the addition of a shredder as the fourth preparatory device.

The shredder, however, has one advantage over knives. In countries where numerous stones occur in fields and mechanical loading allows them to arrive at the mill, they cause much less damage to the shredder than to the knives. However, the knives are unfortunately the first in line, and receive all the stones which have escaped the means of detection and so suffer considerably. The first knives thus detect the stone, the carrier is stopped, the following knives and shredder are protected, but several blades have been damaged and these accidents are very expensive, owing to the stoppage, the loss of time and the loss of knife blades.

It is thus very important to take all possible measures to detect stones before the first set of knives. The factory of Bois-Rouge, in Reunion, which crushes at 100 t.c.h., has 3 sets of knives, arranged and driven as follows:

Knives	Number of blades	Pitch		Speed (r.p.m.)	Clearance		Nominal power		Mean power absorbed	
		(mm)	(in.)		(mm)	(in.)	(kW)	(h.p.)	(kW)	(h.p.)
1st	32	50	2	480	305	12	130	175	55	75
2nd	32	50	2	480	10	$\frac{3}{8}$	130	175	75	100
3rd	80	22	$\frac{7}{8}$	640	6.5	$\frac{1}{4}$	255	340	90	120

The value of the third set is debatable: it does not give an appreciable gain in capacity, but its supporters assert that it assures that a high extraction can be regularly maintained. Its effect is often compared with that of a shredder. It would be particularly indicated in the case of extraction by diffusion, as the mass of chips furnished by 3 sets of knives permits of a better circulation of liquid than the mattress furnished by 2 sets of knives and a shredder.

Speed of rotation

The speed of rotation of a knife set varies from 400 to 720 r.p.m. The speed most frequently used is 500 r.p.m. In South Africa³ it varies from 500 to 700, with an average of 600 r.p.m.

The firm of Fletcher considers that the optimum speed lies between 500 and 600 r.p.m., and that higher speeds give scarcely any better results while, on the other hand, one or two broken blades can put the set badly out of balance and cause more serious damage with higher speeds.

In certain countries the speed has been pushed up as far as 1200 r.p.m. It may be considered that this would give very fine preparation, but at such speeds wear on the blades must be very rapid, power consumed must be considerable and accidents would be more serious.

We recommend 500 r.p.m. for the first set and 600 r.p.m. for the second, while agreeing that if, for the sake of uniformity, it is preferred to adopt the same speed for both, there would be little disadvantage in doing so.

Power

General considerations. The average power absorbed by a set of knives depends on:

- (1) The tonnage of cane
- (2) The fibre in the cane
- (3) The nature of the fibre, whether more or less resistant
- (4) The proportion of cane actually cut - that is, on the clearance
- (5) The number of blades
- (6) The speed of rotation
- (7) The radius of the cutting circle
- (8) Diverse variable factors: friction, lubrication, knives more or less worn.

These latter factors cannot be measured accurately and it is scarcely easier to take into account the first three factors, although they are measurable, since their influence is variable and problematical. The number of blades in particular seems paradoxically to have little influence. Nicklin⁴ in Australia cites a set of 70 blades which, in analogous conditions, consumed less power per tonne of cane than another set with 36 blades. Consequently we shall be

content with relating the power to the tonnage, which remains the most important factor. While sometimes expressed in terms of t.c.h., it is only the fibre rate (t.f.h.) that is relevant.

More recently, Australian workers⁵ have found a very marked difference between the power necessary when the canes are delivered in pieces furnished by a chopper-harvester, relative to the power normally required for whole canes. This difference is obviously more marked for the first set of knives than for the second. They give the figures shown in Table 3.1.

TABLE 3.1.

POWER CONSUMPTION OF KNIVES (QUEENSLAND)

	Whole cane			Chopped cane		
	P_m	PM	P_i	P_m	PM	P_i
1st set	0.75 (1)	3.2 (4.2)	1.8 (2.4)	0.33 (0.4)	0.75 (1)	0.45 (0.6)
2nd set	0.82 (1.1)	3.3 (4.4)	1.9 (2.5)	0.45 (0.6)	1.2 (1.6)	0.60 (0.9)

P_m = mean power absorbed, in kW/t.c.h. (in brackets, in h.p./t.c.h.)

P_M = maximum peak power, in kW/t.c.h. (h.p./t.c.h.)

p_i = power to be installed, in kW/t.c.h. (h.p./t.c.h.).

This table is based on the principle that an electric motor can furnish for 15 seconds a power equal to 1.7 times its nominal power.

Farrell, in 1971, gave figures for power as shown in Fig. 3.11 for electric motors to be in-

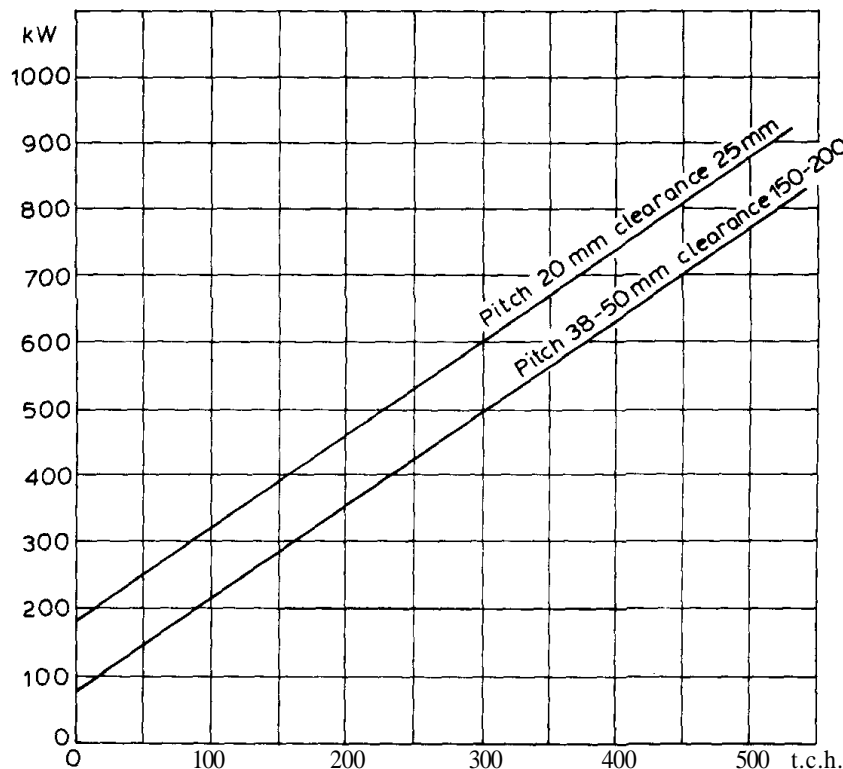


Fig. 3.11. Knife absorbed power (Farrel).

stalled for driving knives at 600 r.p.m. The upper line in the graph is applicable for knives of 20–22 mm ($\frac{3}{4}$ – $\frac{7}{8}$ in.) pitch at a setting of 25 mm (1 in.); the lower line for knives of 38–50 mm ($1\frac{1}{2}$ –2 in.) pitch, set at 150–200 mm (6–8 in.). If the knife set is driven by a steam turbine, Farrell advises providing a turbine of nominal power 33% higher than that indicated for the electric motor. He recommends a rotational speed of 3,600 r.p.m., with 6–1 reduction gear and power 50% above the indicated power requirements.

Farrell's lower curve (for leveller knives) still remains appropriate for the present day; it allows for motors as generally provided. However, technique has developed since 1971 and, for shredding knives, has led to much higher power requirements.

A good example is given by South African practice. Table 3.2 gives the power installed for knives in that country. The figures are in kW/t.f.h. and are derived from the mean of 22 tandems; extreme values are indicated in brackets. Power consumed is of the order of 60% of the mean power installed.

TABLE 3.2

POWER INSTALLED FOR CANE-KNIVES, IN kW/t.f.h.

<i>Method of operation</i>	<i>1st knives</i>	<i>2nd knives</i>	<i>3rd knives</i>
Operating as levellers	7 (3-15)		
Levellers in reverse rotation	10 (7-14)		
Working as shredder	20 (14-30)	20 (10-40)	12 (9-18)
Shredder in reverse rotation	24 (20-30)	40 (30-70)	16 (14- 19)

Influence of clearance. We have sometimes found that factory employees have the impression that by diminishing the clearance by half, passing for example from 10 cm to 5 cm, one would double the necessary power. But, as we have just seen, the clearance is concerned only as it affects the depth by which the knives plunge into the layer of cane. If this depth is 0.65 m (2 ft.), a clearance of 0.15 m (6 in.) ($h - r = 0.5$ m) will require a power only 50% greater than a clearance of 0.30 m (12 in.) ($h - r = 0.35$ m).

Effect of wear. In proportion as the blades become worn, the power consumption increases. It has been found in Australia⁶ for knives which had been sharpened on Sunday, an increase of 11 - 17% in power consumption between Monday and Friday.

Variations in power. It is necessary merely to stand alongside an electrically driven knife set furnished with an ammeter in order to see the continual and fierce variations in the power consumption of such a set.

The mean value which we have given corresponds to the sum of instantaneous values ranging from a little more than half to more than double, according to the density of loading of the portion of the carrier arriving at the knives. A bundle of parallel canes would cause the circuit breaker to function if the crane were to deposit such a bundle directly into the carrier.

Automatic regulation of carrier speed. In order to avoid:

- (a) The disadvantages of an excessive variation in rate of cane arriving at the knives

(b) Irregularity in feeding of the mills

(c) Chokes at the knives and stoppages involved for clearing these

Most factories interpose a coupler of the "Heenan and Froude" type between the carrier and its driving motor (*cf.* p. 27) and actuate the coupler by means of one phase of the motor driving the first set of knives, for example. With this arrangement, the carrier slows down when the knives are overloaded, and this allows more time to treat the heavily loaded pile of cane, and evens out the distribution of the cut cane on the carrier. The overload on the second set of knives is much lower and this in turn is evened out.

When two sets of knives are installed, it will be satisfactory to provide for each of these an installed power equal to approximately $\frac{4}{5}$ of the preceding values.

However, many factories tend to install a second set of knives much more powerful than the first, the power of the motors being for example in the proportion of 2 to 3. The total power in such cases is not always designed to be different from the foregoing, and is maintained equal to double the value given above. Everything depends on the duty demanded of each of these two sets; a very powerful second set can well recover or compensate for the inadequate work of a first set which is old, slow or has only a small number of blades. The opposite is sometimes seen, the major part of the work sought being obtained by the first set.

Drive to adjacent knife sets. The two sets of knives may be connected by a belt, provided of course that the pulley sizes correspond to the respective speeds of the two motors. This arrangement, which is logical, allows the two motors to assist each other when one or the other is overloaded; at the same time it allows the total power provided for the two motors to be reduced by perhaps 25%. It may be employed wherever the distance between the two sets of knives permits. Both sets of knives in fact may be driven from a single motor directly coupled to one set.

Such an installation thus provides great flexibility. The nominal power of the two motors may well be reduced to 7.5 and 12 kW/t.f.h. (10 and 15 h.p./t.f.h.) respectively, with little adverse affect on the results sought. To be able to drive the knives in this way, it is advisable that, even with V-belt drives, the distance between the shafts of the two sets should not be more than 4.25 m (14 ft.) or preferably 3.6 m (12 ft.). It is also recommended that the two motors be of the same type and naturally of the same speed, otherwise the amount of slip would be different. It is also recommended in Australia⁷ that the ratio of the power of the first motor to that of the second should be between 0.8 and 1.0 if the load on the belt is to be as low as possible.

Clearances for two knife sets

The case of two sets of knives is by far the commonest. The first set is then given a high clearance and, for this role of leveller, knives of fairly high pitch are provided: generally 50 mm (2 in.), with a small number of blades; it is driven at 500 r.p.m., sometimes 600.

The second set on the other hand is very close to the carrier, for example, with a clearance of 50 mm (2 in.), more often 25 mm (1 in.) or even as low as 10 mm ($\frac{3}{8}$ in.). It may be driven at 500 r.p.m., more often at 600, sometimes at 750 r.p.m. It has a greater number of blades, the pitch being as low as 25 mm (1 in.) and even 22 mm ($\frac{7}{8}$ in.).

The clearance adopted for the first set should be approximately one-quarter of the height of the layer of cane.

Bulk density of cane prepared by knives

The bulk density of cane after passing through two sets of knives is given by Kerr⁸ as being of the order of 480 kg/m^3 (30 lb./cu.ft.); this value corresponds to an excellent degree of preparation. In Reunion, it is only of the order of 320 kg/m^3 (20 lb./cu.ft.).

Ducasse knives

Among the interesting devices for cane preparation, Ducasse knives are worthy of mention (Fig. 3.12). This equipment consists of a shaft carrying discs or bosses to which knife blades are fixed, each by three bolts; this shaft is enclosed in a casing and rotates at $580 - 1000 \text{ r.p.m.}$ The knife set is located at the top of the cane carrier and arranged so that the plane of the conveyor bringing the cane (or chips of cane cut by a preceding leveller) is tangential to the circle of cut of the blades. The direction of rotation of the assembly is such that the blades pass upwards through the bed of cane on the carrier; they thus throw the cut pieces toward the roof of the casing. They strike an inclined plate which is prolonged to a toothed anvil set very close to the circle of cut, 35 mm ($1\frac{3}{8} \text{ in.}$) at entry, 10 mm ($\frac{3}{8} \text{ in.}$) at delivery. The knives force the pieces through this narrow passage and are furnished with plates which act as hammers and so add a shredding action to their cutting action. The inventor of this equipment has called it a **chopper-fiberiser**. Power installed is approximately 9 kW/t.f.h. (12 h.p./t.f.h.).

Ducasse has also developed a knife set in which the knives are mounted on stout pins, on which they can oscillate freely about a vertical plane; thus they suffer much less damage due to stones or other foreign bodies. This type of knife presents two advantages:

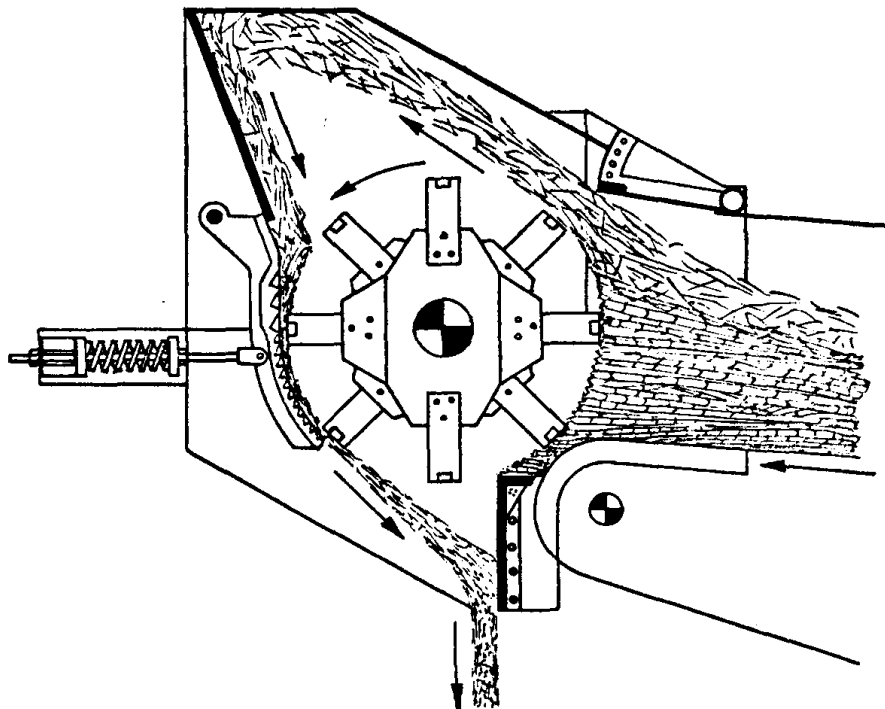


Fig. 3.12. Ducasse knives - operating principle.

- (a) The free oscillation permitted for the blades forms a valuable protection against the effects of damage due to stones
- (b) The size of the pins assures a long life and much less wear than in the standard models.

Mechanical loading and stones

With the mechanisation of cane growing, the traditional hand cutting of cane has been replaced in many countries by mechanical harvesting, and is progressively being replaced in other countries. This development has led to the use of chopper-harvesters which cut the cane into short sections which are transported to the factory by tipping trucks or trucks with other types of mechanical unloading. This method of harvesting presents great risk of deterioration of the chopped cane: the cane should go to the mills as rapidly as possible, preferably in 2 - 6 hours after cutting, otherwise there will be considerable losses of sugar. The chopped cane may contain tops and leaves but these do not include any hard material. Unfortunately, on account of the costs of chopper-harvesters, introduction of mechanisation generally comprises an intermediate phase between manual and chopper-harvesting, which consists of placing the canes in piles on the ground and picking them up by conveyor-elevators or other machines such as rakes or grabs. This method of harvesting presents the grave disadvantage of sending to the factory, among such bundles of cane picked up, stones and sometimes rocks of some 30-40 kg (70 - 90 lb.) in weight (one factory in Hawaii has even thus picked up a pig, somewhat mutilated, which was detected and taken out only seconds before the first set of knives). These rocks, if not detected before passing to the knives, have a disastrous effect on the latter: knives are damaged or broken, sometimes knife holders similarly, knife shafts are bent, and the pieces of knives not removed on reaching the mills may damage the grooving of the rollers.

It is therefore desirable, if possible, to avoid picking up stones with the cane, and the only sure method of doing this is the careful and complete removal of stones in the field; and then those which persist into the factory must be detected in time. One such method is to tip the cane into a pit from which it is removed by an elevator, set at a steep slope, which feeds the carrier. The Cameco organisation has adopted 28.5°; others use 35°, and our recommendation is 30°. Conveyor chains with hooks attached and spaced at about 60 cm (2 ft.) pick up the cane from the pit, take it up the slope, and deliver it into the carrier. The stones, being heavier, remain at the bottom, or roll or slide down the slope. The accumulated stones are removed from the pit once or twice a day according to the quantity.

It is sometimes considered sufficient to use an ordinary lateral conveyor which drops the cane onto a steel plate inclined at 40 or 45°, from which the cane falls onto the carrier. A stone falling from the height of some 2 m (6 ft.) onto this plate produces a characteristic bang: the conveyor and the carrier are then stopped and then the stone removed. For a more positive arrangement, a microphone may be placed below the plate and connected to an amplifier, so that above a certain sound intensity, an alarm is set off.

Without such precautions, factories working at the stage of pusher-loading of cane can suffer a very unfortunate season; we may quote one case of 45 hours' stoppage due to this cause only in the course of one season, with more than 200 blades broken and almost 100 bent.

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4. *Tramp-Iron Separators*

Generally, no strict account is kept of the number or weight of pieces of steel and cast iron which pass through a mill in the course of a season.

The commonest objects are: pieces of knife-blades, sling-hooks, broken pieces of slings, monkey wrenches, horse-shoes, bolts and nuts. Pieces of cast iron or special steel are the most dangerous. Sometimes, the engine-driver at the mills hears the noise made by the piece of iron falling into the feed hopper, or the abnormal noise of the crusher or first mill. The mill is then stopped, the foreign body found and removed.

Whether it is detected in time or not, but especially in the latter case, such incidents are expensive. In particular, when contemplating the destruction of a certain length of roller grooving, one may estimate the loss suffered by calculating the value of 1.5 mm ($\frac{1}{16}$ in.) of thickness or of a cubic centimetre of roller metal. The rollers are often turned down during the slack season to restore their grooving, and in this operation several sixteenths in diameter are lost - this loss being added to that due to normal wear. It is necessary to take deeper cuts when the roller surface has been more severely damaged.

To avoid such damage and losses, efforts are made to eliminate the pieces of "tramp iron". The amount of tramp iron may be reduced by taking precautions in the loading and unloading of the cane, by insisting on strict tidiness at the cane platform, and by supervising

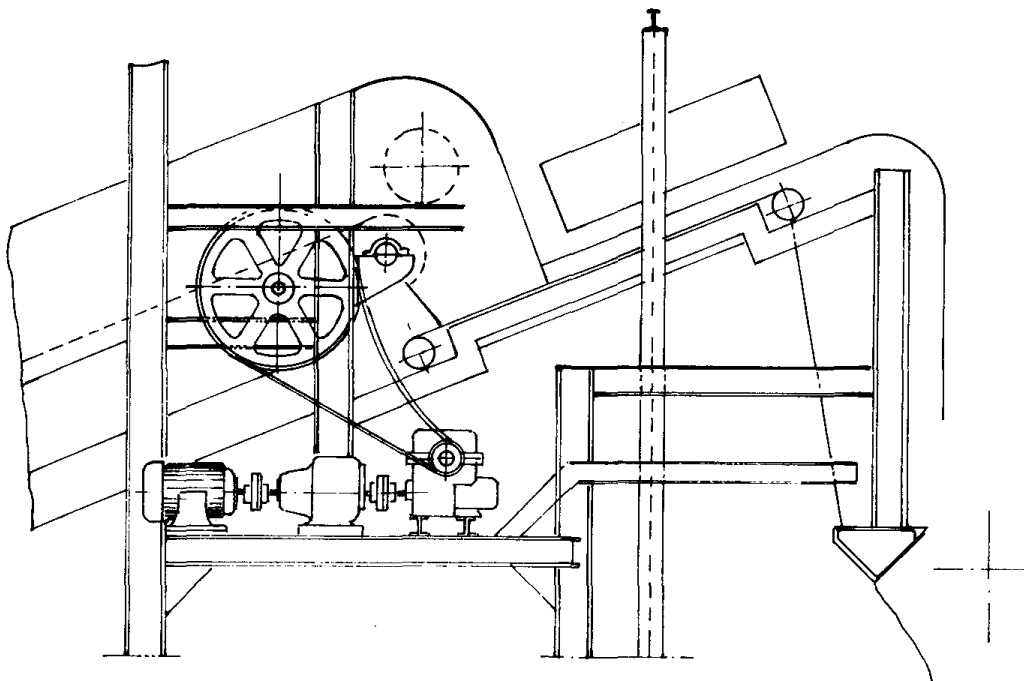


Fig. 4.1. Suspended magnetic separator.

the tightening of bolts at the carrier and the knives. But the best guarantee lies in the installation of a magnetic separator, generally known as a "tramp-iron separator".

Description

Originally, magnetic separators took the form of electromagnets fixed on the bottom of the feed chute to the first mill, or constructed in the form of a rotating cylinder projecting into that chute. These devices had some disadvantages and were not effective.

The device now recommended consists of an electromagnet contained inside a rectangular box suspended above the last cane conveyor before the entry to the first mill, and covering the full width of the conveyor (Fig. 4.1). This is generally a rubber band conveyor and should run at a minimum speed of 1.20 m/s (4 ft./s), preferably 2 m/s (6 ft./s); thus the pieces of cane will form a thin layer less than 140 mm (5½ in.) in thickness, preferably 50–100 mm (2-4 in.). It is advisable to arrange for the conveyor to pass closely over a supporting flat plate, to avoid sagging of the conveyor. The box holding the magnet should be at most 400 mm (16 in.) above the surface of the conveyor, preferably 250—300 mm (10—12 in.).

This device consumes about 7 kW (9 h.p.) and has an efficiency of 80–90%, nearly 100% as far as large pieces of iron are concerned. We have known this type of separator to pick up from the cane carrier a knife holder with its two blades, weighing 70 kg (155 lb.). It may be reckoned that the magnetic separator avoids about 90% of the damage that would otherwise be suffered by the roller surfaces.

Financial aspect

The magnetic separator is expensive, but consumes little power and gives efficient protection to the rollers. It therefore pays for itself rapidly; it is a profitable and useful piece of equipment.

It becomes all the more necessary: (a) with a long train of mills; (b) when a higher hydraulic pressure is employed. Actually, much tramp iron passes undetected through the whole tandem and so damages all the rollers. Further, the damage suffered by the rollers rapidly becomes more serious with increasing pressure.

5. *Crushers*

OBJECT

The crusher is the first machine applying pressure which the cane encounters on arriving at the milling plant. It consists of a mill, generally of two rollers, which performs two main functions:

- (a) It assures the feeding of the whole tandem
- (b) It prepares the cane in such a way as to facilitate the grip of the rollers and the extraction of juice by the mills.

The characteristics of a crusher therefore are the following:

- (1) It possesses a surface especially constructed to permit it to grip the cane or pieces of cane which are fed to it, in the best conditions possible.
- (2) This surface must at the same time be designed in such a way as to break, tear up, and crush the cane, in order to permit the mills to get to work immediately and effectively on the broken-up material, which is already of the nature of bagasse rather than cane.
- (3) It should have a peripheral speed superior to that of the mills which it has to feed, since the material which is fed to it has not yet assumed the form of bagasse, and is still difficult to grip. If it had the same speed as the mills, it would be handicapped in its attempt to supply the quantity of material which they are capable of treating.

2-ROLLER CRUSHERS

Types

Many types of crushers have been designed and tried, but only two have achieved general importance. These were:

- (a) The Krajewski (now of historical interest only; see first edition, pp. 38 – 45)
- (b) The Fulton.

The Fulton type is named after the firm which has predominantly contributed to its design and wide use. It is the only type used at the present day.

The Fulton roller is really a mill roller treated so as to serve in a crusher. It is a surface of revolution, obtained by taking a serrated line, enclosed between two lines parallel to the axis, and rotating this around the axis of the roller. Thus there is obtained circular grooving of V form, separated by ridges or teeth of inverted V shape (Fig. 5.1).

Since a surface of revolution, even with grooving, is scarcely suitable for assuring a grip on the cane, grooves are cut longitudinally in the form of chevrons which form a number of notches in each tooth of the roller. The profile of these teeth is so chosen as to favour the grip on the cane (Fig. 5.2).

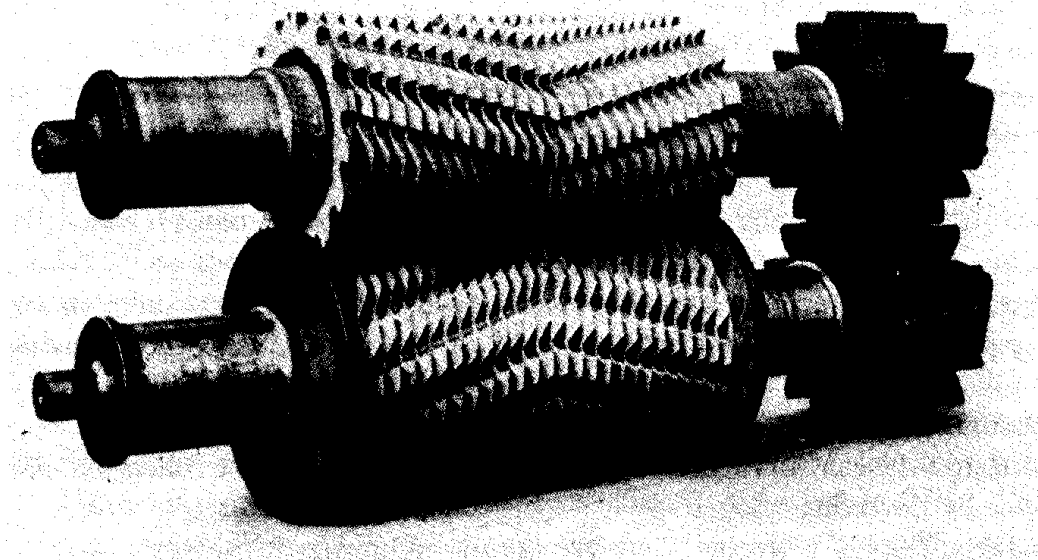


Fig. 5.1. Fulton crusher rollers (Fives Cail - Babcock).

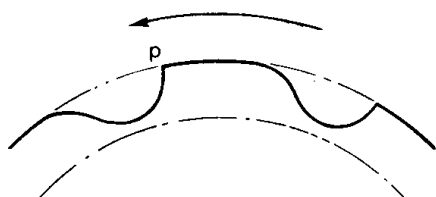


Fig. 5.2. Notches of Fulton roller.

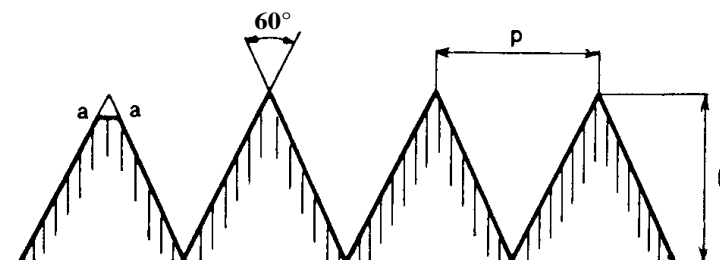


Fig. 5.3. Fulton grooves.

Since the Fulton grooves tend to become blocked with bagasse, it is general practice to fit scrapers to the two rollers of this crusher.

Grooving

The angle of the grooves is about $35^\circ - 55^\circ$ (Fig. 5.3), often 45° with French firms. The pitch, which here means the distance from one tooth to the next, varies from 38 to 100 mm ($1\frac{1}{2}$ to 4 in.). The most frequent values are 50-75 mm (2-3 in.).

When the angle is 50° , the depth is slightly greater than the pitch. In practice, however, the manufacturer does not finish the tooth profile at a point; it would be too fragile. He finishes it with a small flat surface *a a* (Fig. 5.3) the width of which is approximately one-tenth of the height of the tooth, without going lower than 1.5 mm ($\frac{1}{16}$ in.).

What pitch should be chosen? There are two trends: the first consists of proportioning the pitch to the diameter of the roller. The rule could be suggested:

$$p = 0.075 D \quad (5.1)$$

D = mean diameter of the rollers

p = pitch of the grooves, in the same unit.

However, it is not practical to have a different pitch for every roller diameter, and the second trend is to proportion the teeth, not to the dimensions of the mill, but to those of the cane, which are independent of the equipment. In this case, for the usual roller dimensions of 660 X 1,220 mm to 1,065 X 2,134 mm (26 X 48 in. to 42 x 84 in.), grooving of 39 X 40 mm could be recommended, *i.e.* 39 mm in depth and 40 mm in pitch, or 44 x 45, 49 x 50, 60 x 60, or 80 x 81 (50° angle).

Chevrons. The helicoidal "chevron" grooves should have a maximum depth slightly less than that of the teeth (Fig. 5.2). If these grooves were to be taken to the bottom of the circumferential grooving there would be some risk of the teeth of the scraper plates catching in them and causing accidents.

However, we favour much shallower chevrons. The chevron in effect removes the tooth for part of its length, sometimes almost half of it, thus reducing the pressure at the position of the chevron and consequently reducing the efficiency of the crusher. The quantity of metal lost is proportional to the square of the depth of the chevron; and it is mainly the point of attack of the chevron (p in Fig. 5.2) which does the work and seizes the cane. A chevron 13 mm ($\frac{1}{2}$ in.) deep operates nearly as well as one of 25 mm (1 in.), particularly when it is new and not worn. For this reason we recommend using chevrons the depth of which is no more than half the height of the tooth. There will then be little loss of capacity, and extraction will be much improved.

The pitch of the chevrons (distance from the end of one chevron to the end of the next, measured circumferentially) is of the order of 20 cm (8 in.). Manufacturers often adopt the rule of placing the point of one chevron on the longitudinal line joining the extremities of the V formed by the preceding one. In these conditions, the pitch is proportional to the length of the roller, for a given angle β of the chevrons with the longitudinal lines (Fig. 5.4). This angle β is generally 18°. It varies according to the manufacturer from 16 to 25°.

The chevrons of the upper and lower rollers are generally arranged so that they cross in the angle of the axial plane of the two cylinders, on the line of maximum pressure. An observer located at the top of the cane carrier would see the chevrons of both rollers, upper and lower, both orientated with their points upward (Fig. 5.5). In this case, only the chevrons of the upper roller tend to push the cane towards the centre of the roller. Hence there is some

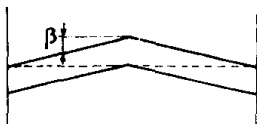


Fig. 5.4. Arrangement of chevrons.

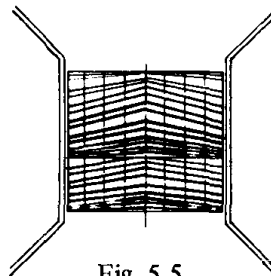


Fig. 5.5

Fig. 5.5. Chevrons point upwards (view from feed side).

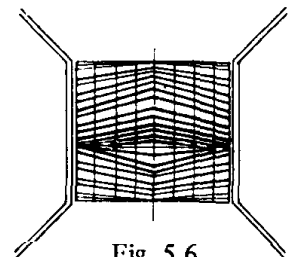


Fig. 5.6

Fig. 5.6. Chevrons in "ace of diamonds" (view from feed side).

risk of pressure on the roller flanges. Also, certain manufacturers provide the lower roller with chevrons arranged in the opposite sense; the same observer would see the chevrons of the two rollers "in ace of diamonds", *i.e.* forming lozenge-shapes (Fig. 5.6). In this case, it is suggested that the rollers should mesh so that the chevrons of the one come in between those of the other. This arrangement, however, presents no marked advantage compared with the preceding one.

Inclination of the housing

The original two-roll crushers were constructed with the axes of the two rollers in the same vertical plane. It was then realised that the feeding was better and the grip better when the supports were inclined. At the present time, crushers are built with the axial plane inclined at $60 - 75^\circ$ to the horizontal (Fig. 5.7).

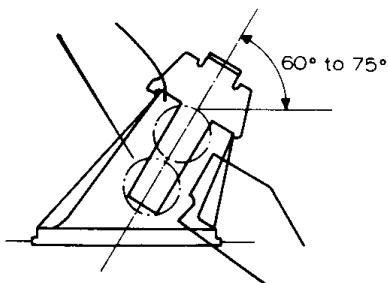
The angle of 60° is still considerably greater than the axial plane of the top and feed rollers of a mill, the inclination of which varies from 45 to 56° . The flow of juice on the feed side of the bottom roller of a crusher is therefore not more difficult than in a mill. Since, moreover, the inclination of the crusher facilitates feeding and the crusher is above all a device for improving capacity and for cane preparation much more than for extracting juice, we prefer the inclination of 60° to one of 75° or an intermediate value.

Diameter

The "diameter" of a roller of a Fulton crusher, or of a mill, is the mean diameter of the roller, measured at the mid-point of the teeth. This diameter D is the arithmetic mean between the diameter D_M taken at the top of the teeth and the diameter D_m taken at the bottom of the grooves (Fig. 5.8):

$$D = \frac{D_M + D_m}{2}$$

Certain American firms give their crushers a diameter greater than that of their mills: for example, crusher $940 \times 2,135$ mm (37×84 in.), ahead of mills of $915 \times 2,135$ mm (36×84 in.). This arrangement has the object of increasing the capacity and facilitating the feeding of the crushers, in such a way as to increase the tonnage handled. But this object should be achieved by adjusting the speed of the crusher, without breaking the uniformity of the tandem.



Dif. 5.7. Inclination of crusher frame.

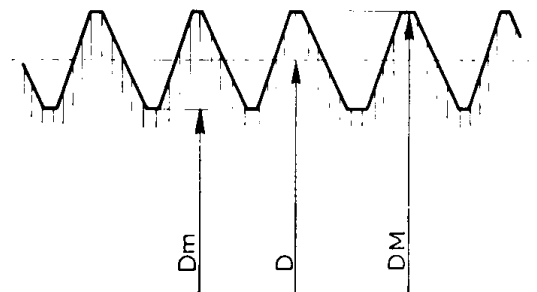


Fig. 5.8. Mean diameter.

Speed

The crusher, having to accept a material less compact and more slippery than that which it feeds to the mills, is handicapped compared with the latter as regards feeding. To overcome this handicap, it is given a surface speed higher than that of the mills. The tonnage crushed being nearly proportional to the speed, for the same setting between top and bottom rollers, the difference between the speeds chosen should represent the handicap involved.

The peripheral speed generally adopted for the crusher is 25 - 50% greater than that of the mills, most often 30 - 40%. For a well-designed and well-regulated crusher, a difference in speed of 30% should be sufficient.

Pressure

We shall examine later (Chap. 10) the subject of hydraulic pressure. It will be sufficient to state here that, in **general**, the total hydraulic pressure applied to ordinary two-roller crushers is about 40 - 80% of the pressure applied to mills (assuming the same roller dimensions), in most cases 50 - 75%.

We must not overlook the fact that the crusher is primarily a device for feeding and preparation, and not **primarily** one for extraction. If the mill is working comfortably, and if the crusher easily takes the desired tonnage, there is no disadvantage in loading the crusher; the cane preparation and the extraction cannot but gain thereby. If, on the contrary, it is necessary to push the mill to the maximum of its capacity, and if the crusher does not feed well when heavily loaded, there is much to be gained by reducing its hydraulic loading to 50% of that of the mills.

The procedure to adopt also depends greatly on the pressure adopted for the mills. If the mean specific pressure at the mills (*cf.* p. 128) is high, that of the crusher may more readily be reduced.

The specific hydraulic pressure (**s.h.p.**, *cf.* p. 128) of a crusher should be at least 5 tonnes/dm² (45 tons/sq.ft.). A good value would lie between 8 and 12 tonnes/dm² (75 and 100 tons/sq.ft.). There is generally no advantage in exceeding 15 tonnes/dm² (140 tons/sq.ft.): the crusher does not lift; it does not feed so well, and the preparation is not appreciably better.

Lift

The crusher, receiving a material which is not yet crushed, and transforming it to a broken material, is the pressure device which, for this reason, has to do most work of compression, as regards extent of crushing. Further, being at the head of the tandem, it is not protected by a preceding crushing device and is the first to receive foreign bodies, stones, lumps of earth, and various objects, some of them vary hard and resistant. For these reasons, crushers are designed with a greater margin of lift than are mills, of the order of 38 - 63 mm (1½ - 2½ in.) in general, shall we say with greater precision of the order of 6 - 7% of the roller diameter. **Fulton**¹ has provided the crusher at Belle Glade with a maximum lift of 3¾ in. (95 mm), or 9.6% of the roller diameter.

Power

We shall consider the power consumption of a crusher and the power to be provided for its drive when discussing the power requirements of mills (*cf.* Chap. 14).

Here and now, we may indicate that the power consumed by a crusher is about 75% of that of a mill of the same roller dimensions and the same speed, and utilising the same total hydraulic pressure. If the hydraulic pressure of the crusher is 70% of that of the mills which follow it, and if its speed is 30% greater, the power consumed by the crusher will be (*cf.* p. 234):

$$0.75 \times 0.70 \times 1.3 = f \quad (5.2)$$

of the mean power absorbed by each of the following mills.

This proportion is easily remembered with the aid of the mnemonic: the power of crushers is to that of mills in approximately the same proportion as the respective number of rollers.

Also, when comparing the power requirements of two tandems of mills, it is logical to take into consideration the number of rollers in each of the two tandems. A tandem of 17 rollers should take $\frac{17}{15}$ of the power necessary for a tandem of 15 rollers of the same characteristics.

VALUE OF 2-ROLLER CRUSHER

At one time, it was not considered that a tandem could be installed without a 2-roller crusher. The few tandems of 3-roller mills had all their mills similar, both in roller surface and roller speed; they achieved a good extraction, but a crushing rate inferior to that of tandems with a crusher. The situation changed completely when, after discarding the crusher, someone had the idea of using the first mill in such a way as to replace it. We shall see that this idea succeeded very well and since then the 2-roller crusher has become an unprofitable and cumbersome machine. It could not always be made to function properly; it lifted badly or not at all, it sometimes used to feed badly, it lengthened unduly the cane carrier and the space taken up by the tandem. We are strongly in favour of eliminating it and installing only 3-roller units.

3-ROLLER CRUSHER

The name "3-roller crusher" or "mill-crusher" is used for a mill similar to others of the tandem but placed at the head of the train, forming the first unit of application of pressure and extraction of juice, and designed to perform the duty of a crusher. The differences relative to the following mills are then:

- (1) A speed greater than that of the mills situated immediately after it
- (2) The roller surfaces are provided with deep grooving and with numerous gripping chevrons.

Speed

The speed of the mill-crusher is made appreciably higher than that of the first mill. For a factory following the Java school (speeds decreasing from the first to the last mill) it will be

the highest-speed mill of the tandem. For a factory following the Hawaiian school (speeds increasing), its speed will be of the order of that of one of the later mills of the tandem. The following are two examples of the scale of speeds:

(1) Central Igualdad, in Puerto Rico. Tandem of 15 rollers (mill-crusher of 863 x 1,620 mm (34 x 60 in.) and other mills of 810 x 1,620 mm (32 x 60 in.), preceded by one set of knives:

TABLE 5.1

CENTRAL IOUALDAD, SEQUENCE OF SPEEDS

	<i>V</i>		<i>n</i> (<i>r.p.m.</i>)	% <i>last-mill</i> <i>speed</i>
	(<i>m/min</i>)	(<i>ft./min</i>)		
Mill-crusher	12.30	40.4	4.53	95
1st mill	11.10	36.4	4.35	86
2nd mill	11.70	38.4	4.58	91
3rd mill	12.30	40.4	4.82	95
4th mill	12.90	42.3	5.05	100

V = peripheral speed of rollers; *n* = speed of rotation.

(2) Caymanas, in Jamaica. 15-roller tandem with three-roller crusher 635 x 1,220 mm (25 x 48 in.) and four mills 610 x 1,220 mm (24 x 48 in.), preceded by a set of knives:

TABLE 5.2

CAYMANAS, SEQUENCE OF SPEEDS

	<i>V</i>		<i>n</i> (<i>r.p.m.</i>)	% <i>last-mill</i> <i>speed</i>
	(<i>m/min</i>)	(<i>ft./min</i>)		
Mill-crusher	9.18	30	4.61	82
1st mill	7.82	26	4.09	70
2nd mill	8.80	28.9	4.60	79
3rd mill	10.00	32.9	5.23	90
4th mill	11.15	36.6	5.83	100

With the Hawaiian practice, followed almost universally today, we support the idea of giving the mill-crusher a peripheral speed equal to that of the last mill of the tandem.

Hydraulic pressure

If the mill-crusher takes the feed well, it is advantageous to apply to it the same hydraulic pressure as on the heaviest-loaded mills of the tandem. If it feeds badly, the pressure may without serious disadvantage be reduced to 75% of the mean pressure used on the following mills; but care should be taken to increase it again when feeding returns to normal.

Power

The mean power consumed by a mill-crusher is determined exactly as for an ordinary mill

(*cf.* Chap. 14). The coefficient of the term allowing for the work of compressing the cane, however, is increased by 20 - 25% by reason of the substantial extra work required for breaking up a harder and more resistant material (fragments of rinds and nodes not yet disintegrated).

Use of 3-roller-crusher

A tandem without a crusher would be incomplete. Such would be, in particular, the case of a tandem of 12 or 15 rollers, for example, in which the first unit was provided with rollers running at the same speed and having the same grooving as those of the following unit.

On the other hand, if this first unit really forms a crusher, by the nature of its roller surface, and its speed, we prefer it to have three rather than two rollers. Perhaps the capacity would not show any great gain, but the following advantages may be confidently expected:

- (1) Extraction definitely superior.
- (2) Uniformity of all units of the tandem. For an electrically driven tandem, for example, there would be no disadvantage in having all the motors identical.
- (3) Head room required is much less. A tandem with ordinary crusher feeding the first mill by gravity, as is usual, requires 2 - 3 m (7-9 ft.) extra height compared with a tandem all units of which are of three rollers and placed at the same level.
- (4) Length and space taken up by the carrier are much less. For an ordinary crusher, the carrier must ascend, at its slope of 30 - 60%, this extra height of 2 - 3 m. This means some 6- 10 m (20 - 30 ft.) extra length to be provided for the sloping portion of the carrier.

Moreover, the cost of a tandem of 12 rollers, including accessories, is not appreciably greater than that of an 11-roller tandem; similarly, for 15 rollers compared with 14, and so on.

Table 5.3 gives the results reported in Java for 1930, reduced to a standard roller size of 1,520 x 760 mm (60 X 30 in.).

TABLE 5.3

CAPACITY AND EXTRACTION OF MILLING TANDEMIS IN JAVA

<i>Number of factories</i>	<i>Tandem</i>	<i>Number of rollers</i>	<i>Capacity (t.c.h.)</i>	<i>Extn.</i>	<i>Lost juice % fibre</i>
3	Crusher + 3 mills	11	31	93.5	48
26	4 mills	12	32.2	93.9	44
93	Crusher + 4 mills	14	41.7	94.6	39
15	5 mills	15	34	95.4	34

The rather low relative capacity which will be noted for the 15-roller tandems arises from the fact that in Java the mill-crushers in 1930 had not the speed nor the grooving which they would have required to be real crushers. If it were possible to compare 14-roller tandems with similar tandems of the same design, but of 15 rollers, and provided with true mill-crushers, it would be seen that the difference in capacity would be reversed in favour of the 15-roller trains.

EXTRACTION OF A CRUSHER

The extraction obtained in a crusher is very variable: the crusher is a device for feeding and preparation, and not for extraction. However, it is advantageous, for the total extraction of the tandem, that it should extract as much as possible, since the less juice it leaves in the cane entering the first mill, the less will remain in the bagasse leaving the last mill. The mills do recover a very large proportion of the juice which the crusher leaves, but never all of it.

Table 5.4 gives the proportion of juice extracted by the various types of crusher, as a percentage of sugar in cane:

TABLE 5.4

EXTRACTION WITH DIFFERENT TYPES OF CRUSHER

<i>Crusher</i>	<i>Extraction % absolute juice</i>
Krajewski (2 rollers)	40 - 50
Fulton (2 rollers)	45 - 55
Double crusher (4 rollers)	60 - 70
Mill-crusher (3 rollers)	60-75
Fulton crusher followed by mill (5 rollers)	70 - 83

This table demonstrates the value of the mill-crusher. The figures are given with a large margin, since they vary with diverse factors, and particularly with fibre in cane.

LARGE 3-ROLLER CRUSHER

In Australia, it is not uncommon to find tandems of 3-roller units, in which the first mill is larger than the following ones, not only in diameter, but also in length of the rollers; for example, a tandem of 5 mills, the first being 915 x 1,980 mm (36 x 78 in.) and the four others each 846 x 1,830 mm (34 x 72 in.). This arrangement has the disadvantage of breaking the uniformity of the tandem, but this disadvantage is not very serious for the first mill which normally should have different grooving and different speed. On the contrary, it permits of maintaining peripheral rotational speeds of the same order throughout the length of the tandem. We approve of this arrangement, which permits mills after the first to be used to the maximum advantage.

Certain factories provide a large unit not only for the first mill, but also for the last mill. This arrangement is also partly justified, but less so than the preceding one.

REFERENCES

- 1 *Sugar Azucar*, 59 (5) (1964) 68.

6. *Shredders*

OBJECT

The object of the shredder is to complete the preparation and disintegration of the cane, so as to facilitate the extraction of juice by the mills.

LOCATION

The shredder is placed at the head of the tandem, after the knives and before the first mill. At an earlier period, when crushers were more common, the shredder was sometimes placed between the crusher and the first mill. This allowed it to work on cane already deprived of part of its juice, and with its fibres partly disintegrated. It thus consumed much less power and functioned more smoothly and sweetly; but it does not then fit so readily into the equipment: it breaks the continuity and symmetry of the milling train, and interrupts the supervision of it.

There was also a shredder incorporated with the crusher or the first mill. This was the Maxwell crusher-shredder, which was described and studied in the second edition (pp. 62 – 65). It is no longer of anything but historical interest.

The types most used at present (1981) are the Searby and Gruendler shredders. These render excellent service, and we shall discuss their main characteristics; however, there is a definite trend now to replace them by models devised on the same principle but more powerful, which will also be considered in detail.

FUNCTION OF THE SHREDDER

The shredder owes its existence and its value to the fact that the tissue of cane cells is very resistant: simple crushing between rollers, even under very high pressure, is not sufficient to break all the cells and extract the juice. On the other hand, if these cells can be torn open and disintegrated, the juice is liberated, becoming more accessible and more easily extracted. To obtain such an effect it is necessary to rupture the tissues: this is achieved by forcing the pieces of cane to pass through a very narrow space, thus blocking them on one side and striking them on the other side with a powerful blow; the hammer mill has been chosen with this object.

The shredder thus consists of a rotor working at 500, 1,000 or 1,500 r.p.m., generally 1,000 – 1,200 r.p.m., carrying hammers which are pivoted on discs or plates; the end of the hammer passes very close to an anvil plate which is formed either with a saw-toothed profile

or of rectangular bars. The space or "clearance" between the anvil-bars and the path of the rotating hammers is of the order of a few millimetres. It may vary for example from 25 mm (1 in.) at the entry to the zone of operation to 10 mm (f in.) or even less at the delivery from the anvil-bars.

Since the cane is composed of long fibres and pulp or pith carrying the juice, the material leaving the shredder is a mass of cell material or pith mixed with long thread-like fibres which felt the material together and contribute to hold it together when it is subjected to the pressure of the mill rollers.

The shredder thus fills a role which neither the knives nor the mills can fulfill. It establishes a state of subdivision of the cane which greatly facilitates the work of the mills.

Extent of disintegration

The state of disintegration of the cane is expressed by the Displacability Index (**D.I.**), that is, the percentage of pol in the open cells, or Index of Preparation¹ (**I.P.**), as calculated from laboratory determination of pol obtainable by extraction in cold water, relative to the total pol obtained by hot extraction, or by complete disintegration of the cane in the cold. In South Africa this proportion is expressed in brix instead of by pol: the values are thus fairly close, the free pol being slightly higher than the free brix. We consider however that the determination of the **I.P.** should be made in pol rather than in brix, since it is only the pol extraction in which we are interested (we would be very happy if our mills had a good extraction in pol and a low extraction in brix) and because the extraction of successive mills in brix is, happily, appreciably lower than their extraction of sucrose. After the knives we may have for example 65 or 70% **I.P.**, that is, of open cells, after a conventional shredder, 78–85%, and after a heavy-duty shredder, 86-92%; exceptionally we may attain 94-95% (*cf. p. 66*). In Australia, Bingera mill has found a very close correlation between the extraction of the first pressure-fed mill and the percentage of open cells as expressed by the **I.P.**: $e = 1.02 \text{ I.P.}$

The Index of Preparation is thus defined:

$$\text{I.P.} = \frac{\text{pol extracted by washing (cold)}}{\text{pol extracted after disintegration}} \times 100$$

Extraction by washing is determined on samples of 500 g in 3.5 l of water, with agitation for 20 min. The mean of 6 determinations is taken.

Extraction by disintegration is determined on similar samples (500 g in 3.5 l) disintegrated for 10 min in a "Varigrator" or other machine at a minimum of 7,000 r.p.m. The mean of 3 determinations is taken.

Hammers

The hammers are straight rectangular bars in silicon - manganese steel pivoted on round bars (the arrangement generally known as a "swing-hammer pulveriser" in other industries).

The Searby shredder may have a large number of hammers (for example 144) each weighing 4 kg (9 lb.). The Gruendler uses heavier hammers but fewer in number: for example 66 hammers measuring 325 X 110 x 40 mm ($12\frac{3}{4}$ x $4\frac{1}{4}$ X $1\frac{1}{2}$ in.) and weighing 10 kg (22.5 lb.) each. The hammers of high-power shredders, of the Tongaat type, weigh 15–18 kg

(35 – 40 lb.) and are arranged in 8 rows of 23 – 26, giving a total of 184-210 hammers. In South Africa heavy shredders are also built with 8 rows of 21, giving a total of 168 hammers of 19 or 20 kg (45 lb.) each.

These hammers are tipped on the working face with special steel applied by welding, which lasts for a full season. Thus, in case of wear there is only one facing to be replaced, instead of a full hammer in special steel weighing several kilograms. It has been found, however, that these facings are sometimes damaged or worn by shocks. Then it is a matter of re-welding to renew the facing. Both faces of the hammer are re-welded so that it can be reversed. At Tongaat for example the shredder has 8 rows of hammers; each week 2 rows are reversed and 2 other rows are withdrawn for re-welding and replaced by a row which has been welded. Thus the hammers work on each face for 2 weeks before being re-welded.

The re-welding is made by means of electrodes in special steel. The composition of two such steels is as follows:

	(a) <i>Super-Safor 60</i>	(b) <i>Safdur 800 E</i>
C	3.6 %	5.0%
Cr	32.0	25.0
Si	1.3	0.9
Mn	0.86	0.9
Mo	–	5.0
Nb	–	5.5

Super-Safor 60 is the same alloy as used for re-facing trashplates (*cf.* p. 226). It has a Rockwell hardness of 63, a Brinell of 780 and requires 60 V for welding. The Safdur is even more resistant to shocks.

Index of preparation and power usage

Experience has shown that the efficacy of preparation depends essentially on the power expended to obtain it. Research has been carried out in South Africa to ascertain the correlation between the I.P. and the power expended. Results have been expressed in the formula:

$$\text{I.P.} = 63.7 W^{0.09} \quad (\text{I.P.} = 85.5 W^{0.09})$$

W = power expended for preparation, in kW/t.f.h. (h.p./t.f.h.). The coefficient of correlation was found to be 0.66. This corresponds to the following values:

W (kW/t.f.h.)	I.P.
10	78.4
20	83.4
30	86.5
40	88.8
50	90.6
60	92.1
70	93.4
80	94.5

A good I.P. can thus be obtained with 50 – 80 kW/t.f.h. (70 – 110 h.p./t.f.h.).

SEARBY-TYPE PULVERISER

This is a swing-hammer pulveriser operating in a steel housing: the cane enters by a hopper at the top, and leaves in a broken-up condition between anvil-bars placed at the lower part of the housing at a small clearance from the circle described by the tips of the hammers (Fig. 6.1).

There are two principal makes of this type of shredder, which are very similar, and which differ only in the hammers:

(a) The Searby shredder, installed mainly in Hawaii and in British territories (Fig. 6.1), which has numerous hammers consisting of small rectangular bars

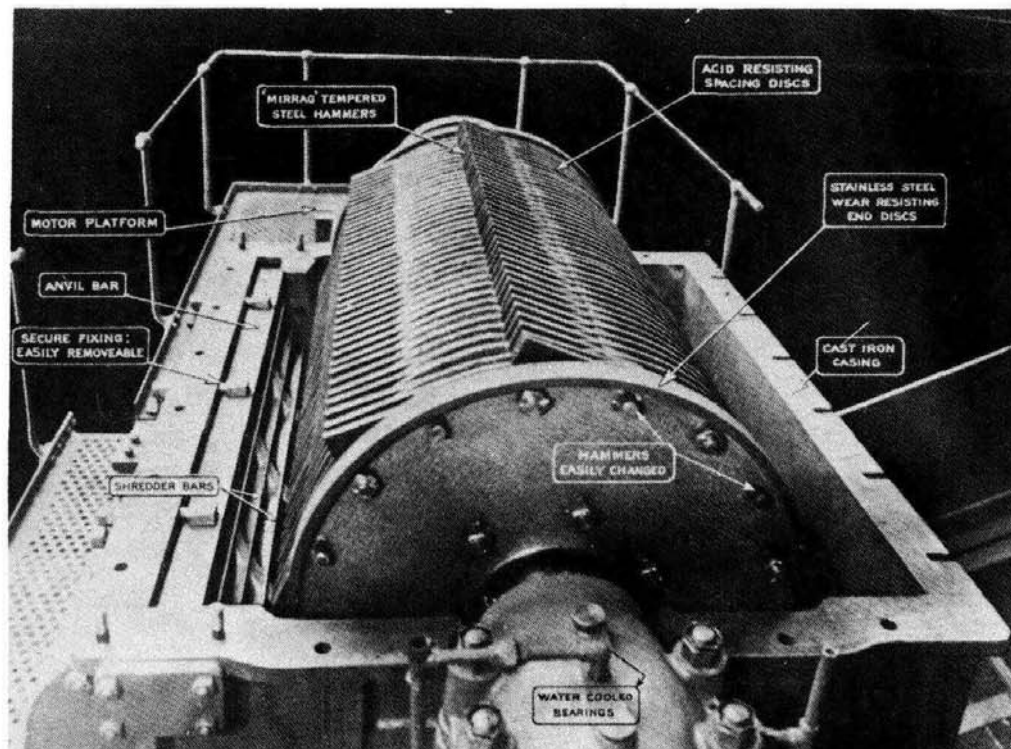


Fig. 6.1. Searby shredder (top casing removed (Smith – Mirrlees))

(b) The Gruendler shredder, installed mostly in Louisiana, Florida and Puerto Rico, which has a smaller number of heavier hammers (Fig. 6.2).

French shredders (Fig. 6.3) follow the Gruendler type.

Fletcher, in England, supplies shredders in 6 sizes:

1,520 x 3,215 mm (60 x 126.5 in.)

1,680 x 3,375 mm (66 x 132.8 in.)

1,830 x 3,525 mm (72 x 138.7 in.)

1,980 x 3,675 mm (78 x 144.7 in.)

2,290 x 3,825 mm (84 x 150.6 in.)

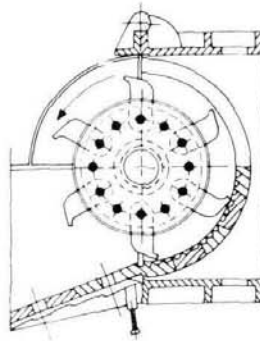


Fig. 6.2. Gruendler shredder.

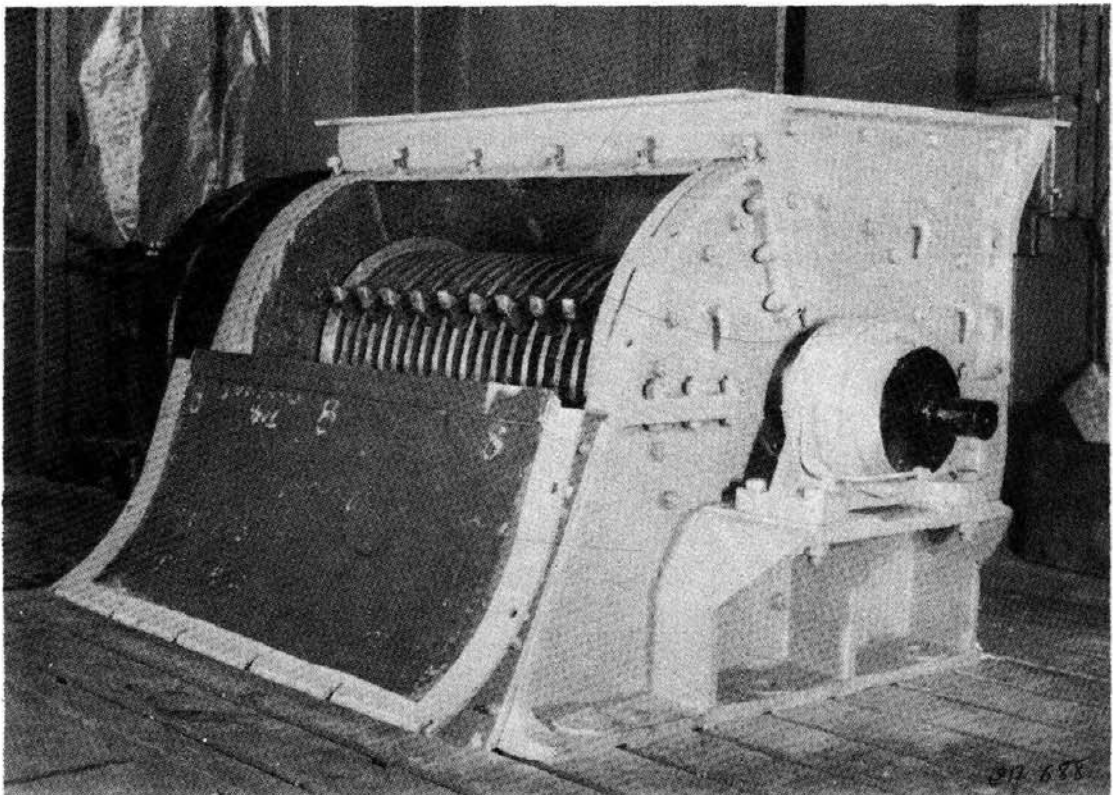


Fig. 6.3. Fives Cail-Babcock shredder, with cover open.

Gruendler offer numerous models for which they indicate the capacities:

1,220 mm (48 in.):	90 – 120 t.c.h.
1,370 mm (54 in.):	100 – 125 t.c.h.
1,525 mm (60 in.):	135 – 165 t.c.h.
1,830 mm (72 in.):	145 – 200 t.c.h.
1,980 mm (78 in.):	195 – 210 t.c.h.
2,130 mm (84 in.):	270–325 t.c.h.
2,440 mm (96 in.):	325 – 370 t.c.h.

They recommend installed power as about 14 kW/t.f.h. (19 h.p./t.f.h.) and estimate power consumed as about half that figure.

Power

In 1970 it was estimated that the shredders then used absorbed a power of 10–12 kW/t.f.h. (13–16 h.p./t.f.h.), and the motors or the drive were provided with a nominal power of around 15 kW/t.f.h. (20 h.p./t.f.h.).

As for knives (*cf.* p. 42), it was stated that the number of hammers had only a small influence on the power consumed by a shredder. Nicklin² found the same power per tonne cane per hour for two shredders one of which had 51, the other, 144 hammers.

However, Crawford³ has made a study, certainly entirely theoretical, in which he gave the power of the shredder as a function of the number and weight of the blades. He arrived at:

$$P = 91 WRhn^3N(\psi + 2\delta) \theta \quad (P = 5.16 WRhn^3N(\psi + 2\delta) \theta) \quad (6.1)$$

P = power absorbed by the shredder, in kW (h.p.)

W = weight of a hammer, in kg (lb.)

R = distance from shaft centre to pivot of the knives, in m (ft.)

N = number of knives

n = speed of rotation, in thousands of r.p.m.

ψ = angle swept by the knives between the first contact with the cane and entry to the anvil, in radians

d = angle swept by the blades in passing over the anvil, in radians

θ = angle by which the knife is moved backwards in its passage over the anvil, in radians.

Crawford estimates 1.5° (0.020–0.025 radian) for θ . ψ depends on the design of the feed to the shredder and varies according to the design from 1 radian to $\pi/2$. d is of the order of $\pi/2$.

This formula gives a power in proportion to θ , an angle which is not known and which can be measured only with a stroboscope, using expensive equipment. Furthermore, the formula takes no account of the weight of fibre treated, which certainly plays a part. Also, and contrary to Nicklin's comment, we state the power as a function of the work required:

Searby-type shredders with light hammers (4–6 kg) (9–14 lb.):

10–12 kW/t.f.h. (13–15 h.p./t.f.h.)

Gruendler-type shredders with medium hammers (7 – 10 kg) (15-22.5 lb.):

12 – 15 kW/t.f.h. (15-20 h.p./t.f.h.)

Tongaat-type shredders with heavy hammers (15 – 20 kg) (35 – 45 lb.):

40 – 50 kW/t.f.h. (55 – 65 h.p./t.f.h.)

The latter types are heavy-duty equipment.

The combination of disintegrator and fiberiser is a very powerful and effective item of preparation equipment, driven by turbines or motors totalling 130 kW/t.f.h. (175 h.p./t.f.h.), and credited with an I.P. of 94-95; this will be discussed in the chapter on diffusion.

Walkers, in Australia, report that the power absorbed is much less when the feed enters vertically than when the knifed cane is fed by a chute inclined at 60°.

Tongaat shredder

The Tongaat shredder (Fig. 6.8) has marked a date in the history of cane preparation, not so much by the originality of its design as by the power which is applied to it. It marks a stage in the amount of power agreed on to attain a high degree of preparation and a very high Index of Preparation. Subsequent installations were inspired by the example of the first, and this has contributed in great part to the high level of energy expended per tonne of cane in recent sugar factories (*cf.* p. 1019). Driven by a turbine of 1,500 kW (2,000 h.p.) and running at 1,200r.p.m., with 188 hammers of 17 kg (38 lb.), the shredder has given an I.D. (Index of Displacement, analogous to the Index of Preparation) of more than 91; this compares to the I.D. of 76 – 80 which was furnished previously by Gruendler and Searby shredders at Tongaat, driven by a motor of 355 kW (475 h.p.). The installed power is thus 45 kW/t.f.h. (60 h.p./t.f.h.).

Consequent on their slight swing, the hammers cover a thickness greater than that of the bagasse layer. The anvil-plate can move aside by 150 mm (6 in.) to allow occasional pieces

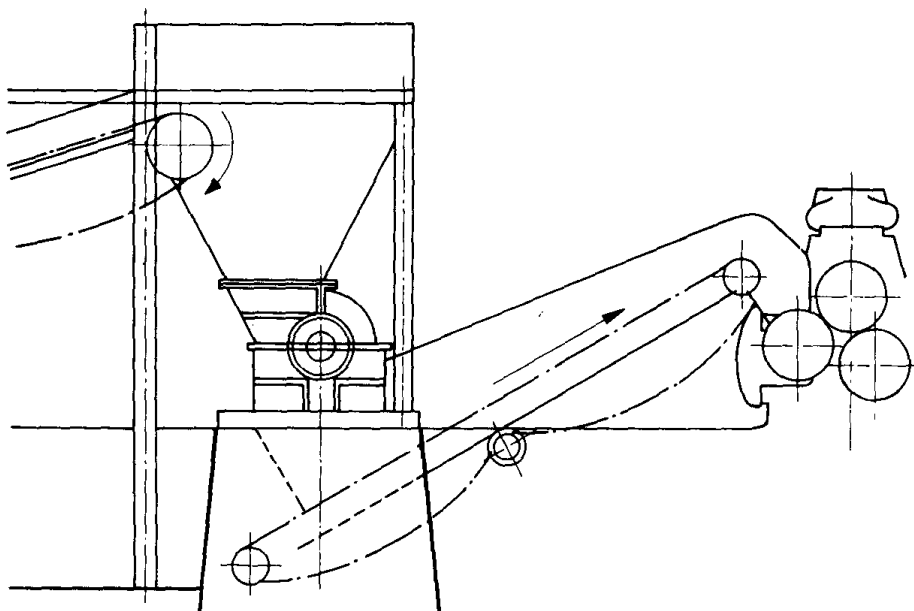


Fig. 6.4. Shredder at head of tandem.

of metal to pass through; the hammers on their part retract by 150 mm (6 in.), so that these pieces can find a passage of 300 mm (12 in.). The moment of inertia is 2,060 m²/kg (4,890 sq. ft./lb.).

Wear is slight: after one million tonnes of cane, the anvil bars lose only 0.1-0.4 mm ($\frac{250}{60}$ in.) on the side where they face the hammers.

Wear and lasting qualities

The main wearing parts are the hammers. For the Searby, the life of the hammers is reckoned as about 90,000 tonnes of cane. For the Gruendler, the hammers, according to the makers, would last for 400,000 tonnes of cane, provided they are overhauled and re-metalled every 40,000 tonnes; we have been able to verify from experience that the former figure is close to actual performance, while the second has been considerably exceeded and could attain 200,000 tonnes.

Saving in power and extraction

The power consumed by the shredder is recovered in the mill, which is relieved of some of its load.

On the other hand, the preparation is very complete: the cane is reduced to a veritable lint and the gain obtained in extraction is very definite. In Hawaii, it is considered that the addition of a Searby improves the extraction of a tandem by

$$\epsilon = \frac{7.5}{N - 8} \% \quad (6.2)$$

ϵ = improvement in extraction due to the shredder

N = number of rollers in the tandem.

e.g. $\epsilon = 2.5\%$ for an 11-roller tandem (e.g. 92% increases to 94.5%)

1.25% for a 14-roller tandem

1 % for a 15-roller tandem.

It is seen that the shredder is more beneficial with a shorter tandem, which is readily understood, since it presents the cane to the mills in a state which permits the mills to extract the maximum of juice. In a long tandem, where the action of the successive mills of itself produces a very complete disintegration, its relative effect is much less.

At the same time it must be kept in mind that, since the extraction improves as the number of rollers increases, the addition of a shredder can only work on a decreasing difference relative to 100%; when the extraction is 94, there is a remainder of 6% on which improvements can be effected, but when the extraction is 96, there is only a remainder of 4%, and improvement on this is much more difficult.

Moor⁴ proposes the application of this formula to the case of gain in extraction by replacing a normal shredder by one of high power, and gives:

$$\epsilon = \frac{10.7}{N - 8}$$

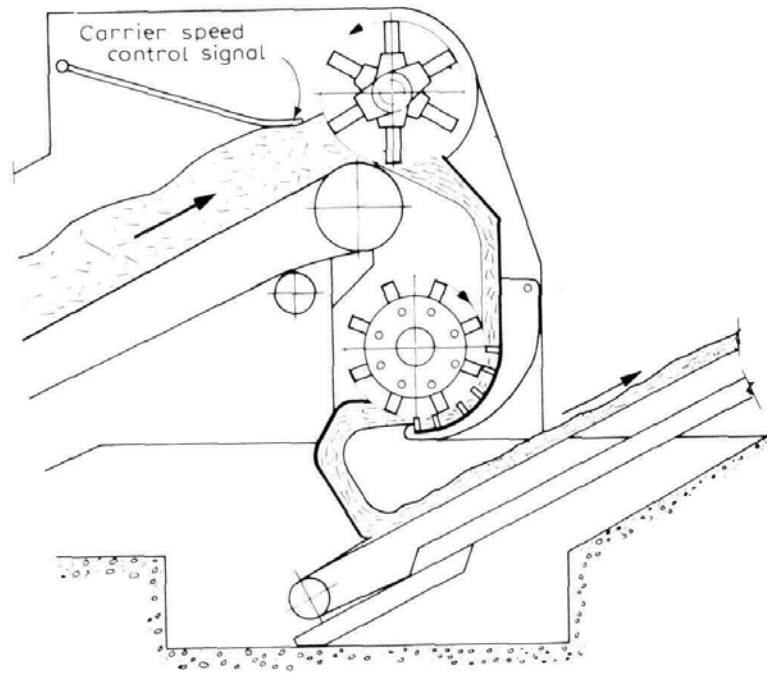


Fig. 6.5. Shredder feeding arrangement (Fletcher and Stewart).

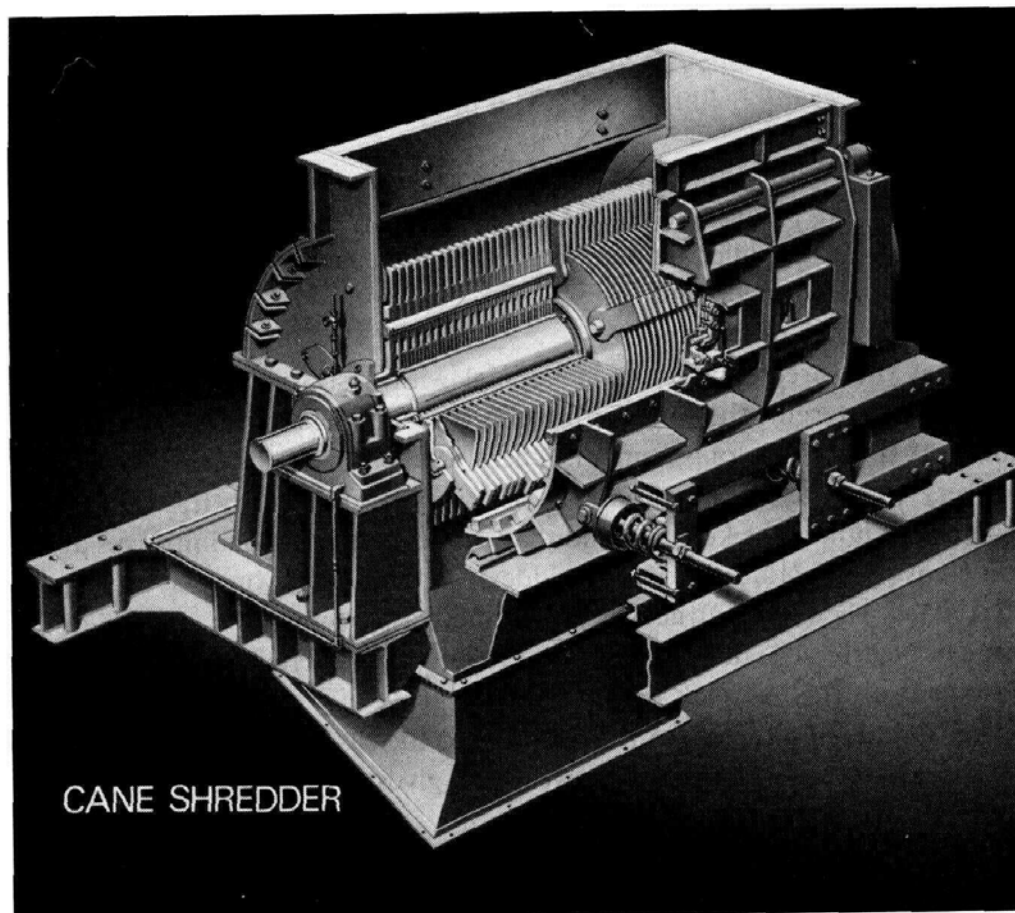


Fig. 6.6. Searby-type shredder (Fletcher and Stewart).

which indicates a very high gain as a result of the simple change in power usage at the shredder.

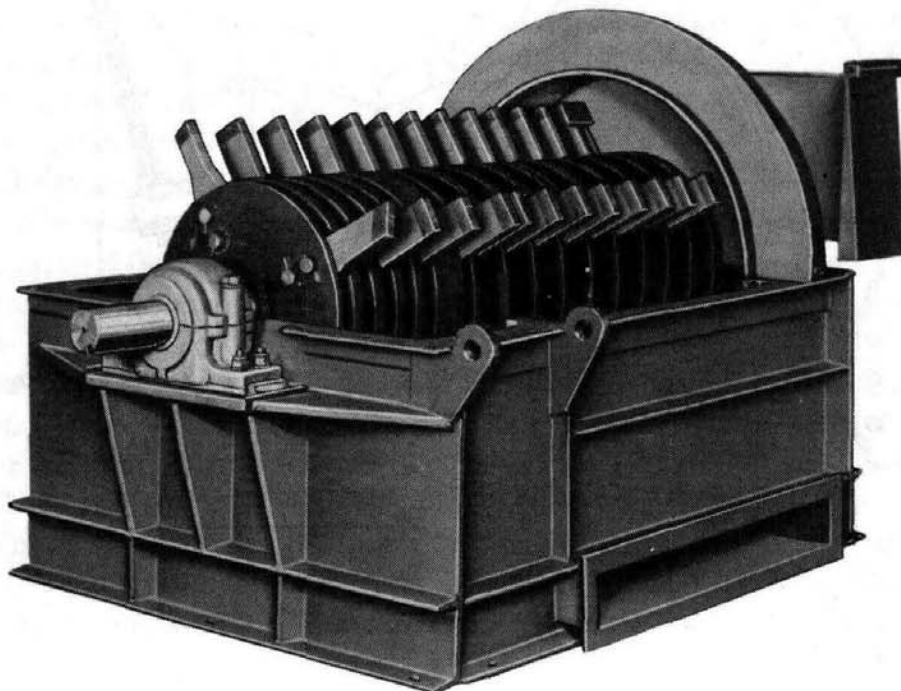
Crawford² reports tests in Queensland, consisting of measuring juice extraction of a pilot mill at three different speeds, first with cane prepared simply by knives, then with cane prepared with the same knife set followed by a shredder. The extraction in weight of juice in the three cases was higher by nearly two points for the shredded cane relative to the cane prepared with knives only. At the limits of capacity attained in each case, the results were:

<i>Roller speed</i>	<i>Knives</i>	<i>Knives and shredder</i>
4.45 m/min (14.6 ft./min)	74.9	76.6
8.9 m/min (29.2 ft./min)	72.5	74
13.6 m/min (44.5 ft./min)	69.1	71

These figures suggest that the extraction of sucrose would be improved by at least three points by the use of the shredder.

Conclusion

The hammer mill is an interesting machine, giving a definite gain in extraction. To it, to a great extent, the Hawaiian Islands owed the fine results obtained with their mills before they had to adopt push-rake harvesting. The cost is fairly high, but is rapidly repaid.



• Fig. 6.7. Gruendler shredder (rotor).

BULK DENSITY OF CANE PREPARED BY SHREDDER

Kerr⁶ gives this density as being of the order of 400 kg/m^3 (25 lb./cu.ft.). In Reunion, it is only $250\text{-}275 \text{ kg/m}^3$ (16- 17 lb./cu.ft.). In any case, it is thus lower than that furnished by passage through the two sets of knives (*cf.* p. 46). Tests in Mauritius⁷ have given $202\text{--}253 \text{ kg/m}^3$ (12.5 – 16 lb./cu.ft.). If the capacity of the following mills is increased by the supplementary passage of the cane through the shredder, this will be due mainly to the texture of the cane prepared by the shredder, characterised by long fibres and the consequent coherence of the mass.

TOTAL POWER FOR CANE PREPARATION

We have seen that the total installed power for cane preparation may vary from 24 (32) to approximately 60 kW/t.f.h. (80 h.p./t.f.h.) (average 40). This is divided into approximately 46% for the knives and 54% for the shredder. The power consumption is about 50 - 60% of the installed power, or approximately 20 kW/t.f.h. (27 h.p./t.f.h.) for the knives and $15\text{--}20$ ($20\text{--}27$) for the shredder.

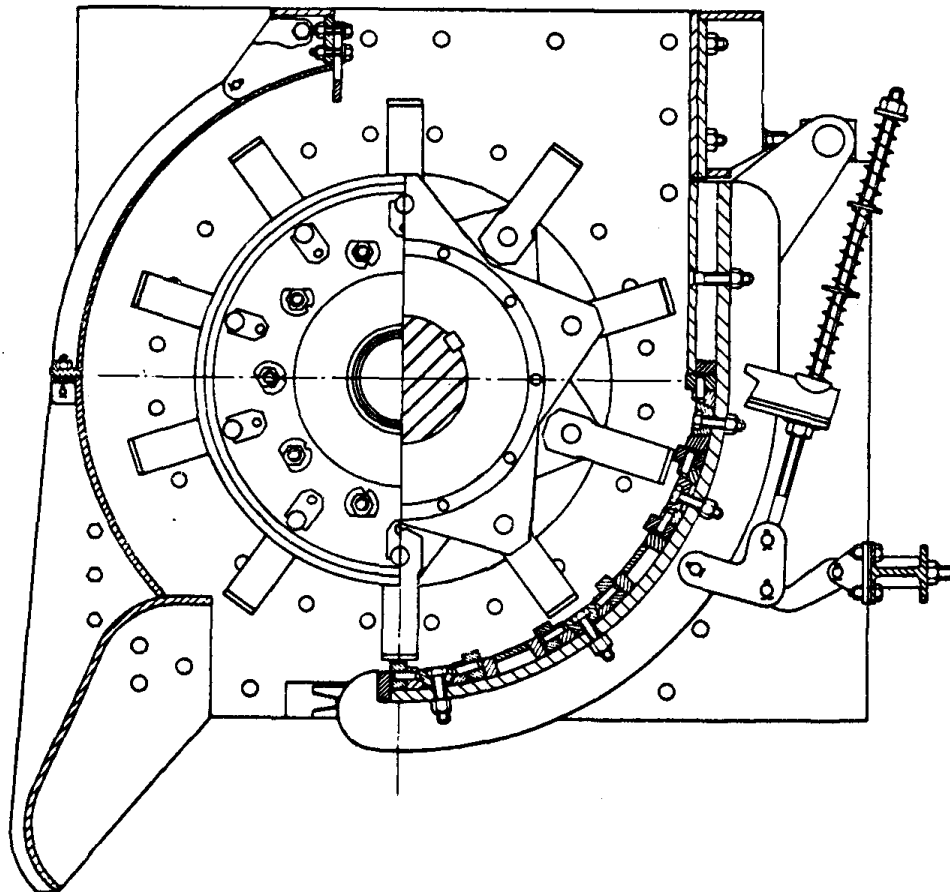


Fig. 6.8. Section view of a *Tongaat* shredder (Smith - Mirrlees).



Fig. 6.9. Shredded cane (photo Gruendler).

The power consumed for good preparation is of the same order as the power consumption in the mills themselves. In South Africa, for example, the proportion is about 46 - 54 in terms of nominal power; it varies from 38/62 to 56/44. In terms of power consumption, this proportion approaches 40/60 (preparation/mills).

Finally, the shredder is particularly valuable from the financial point of view: it provides substantial improvement in extraction, it constitutes protection against stones and tramp iron which have escaped the magnetic separator, and it costs only one-third the cost of a mill and returns sometimes much more. The Tongaat shredder has paid for itself in slightly more than one season.

Two examples, both from Tongaat, are of interest. This factory has two extraction trains. One consists of three sets of knives of 270, 520 and 300 kW (360, 700 and 400 h.p.), a shredder driven at 1,200 r.p.m. by a steam turbine of 895 kW (1,200 h.p.) and followed by 7 mills of 965 x 2,134 mm (38 x 84 in.). This set handles 31 t.f.h., and gives an I.P. of 91 and an extraction of 97.48%. Unit B consists of 2 sets of knives of 670 and 375 kW (900 and 500 h.p.), a shredder driven at 1,200 r.p.m. by a 1,500 kW (2,000 h.p.) turbine and followed by a diffuser. It handles 25 t.f.h. and gives an I.P. of 91 and 97.96% extraction.

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- 3 W. R. CRAWFORD, *Int. Sugar J.*, 72 (1970) 259.
- 4 B. ST. C. MOOR, *Proc. 15th. Cong. ISSCT, 1974*, p. 1596.
- 5 W. R. CRAWFORD, *Int. Sugar J.*, 72 (1970) 372.
- 6 H. W. KERR, *Conference in Mauritius, September 1957*, p. 235.
- 7 *Rev. Agr. Maurice, (Jan. – June 1976)* 82.

7. *Combinations of Cane Preparators*

We have just discussed the various types of equipment which are placed ahead of the mills with the object of preparing the cane, so that the pressure applied at the mills will yield maximum extraction. What is the best combination which can be made of these devices, particularly in a new factory to be built, in order to obtain the best results without excessive expenditure in investment, maintenance, or power consumption?

We shall not discuss the crusher, which is no longer of interest. There remain the knives and the shredder, or analogous disintegrators. We may try to be content with two sets of knives. However, this simplification is indicated only on condition that the first set operates at a relatively low clearance, say only a few decimetres above the carrier. Otherwise it is necessary to go to three sets, with the first as a true leveller. If the last set can be arranged to operate in the reverse direction, this would be an improvement. It should be driven by a powerful motor 20–30 kW/t.f.h. (30-40 h.p./t.f.h.) if running in the normal direction; 40-50 (55–70) for inverse rotation).

The shredder is strongly recommended, and it is advisable not to stint the power provided; allow 40-45 kW/t.f.h. (55-60 h.p./t.f.h.). It may be replaced, if desired, by a Ducasse knife set or other effective device.

Anticipating somewhat the following chapters, we may suggest a first mill of dimensions appreciably superior, in roller diameter and length, to those of the later mills; this should be followed by four or five mills, all with grooving of 49 x 50 mm. If the first mill is effectively larger than the others, its grooving could without any disadvantage be 60 x 60 mm.

The top and feed rollers would be provided with chevron grooving to the depth of half that of the teeth. These mills would run at peripheral speeds of 15 m/min (50 ft./min) for the first and 12–15 m/min (40 - 50 ft./min) for the later mills. If the factory has a capacity lower than 250 t.c.h. it would be sufficient to have five mills; at higher rates, it would be of advantage to provide six mills, and for very high tonnages, seven mills. Such decisions, however, depend on several other economic and financial factors which demand a full study of the price of sugar, costs of operation, etc. An alternative solution would be to provide five mills, all with pressure feeders (total of 25 rollers).

In planning a five- or six-mill train, it is always wise to allow space for the eventual installation of an additional mill.

8. *Feeding of Mills and Conveying of Bagasse*

The prepared cane which enters the crusher or the first mill, as well as the material which leaves the latter and travels from mill to mill to the end of the tandem (which we shall call in all cases bagasse), moves under the effects of three different forces:

- (1) Gravity
- (2) Gravity, restrained by sliding on an inclined plane
- (3) A thrust, provided by an endless belt, a pusher or other mechanical device.

These three methods act on the material all the way from its entry to the mills till it leaves the tandem.

FEED PLATE TO CRUSHER

The prepared cane arriving at the head of the carrier is fed to the crusher by a hopper, the important part of which is the bottom plate.

Length. When the cane has not been passed through a shredder and when the second set of knives is not set to a small clearance, there are always some whole sticks at the bottom of the cane layer. The length of the feed plate should be greater than that of the largest pieces, say approximately 2 m (7 ft.) in most cases.

Width. The cane carrier, the feed hopper and the crusher rollers should all be of the same width (*cf.* p. 22).

Slope. The feed plate should have a slope of at least 45° (Fig. 8.1). Whole cane will slide on a plate with a much smaller slope - 40° , and even 30° - but prepared cane slides much less readily. In either case, it is advisable that the material in the feed hopper should exert a pressure on the crusher equal to the greatest possible component of its weight.

Location. The line of the feed plate should, if produced, pass through the axis of the lower roller of the crusher (Fig. 8.2). In countries such as Cuba, where capacities are forced to the maximum, the line of the feed plate is placed somewhat to the feed side of this axis, as A_1 , with the object of increasing the space a ; but the canes on the feed plate then have some difficulty in mounting from below to feed between the two rollers; whereas, when the plane of the feed plate passes through the axis, the plane of the canes lying along the feed plate makes an angle of 90° or less with the surface of the roller, and the stalks then slide naturally towards the opening of the crusher.

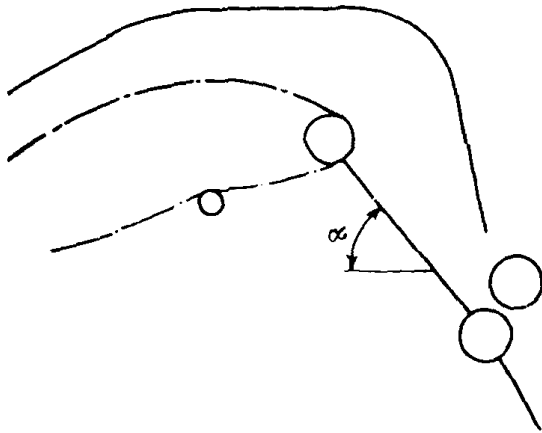


Fig. 8.1. Feed plate to crusher.

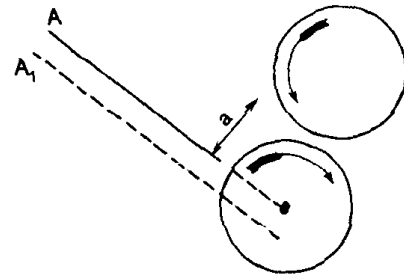


Fig. 8.2. Arrangement of feed plate to crusher.

Clearance from roller. The feed plate of the hopper should approach as close as possible to the roller. A clearance of only 6.5 mm ($\frac{1}{4}$ in.) is left, the plate being cut with a serrated edge to match the grooves.

FEED HOPPER BETWEEN CRUSHER AND FIRST MILL

At the top, the line of this feed plate should pass through the axis of the lower roller of the crusher (Fig. 8.3).

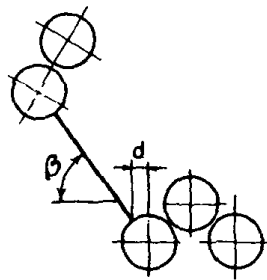


Fig. 8.3. Arrangement of feed plate between crusher and first mill.

At top and bottom, the feed plate should have a clearance of 6.5 mm ($\frac{1}{4}$ in.) from the roller surface. It should preferably be cut so as to match the roller grooving. Its inclination β should be about 50 - 55°. It should meet the bottom roller on the feed side of the vertical axial plane of the roller, and at a distance from this plane equal to $\frac{1}{6}$ of the roller diameter:

$$d = \frac{D}{6} \quad (8.1)$$

The feeding of the mill will be better with a longer chute. The length is generally made 1.5 - 2 m (5 - 7 ft.).

If, instead of allowing the feed to slide down an inclined plate as described, a vertical feed chute is used or one very steeply inclined (*cf. p. 87*), feeding can only be better.

INTERMEDIATE CARRIERS

The intermediate carriers are the conveyors which move the bagasse from one mill to the feed of the next. There are three principal types:

- (a) Drag conveyors, or rake conveyors
- (b) Moving-band conveyors, apron or slat conveyors
- (c) Fixed carriers, Meinecke type.

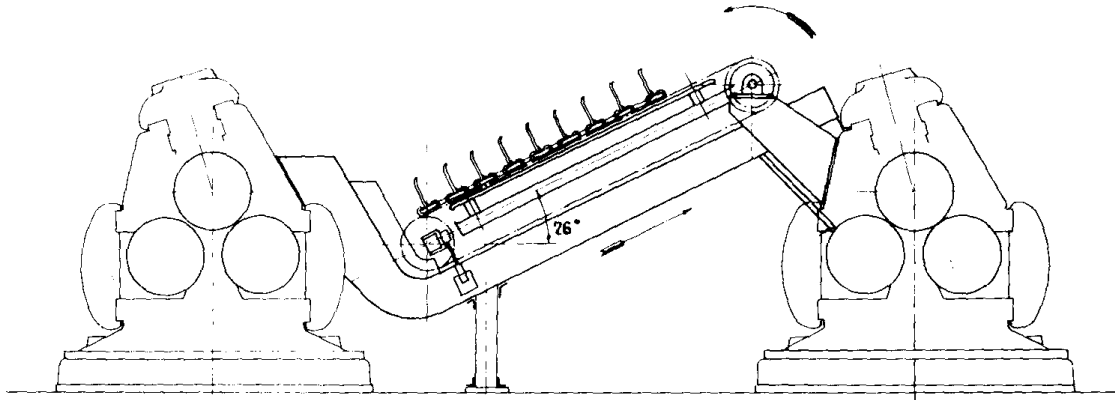


Fig. 8.4. Rake-type intermediate carrier.

(a) Drag conveyors (Fig. 8.4)

In these it is the lower run of the conveyor which is the active one. The conveyor is driven from its top sprockets, which in turn are driven from the top roller.

The headsprockets and returnsprockets are now installed mounted on one rigid frame, which is pivoted about the axis of the upper sprockets so that it can rise, if necessary, when the layer of bagasse becomes thicker. This frame carries the runners for the upper strand of the conveyor, and these will be fitted with a pad of hardwood to avoid shocks.

The chains are fitted with angle-irons carrying the teeth and forming the rakes. A special corner-piece serves to attach them to the links of the chain (Fig. 8.5).

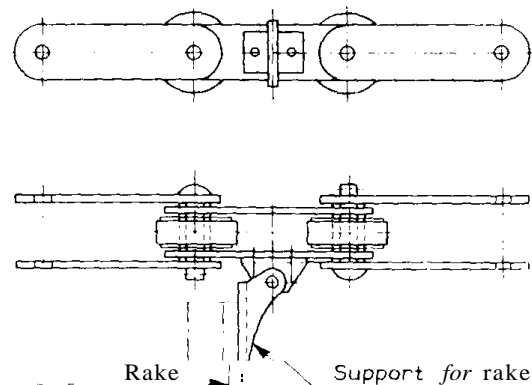


Fig. 8.5. Attachment of angle-irons.

The frame should be supported at its lowest position in such a way that the ends of the tines of the rakes pass about 22 mm (4 in.) from the bottom plate of the carrier.

The thickness of bagasse layer entrapped is limited by the bottom plate of the carrier on the one hand and by the free wing of the angle-iron on the other. The thickness of bagasse, which fixes the lowest position, may be calculated by assuming that the bagasse blanket is continuous, that it has the same speed as the conveyor, and that it contains 95 kg of fibre per m^3 (6 lb./cu.ft.). The adoption of a weight of fibre per unit volume avoids the necessity of considering the different specific weights of bagasses from different mills, which vary according to the rate of imbibition, whereas the volume of these various bagasses is nearly constant and independent of the imbibition rate and of the number of the mill. On the other hand, the value adopted, 95 kg/ m^3 , indicates that allowance has been made for a certain compression exerted by the conveyor on the bagasse.

Advantages and disadvantages. Drag conveyors are simple, but wear quickly, because they work in acid juice and bagasse. They contribute nothing to the feeding of the following mill.

Due to wear, they are subject to breakages of the angle-irons, or of the rake teeth. A broken tooth generally escapes unnoticed into the bagasse, and passes through the following mills, damaging the roller grooving. The acid bagasse, rubbing on the bottom plate of the carrier, also corrodes it.

For these various reasons, it pays to make the roller bushes in phosphor-bronze, the pins and links of the chains and the bottom plates in corrosion-resisting steel.

Location and arrangement. The angle of the feed plate to the following mill, which receives the bagasse delivered by the rake conveyor, is not unimportant (Fig. 8.6). This angle should be fairly steep in order to permit the bagasse to slide freely on the plate, but should not exceed the value at which it would produce an "avalanche" of bagasse. Now the bagasse slides above an angle of 40° or 45° , according to its moisture content, and it avalanches at an angle of $55-60^\circ$ (bagasse soaked with imbibition). The angle β should therefore be about 50° , preferably $52-54^\circ$.

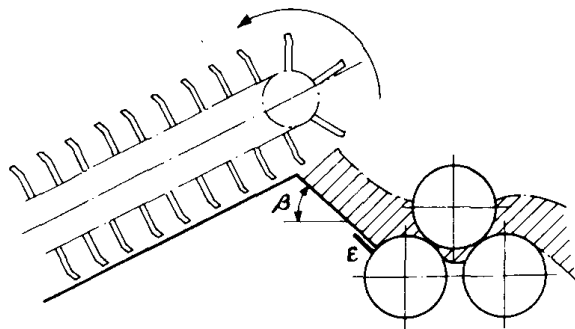


Fig. 8.6. Feed plate of an intermediate mill.

The space ϵ left between the feed plate and the feed roller should be as small as possible, especially in the later mills, to avoid dropping fine bagasse in front of the feed roller (Fig. 8.6). For this purpose, the feed plate should terminate in an adjustable plate held by three

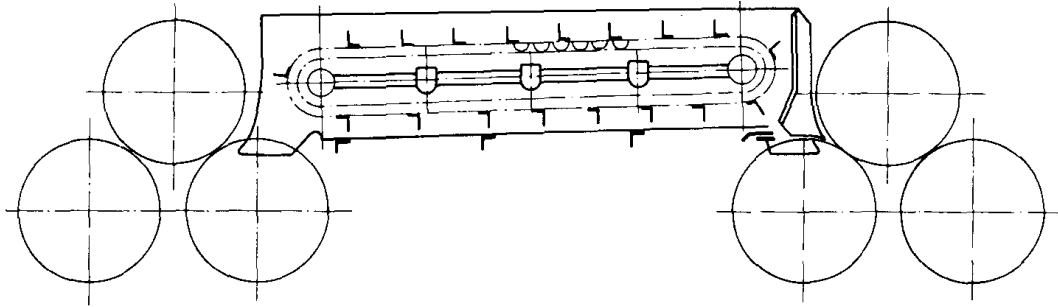


Fig. 8.7. Horizontal intermediate carrier (Ewart).

bolts, and cut to match the grooving. This precaution is not always necessary except for the later mills.

Horizontal carriers. Certain manufacturers make the rake conveyor horizontal (Fig. 8.7), with the bottom run tangential to the upper surface of the bottom rollers. This makes it lighter and simpler; a useful feature, especially when bottom rollers are to be removed.

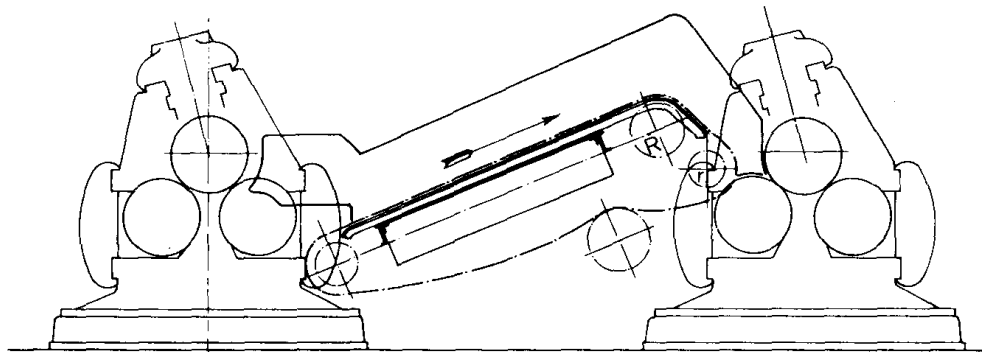


Fig. 8.8. Slat-type intermediate carrier.

(b) Slat conveyors or apron conveyors (Fig. 8.8)

Here it is the upper part of the conveyor which carries the bagasse. The conveyor is driven by the upper sprockets at the head of the conveyor, which in turn are driven from the feed roller.

The carrier consists of a continuous apron, composed of overlapping steel slats, which are able to move relative to each other (Fig. 8.9), as in the cane carrier.

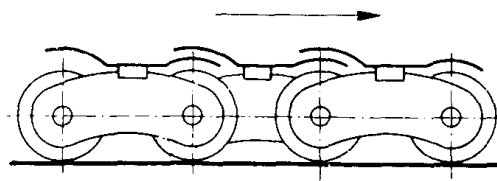


Fig. 8.9. Slat conveyor (Fives Cail - Babcock).

The head of this type of carrier is always provided with idlers in such a way that a portion of the moving apron forms the feed plate of the following mill and so helps to push the bagasse into the mill.

Advantages and disadvantages. These slat conveyors have the disadvantage of being heavier than the foregoing type. The slats carry, nipped between them, fragments of bagasse which they drop between the mills, making the floor untidy if the plant is not provided with a continuous juice tray. When the following mill has a tendency to choke, it often happens that the slats become bent owing to the pressure exerted by the bagasse accumulated between the top roller and the part of the conveyor forming the feed hopper.

However, this system has the advantage of assisting feeding by virtue of the pressure of the slats against the bagasse in the feed hopper. This pressure is slight but not negligible, especially when it is assisted by a feeding roller on the bagasse. Further, wear is less than in the rake type.

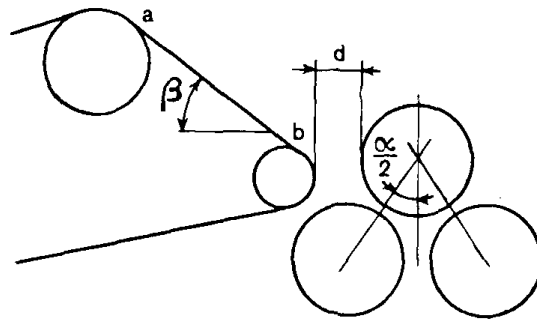


Fig. 8.10. Feeding of mill with slat-type intermediate carrier.

Arrangement. The plane of the feed-hopper portion should be tangential to the feed roller (Fig. 8.10).

The horizontal distance d (Fig. 8.10) between the leading sprockets and the top roller of the mill should be approximately equal to $\frac{1}{3}$ of the roller diameter:

$$d = D/3 \quad (8.2)$$

The plane ab should be as nearly as possible normal to the axial plane of the top and feed rollers, so that the pressure exerted will not be partially lost by acting against the roller surfaces. This corresponds to an inclination equal to half the top angle of the mill (*cf.* p. 261), or about 34-40°:

$$\beta \approx \frac{\alpha}{2} \quad (8.3)$$

Inclination of the sloping portion. The rising portion of rake conveyors or apron carriers is generally about 25°. However, in very compact tandems, it goes as high as 30°. When for any reason the space available between the two mills is limited, it is possible¹, with a rake conveyor, to exceed this value substantially, and to go as high as 50° and even 60°. So that the bagasse will not adhere to the rakes, it is of advantage to give the carrier, at its upper extremity, an end portion at 30°; this is effected by means of two idler pulleys placed at the point of change of slope.

Speed. The linear speed of intermediate carriers should be higher than the peripheral speed of the mill rollers. There is no disadvantage in exceeding these speeds, which are very low; it permits the thickness of the bagasse layer to be reduced, and especially gives the effect of positive feeding sought in the apron carrier.

Maxwell² indicates a speed 50 - 70% higher than the peripheral speed of the mill rollers for rake conveyors, and (p. 200) a speed equal to that of the rollers for apron carriers. Tromp³ advises a speed higher by 7 - 10% for apron carriers. Fives Lille compute about 10% for apron conveyors, and up to 200% higher for carriers of the rake type. We have then:

$$V = 1.5 - 3 V \text{ (rake carriers)} \quad (8.4)$$

$$V \approx 1.1 V \text{ (apron carriers)}$$

V = linear speed of intermediate carriers

V = peripheral speed of the mill rollers, in the same units.

When necessary, it is possible to push the speed of intermediate carriers¹ as high as 40 and even 60 m/min (130 and 200 ft./min). These are abnormal speeds, not generally useful.

Power. The mean power consumption of the intermediate carriers is generally scarcely a matter of concern, since this power is furnished by the mill itself, and in a way forms an integral part of the power required to operate the mill. We shall consider it again later on under this heading (Chap. 14).

Roughly, we may indicate as order of magnitude:

$$T = 0.075 A \quad (T = 0.1 A) \quad (8.5)$$

T = mean power absorbed by an intermediate carrier, in kW (h.p.)

A = crushing rate, in t.c.h.

This power represents approximately 5% of the total power required for the mill.

As we shall see later, these two indications are given only to show typical values; the actual power, as well as the relative power consumption, can vary very substantially above or below these values.

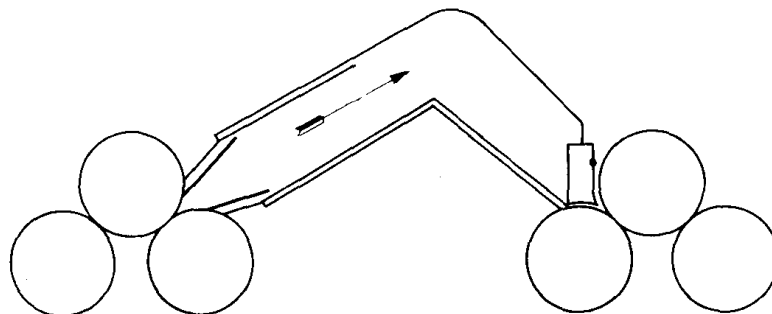


Fig. 8.11. Fixed intermediate carrier (Meinecke).

(c) **Fixed carriers**

These are typified by the Meinecke conveyor (Fig. 8.11). This is an ingenious system, which avoids all moving mechanism: only the bagasse moves from one mill to the next. The bagasse leaving the mill pushes up an inclined plane the bagasse which has already left the mill; when it has attained a certain height along the ascending plane, it slides of its own accord down a second plane sloping in the opposite direction.

The inclination of the rising portion is about 30° ($27 - 35^\circ$), that of the descending portion about 40° . Consequently, the highest point is well beyond the mid-point of the distance between the two mills.

In order to force the bagasse leaving the mill to move as a blanket instead of accumulating in a heap, it is enclosed between the scraper plates, which are placed close to the delivery opening forming a dihedral angle of about 10° ; these are prolonged far enough for the friction on their surfaces to compress the bagasse and give it a certain cohesion, thus permitting it to mount the inclined portion of the chute as a solid blanket without breaking.

Advantages and disadvantages. From the point of view of cost, wear and maintenance, this system would obviously be all that could be desired. Unfortunately, it lends itself badly to:

(1) The feeding of the following mill, which has to grip the bagasse without assistance: we have seen with rake carriers that a feed-plate angle of 50° is necessary for proper feeding. However, this is not provided, as it would necessitate too long a rising portion.

(2) The use of imbibition, and this is much more serious. If imbibition is added as usual at the delivery of the mill (this would be possible with a nozzle inside the scraper plates), it would not be possible to move the bagasse with the added weight of the liquid. If it is applied at any point of the descending portion, the bagasse blanket is broken, and the distribution of imbibition is defective. In addition, many people are opposed to the idea of imbibition applied close to the following mill, considering that it is less effective than if applied as the bagasse leaves the preceding mill.

Though the latter argument has little foundation, the difficulties described as regards feeding, and particularly that of the proper application of sufficient imbibition, have prevented the extension of this system, though it is otherwise very interesting. Moreover, it is applicable only in very compact tandems, in which the mills are very close to one another.

In Cuba, at central Punta Alegre, **Gonzalez**²¹ has arranged to avoid these disadvantages and to obtain complete satisfaction from Meinecke intermediate carriers, by adopting the following arrangements:

(a) Slope of the rising portion $27 - 30^\circ$

(b) Slope of the descending portion $43 - 45^\circ$

(c) The angle at the top of the conveyor between the two branches was replaced by an arc of a circle of radius $0.47D$ ($D =$ diameter of mill rollers) or (in the case quoted) exactly 457 mm (18 in.), the centre of the arc being at $2.84D$ from the axial vertical plane of the top roller of the preceding mill, and $2.21D$ from the vertical plane of the top roller of the following mill. The mills were thus at $5.05D$ between their axes. The bagasse layer then cracked at the summit without actually breaking. Imbibition was added above the summit point by a spray, under a pressure of 3.5 kg/cm^2 (50 p.s.i.).

Feeding of the mill was effected by a feeding roller below the bagasse, of diameter $0.47D$, with 12 ribs each 19 mm ($\frac{3}{4}$ in.) square and rotating at $v' = 1.36 v$ ($v =$ peripheral speed of the mill rollers) and tangential to the plane of the descending portion.

Setting of the Meinecke scrapers. Gonzalez recommends that the entry opening between the scrapers should be:

1st mill: $5.5 e_A$
 2nd mill: $6.5 e_A$
 3rd mill: $7.0 e_A$
 4th mill: $7.5 e_A$
 5th mill: $8.0 e_A$
 6th mill: $8.5 e_A$

($e_A =$ delivery work opening of the mill)

The opening at the parallel portion following the tapered section should be 1.23 times the entry opening.

Belt-type carriers

The cost of chains and slats of apron-type carriers has suggested the idea of replacing them by a simple rubber belt, as in belt conveyors (Fig. 8.12). The belt simply passes over two drums, an upper driving drum and a lower return drum. The assembly thus becomes very light and may be driven by a small electric motor and reduction gear situated close to, and sometimes even inside, the upper drum. A flexible scraper, placed above the feed roller and at the top of the return run of the conveyor, removes the pieces of bagasse which stick to the belt due to the imbibition.

These belt carriers form a modern and economical solution. Initial cost is no higher than

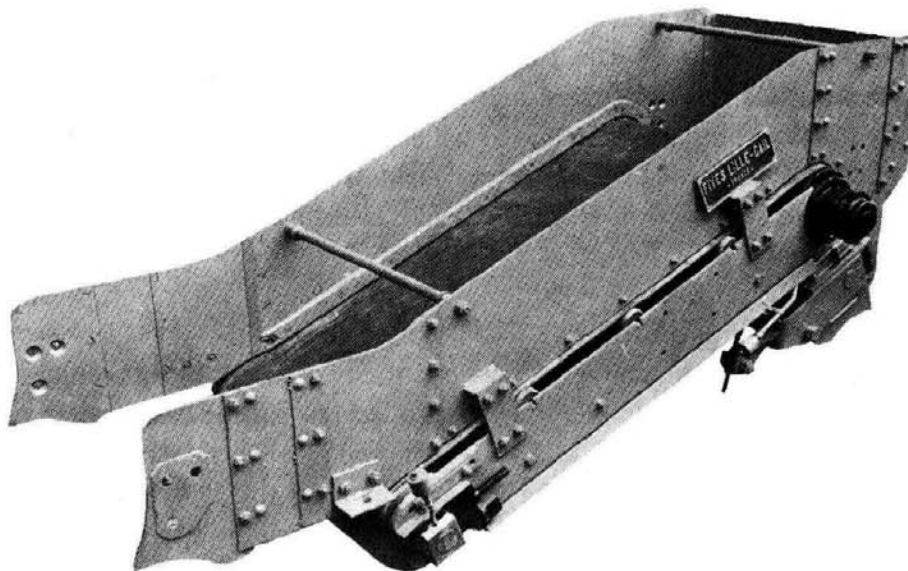


Fig. 8.12. Belt-type intermediate carrier.

for other types, and upkeep cost is much lower. The belt hardly wears at all, provided that it runs true and does not rub on the edges. To ensure this, it is necessary to set the two drums strictly parallel, to maintain the belt well centred by a very slight convexity of the upper drum and, above all, to take all precautions to avoid access of bagasse, even bagasse powder, between the belt and the drums. With this object, it is necessary to provide a belt wider than the length of the rollers by approximately 10 cm (4 in.) on each side, and to support it on the sides of the upper run by a number of small rollers slightly inclined towards the middle of the belt, also to provide a flexible rubber strip to prevent loss of bagasse between the side plates of the carrier and the belt. It is necessary also to block on both sides the free space between the upper and lower runs of the belt, by means of a plate extending from one drum to the other.

The maximum slope of a rubber belt conveyor is 27° (51%); it should preferably be $24 - 25^\circ$. The belt is generally 5-ply, with a layer of 1 mm on the underside and 2 mm above.

High-speed carriers

As soon as the belt had rendered the carrier so light, flexible and manageable, it was indicated that it could be run faster. Maxime Riviere, in Reunion, has exploited this idea thoroughly and put into operation a high-speed intermediate carrier. His main objective was to reduce the bagasse on the carrier to a layer so thin that the imbibition could contact all the bagasse in transit. With ordinary intermediate carriers, it is difficult to distribute the imbibition over all the bagasse in a uniform manner, since the surface layer absorbs most of it at the expense of the bulk of the material. To accentuate the advantage, Riviere added a distributor-thrower, acting on the bagasse as it left two Meinecke scrapers, and throwing it onto the belt in a uniform layer (Fig. 8.13). The belt runs at a speed of approximately 8–20 times the

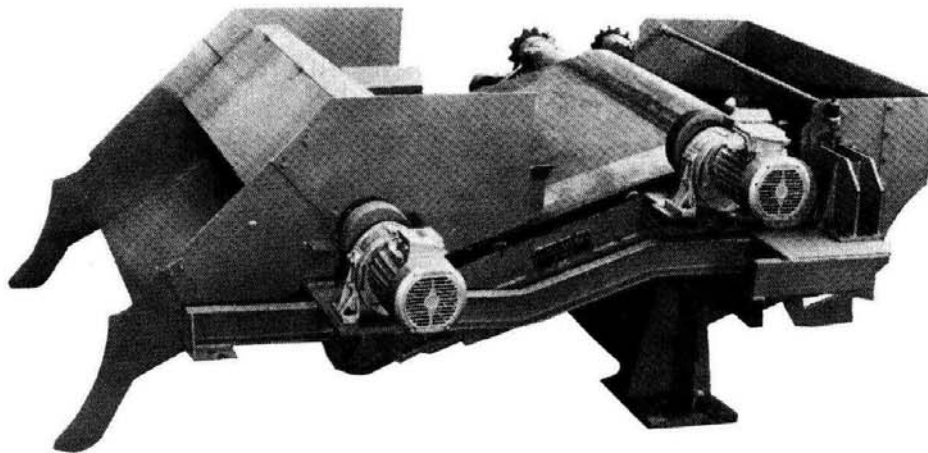


Fig. 8.13. Riviere-type high-speed-belt intermediate carrier.

peripheral speed of the rollers, generally 75–140 m/min (250–450 ft./min). Under these conditions, the bagasse is spread in a layer approximately 13–25 mm ($\frac{1}{2}$ –1 in.) in thickness, and the belt may be seen through the layer. An imbibition trough distributes the water or the juice on a plate of the Maxwell type, assuring a continuous sheet of liquid across the full

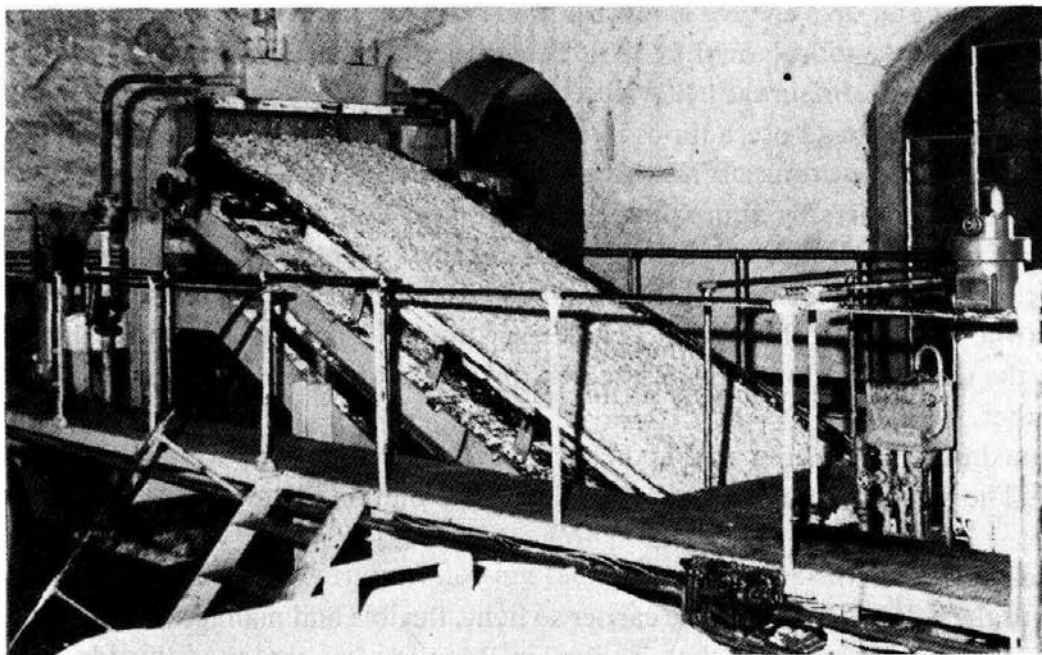


Fig. 8.14. Bagasse layer on high-speed carrier feeding a Donnelly chute

width of the carrier. None of the devices for distribution over a thick layer (*cf.* p. 93) gives such a regular distribution of the imbibition (Fig. 8.14).

The motor driving the belt is of about 2.5 — 3.5 kW (3 — 5 h.p.), that for the distributor-thrower 4.5 — 6 kW (6 — 8 h.p.). The latter should have a diameter at the bottom of the teeth of about half the roller diameter, and a number of teeth of about 100 — 115 per m (30 — 35 per ft.) of width. These can be distributed over 12 rows around the small roller which carries them. It should rotate at 300-400 r.p.m.

We have not ascertained that, in the range indicated, the belt speed has a perceptible effect on the extraction: 120 m/min (400 ft./min) shows no difference relative to 60 m/min (200 ft./min), although the bagasse layer should theoretically be half the thickness in the former case. We also consider that there is no advantage in going to a very high speed, and that a good belt speed is that which throws the bagasse just into the feed opening of the mill. This condition appears favourable to feeding the mill, and should be preferred to a high belt speed which throws the bagasse against the top roller.

This good regular feeding is one of the favourable characteristics of the high-speed carrier. Although its original purpose was the optimal distribution of the imbibition, the benefit to be obtained from this should not be overemphasised: actually, an even distribution of the imbibition is always effected in the nip of the two front rollers, the excess liquid from the wet portion being expressed by the pressure and rapidly absorbed by the dry portion. For this reason, an excellent distribution of imbibition on the carrier is not really valuable except for low imbibition rates, for which the automatic correction in the mill is insufficient. We know that some factories in South Africa, using heavy imbibition, have tried to Riviere carrier and have abandoned it, as not improving their extraction. However, we are in favour of the system: it assures good use of imbibition, assures very effective feeding and is not expensive either in initial cost or in upkeep.

The high-speed intermediate carrier can be adapted very well for use with an underfeed roller (*cf.* p. 92). The latter is then useful only in the case of an overload, but the combination proves effective if the two systems are well regulated (conveyor speed and adjustment of the underfeed roller).

One-piece intermediate carriers. The intermediate carrier obstructs access to the mill, and must be removed every time rollers have to be removed or changed. To save labour and loss of time, it is made today in such a manner that it can be raised vertically in one piece by the travelling crane, the side plates forming one assembly with the carrier proper. Hooks fixed at the four extremities of the side plates allow the carrier to be lifted bodily by means of two slings. This is an improvement leading to a saving of valuable time, which is very necessary.

DELIVERY PLATE AT LAST MILL

The relatively dry bagasse from the last mill, which does not receive inhibition, slides more easily than wet bagasse. However, even on a smooth surface such as steel polished by friction, or waxed wood, it still requires a slope of 38 - 40° to slide freely. It is wise to allow 40° in general, and not to go lower than 45° on an inclined plane of any length, to avoid the risk that bagasse accidentally wetter than normal may refuse to slide.

These values are obviously valid also for the feed chutes to boiler furnaces.

FEEDING ARRANGEMENTS

Mill feed by gravity

We should have no illusions on the feeding effect due to the pressure generated by a slat conveyor. This action is very feeble and it is sufficient, to demonstrate this, to interpose one's hand in the layer of bagasse carried by the conveyor.

Gravity feed, that is, the feeding effected with the aid of a simple chute (Figs. 8.4 and 8.6) in the mill hopper which follows a rake conveyor, does not give any direct "push"; the component of bagasse weight parallel to the feed plates acts to a great extent against the surface of the feed roller (Fig. 8.6). This feeding force, however, is not lost; on account of the friction of the bagasse, it produces a corresponding adhesion of the bagasse to the feed roller, and hence a feeding force which, while being indirect, is no less effective.

In standard rake carriers (Fig. 8.6) this effect is not superior to the pressure given by slat conveyors (Figs. 8.8 and 8.10), but it is possible to increase it substantially.

We shall calculate the pressure in kg/dm² (lb. per sq.ft.) which is exerted at the bottom of the feed chute on a section perpendicular to the plane of the feed plate. It has a value (Fig. 8.15).

$$F = \frac{P}{s} (\sin \beta - \phi \cos \beta) \quad (8.6)$$

P = weight of bagasse, in kg (lb.)

ϕ = coefficient of friction of the bagasse on the feed plate

S = area of the section considered, in dm^2 (sq.ft.)

β = angle of inclination of the feed plate.

We have:

$$\frac{P}{S} = \frac{LEZd}{LE} = Zd \quad (8.7)$$

L = width of rollers, in dm (ft.)

E = thickness of the bagasse layer, in dm (ft.)

Z = length of the feed plate, in dm (ft.)

d = apparent density of the bagasse, in kg/dm^3 (lb./cu.ft.):

d is of the order of 0.40 (25) for the first mill (without compaction), but can vary from 0.30 to 0.60 (20 to 40) (0.60 with compaction⁴), and may be taken in the other mills as approximately equal to:

$$d = 0.10 \left(\frac{1}{1-h} + \lambda \right) \quad \left(d = 6 \left(\frac{1}{1-h} + \lambda \right) \right) \quad (8.8)$$

h = moisture per unit of bagasse

λ = imbibition per unit of fibre (*cf.* p. 90).

Hence d = say approximately 0.30 — 0.40 (20 — 25), according to the quantity of imbibition.

The coefficient of friction ϕ depends on the bagasse and on the degree of polish of the

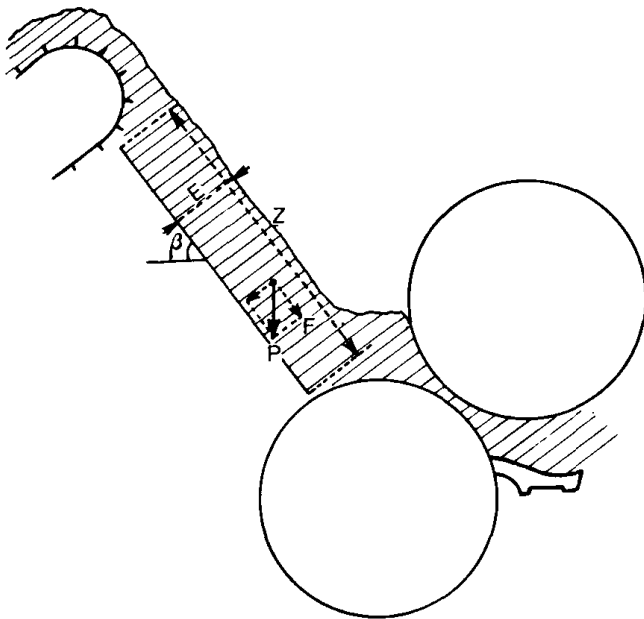


Fig. 8.15. Gravity feed.

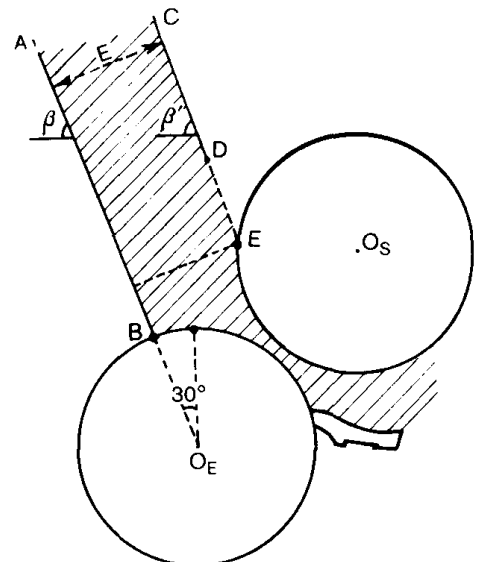


Fig. 8.16. Donnelly chute.

feed plate. We may take in general $\phi = 0.25 - 0.30$ (the values given on p. 99 apply to grooved rollers and not to smooth plates).

To be more precise, the bagasse density increases with increasing vertical height of the accumulated bagasse. We shall be content here with the mean values given above.

We have then:

$$F = Zd (\sin (3 - \phi \cos \theta)) \quad (8.9)$$

Giving d and ϕ the mean values, we obtain Table 8.1.

We see from this table the great value of long chutes and steep angles, where these are practicable. At Monymusk⁵ the feed plate of the first mill is 6.5 m (21 ft.) long with a slope of 53°. At Pioneer (Australia) the first mill has a 3-m (10-ft.) chute at 65°, the fourth mill a chute of 4.25 m (14 ft.) at 60°, the third mill 3.35 m (11 ft.) at 90° (vertical). Donnelly chutes are now generally 3 - 4 m (10 - 16 ft.), and vertical.

TABLE 8.1

VALUE IN kg/dm^2 OF THE COMPONENT OF GRAVITY FEED AS A FUNCTION OF THE LENGTH Z AND THE INCLINATION θ OF THE FEED PLATE ($d = 0.4 \text{ kg}/\text{dm}^3$; $\phi = 0.30$)

Z (m)	$\theta = 50^\circ$	$\theta = 60^\circ$	$\theta = 70^\circ$	$\theta = 80^\circ$	$\theta = 90^\circ$
1	2.1	2.6	3.1	3.4	3.7
2	4.2	5.2	6.2	6.8	7.4
3	6.3	7.8	9.2	10.2	10.8
4	8.4	10.4	12.0	13.6	14.6
5	10.5	13.0	15.3	16.8	18.3

Feed hopper. With a feed plate slope greater than 55°, it becomes essential to add a front plate CD as the bagasse tumbles forward above this value (Fig. 8.16).

With gravity feed chutes, a feed hopper $ABCD$ will therefore be adopted, the bottom plate AB of which will meet the feed roller at a point B situated 30° before the vertical axial plane; the plane of the front plate CD will meet the top roller at a point E situated on the horizontal axial plane of that roller. The thickness E of the hopper will vary slightly then with normal values of the angle β (55 - 90°). It should be of the order of 0.40 - 0.75 of the roller diameter, according to their peripheral speed v : 0.40 for $v = 14 \text{ m}/\text{min}$ (46 ft./min), 0.75 for $v = 10 \text{ m}/\text{min}$ (33 ft./min) (Donnelly⁶).

Van Hengel⁷ considers that these values are too high and suggests reducing them to: $E = (11 \text{ or } 12) e_A$ for a first or second mill and $8 e_A$ for a final mill, where e_A is the delivery opening of the mill while operating. However, in his calculation he does not take into account either the slip or the coefficient of drag of a column of bagasse on the surface of the roller. Now even if there is no slip, the roller does not pick up the bagasse at its surface speed v (or $v \cos a$ if the bagasse column comes into contact with the roller surface at an angle a ahead of the vertical diameter) except for the surface layer in contact with it. The speed of movement v' of the bagasse layer as a whole can be only a fraction of v , often: $v' = 0.6 v \cos a$ (being moved by one surface only) or $0.8 v \cos a$ when gripped between the top roller

and the feed roller. Thus we would recommend, while following Van Hengel's reasoning, the adoption of the following:

<i>Mill</i>	<i>Value of the thickness E of the feed hopper (or space between top roller and feeding roller)</i>	
	<i>Hopper without feed roller</i>	<i>With feed roller</i>
1st or 2nd mill	16 e_A	14 e_A
Final mill	12 e_A	10 e_A

provided that these values do not exceed those proposed by Donnelly.

Munro⁸ cites the case of Isis Factory in Australia which had obtained an excellent result by adopting a chute at 80° slope, the cross-section of which, where it met the rollers, was 5 dm² per unit (kg/m²) (2.6 sq.ft. per lb./sq.ft.) of fibre loading.

To obtain the desired value of E , the position of point B will be adjusted, as this may vary slightly from that indicated.

If β is less than 60°, we may take for B the point situated at (90° - β) ahead of the vertical axial plane of the feed roller.

The total pressure of the bagasse on the feed roller is equal to the product of the pressure F and the appropriate area; the latter is the area of roller surface in contact with the bagasse, projected on a plane perpendicular to AB .

We consider that the best angle β would be in the neighbourhood of 80°. An angle of 90° would obviously be the ideal, if it could be arranged without difficulty from the point of view of supporting the feed chute.

Since D represents the lower edge of the front plate of the feed hopper, the distance DE can without disadvantage be 0.3 m (1 ft.). This allows an observer to see the bagasse and to gauge the compactness of the mass at the bottom of the feed chute.

The feed hopper should widen out slightly towards the bottom, that is, the front plate CD should have a slope β' about 1° less than the slope β of the plate AB . This avoids any resistance and tendency to arching; it renders negligible the friction on the front plate, and permits the component of the feeding force to reach its maximum value.

The coefficient of friction ϕ' of bagasse on the feed roller is greater than ϕ , since the surface of the roller is designed to give the best grip possible, with its grooving and with a suitable grain of the metal. We may take in general, for grooved rollers, $\phi' = 0.30$ (polished metal) to 0.40 (rough surface).

High tonnages and feeding roller. The foregoing arrangements (Fig. 8.16) sometimes prove insufficient in tandems working at maximum capacity; the feed hopper becomes too narrow. We indicate in Fig. 8.17 another arrangement corresponding to the maximum width permissible for the feed hopper. It consists of locating the plates of the hopper to meet the top roller at a point E and the feeding roller at a point F , given by the figure.

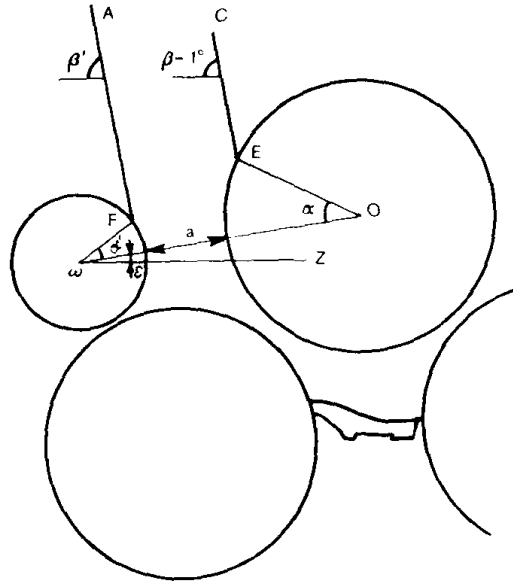


Fig. 8.17. Location of feeding roller with Donnelly chute.

We shall consider the case of the feeding roller, since it is very useful at such maximum tonnages. The bottom plate *AF* may be arranged as already indicated, and the feeding roller then set so as to give the angle α' such that

$$\begin{aligned} \alpha' &= 135^\circ - 03 + e) \\ \beta &= \text{slope of the plate } AF \\ f &= \text{angle } O\omega Z = \text{angle of } O\omega \text{ with the horizontal.} \end{aligned} \tag{8.10}$$

The upper plate *CE* will meet the roller at *E*, a point chosen such that the angle $a = \omega OE$ is defined by:

$$\cos \alpha = \frac{1}{2} \left(1 + \frac{a}{D} \right) \tag{8.11}$$

a = space between feeding roller and top roller

D = roller diameter.

Conclusion. When the space between mills permits, and when it is possible to install a feed plate at least 2 m (6 ft.) in length, gravity can provide a very positive and effective feeding force, superior to that obtained with most of the standard feeding devices such as pushers or feeding rollers.

It is particularly indicated for a first mill, where it is easier to install a high chute. The chute should then not be interrupted by a magnetic separator, which in this case should be located elsewhere.

With the later mills, it is all the more attractive with high imbibition. The feeding component is proportional to the density *d*, which increases with imbibition (*cf.* eqn. (8.8)). With an imbibition of $X = 2$, it is already more than double the density of bagasse without imbibition.

Establishment of eqn. (8.8). The apparent density of bagasse at 45% moisture, in a layer of 0.3 m (1 ft.) or thereabouts, is of the order of 0.180 kg/dm³ (11 lb./cu.ft.). Now 1 dm³ (1 cu.ft.) of this bagasse contains:

$$180 \times 0.45 = 80 \text{ g of water (11} \times 0.45 = 5 \text{ lb. of water)}$$

and slightly less than:

$$180 - 80 = 100 \text{ g of fibre (11} - 5 = 6 \text{ lb. of fibre)}$$

As the volume of bagasse varies but little with its moisture (in fact, under a thick layer, its volume decreases slightly when its moisture increases, on account of the compaction of the lower layers under the increasing weight of the upper layers), we shall assume that the weight per dm³ (cu.ft.) is represented by 100 g (6 lb.) of fibre, to which is added the weight of contained water.

Consequently, if we designate by:

h = moisture of the bagasse, in weight of water per unit weight of wet bagasse, before imbibition,

W = weight of water per unit weight of dry fibre in the bagasse after imbibition,

h' = moisture of the bagasse after imbibition,

X = ratio of weight of imbibition water (or juice) to weight of fibre (*cf.* p. 285).

f = weight of fibre per unit of cane = fibre in cane,

F = weight of fibre per unit bagasse = fibre content of bagasse before imbibition,

B = weight of bagasse per unit of cane, before imbibition:

Neglecting materials in solution in the juice, we have:

$$W = \frac{h'}{1 - h'}$$

Considering for the moment the weight relative to unit of cane, the weight of imbibition water will be λf and the weight of water in the bagasse, after imbibition:

$$Wh + \lambda f = h \frac{f}{1 - h} + \lambda f = f \left(\frac{h}{1 - h} + \lambda \right)$$

After imbibition, the moisture of the bagasse becomes:

$$W = \frac{f \left(\frac{h}{1 - h} + \lambda \right)}{f + f \left(\frac{h}{1 - h} + \lambda \right)} = \frac{h + \lambda (1 - h)}{1 + \lambda (1 - h)}$$

The apparent density of the bagasse, after imbibition, is then:

$$d = 0.10 (1 + W) = 0.10 \left(1 + \frac{h'}{1 - h'} \right) = \frac{0.10}{1 - h'} = 0.10 \left(\frac{1}{1 - h'} + \lambda \right) \quad (8.8)$$

$$\left(d = 6 (1 + W) = 6 \left(1 + \frac{h'}{1 - h'} \right) = \frac{6}{1 - h'} = 6 \left(\frac{1}{1 - h'} + \lambda \right) \right)$$

"Dirty" top roller

It is current practice in Queensland to allow the grooves of the top roller to fill with bagasse. For this it is necessary to use a smooth scraper plate, instead of one meshing with the grooves, and to set this plate so that it has a clearance of approximately 6.5 mm ($\frac{1}{4}$ in.) from the top of the roller teeth.

Generally this in itself is not sufficient. It is necessary that the top roller should have grooves with an angle of $32-35^\circ$, and that the metal surface should be sufficiently rough. This latter condition is obtained in Queensland by spotting the top and sides of the teeth by means of a carbon electrode, an operation described as "arcing" or "spotting". The electrode generally used is "Azucar 80".

The dirty top roller, as it is called, grips the bagasse better. Moreover, the Australians (Donnelly⁹) consider that the bagasse in the grooves prevents the flowback of juice at the bottom of the top roller grooves, to form a pool of juice in front of the top roller. Observation shows in any case that operation with the dirty top roller arrests this undesirable phenomenon. If a groove refuses to fill with bagasse, it permits a jet of juice to issue on the entry side of the mill, an effect which again supports clearly the Australian thesis. They consider equally that the packed bagasse also prevents the passage of juice on the delivery side, a phenomenon much more difficult to observe, but much more serious since the juice is thus reabsorbed by the bagasse leaving the mill. Donnelly has obtained at Pioneer, with a dirty roller, moistures of the order of 42% at the last mill, sometimes as low as 39%, and this strongly supports his point of view.

The first claim above is definite; the second seems to us entirely probable. It is supported by the fact that the upper layer of bagasse leaving the mill always has a moisture higher than the average moisture of the whole bagasse, as has been often verified.

When operating with a dirty top roller, it is necessary for calculating the setting to take into account the volume occupied by the bagasse within the grooves, and in consequence to consider the top roller as equivalent to a smooth roller of the diameter at the top of the ridges.

However, if devices are in use for measuring or registering the lift of the top roller, this approximation is much too rough to give a valid correction of the values indicated or registered: the dirty top roller does not permit of an accurate estimate of the thickness of the bagasse filling its grooves, since this is too uncertain and too variable.

Effect of imbibition on mill feeding

Bagasse saturated with imbibition is less easily gripped by the mills than is dry bagasse. Hence it is a common tendency of the men operating the mills to reduce the imbibition when difficulties are experienced with feeding or with mill chokes; the control of imbibition should not be left in their hands, since this expedient is generally disastrous from the financial point of view, and they should not resort to it but seek out the cause of the feeding trouble.

On the other hand, bagasse saturated with hot water is more difficult to feed than that saturated with cold water. If both types of water are available, certain feeding troubles may be overcome, particularly at the last mill, by changing from hot to cold water, without diminishing the quantity of imbibition.

Mechanical feeding devices

Chokes are the nightmare of engineers in charge of mills. When a mill is working near its maximum capacity, a small fault is often sufficient to provoke troubles: hydraulic plunger sluggish, or sticking; trashplate too high, trashplate too low; toe of trashplate worn, the teeth a bad fit in the grooves of the front roller; insufficient space for the passage of juice behind the heel of the trashplate; poor settings of feed and delivery openings of the mill; wear of one roller, or of the three rollers; major variations of fibre in cane, etc., etc. The multiplicity of possible causes complicates the task of locating the exact cause of the trouble, and sometimes several weeks may pass before the trouble can be effectively remedied.

On the other hand, without being able to blame any item of plant, it is sometimes impossible to apply the desired imbibition in a mill working at high capacity. Consequently, engineers are turning more and more to forced feeding. The latter consists of using a special apparatus to force the bagasse to enter the mill. Such equipment is generally of rotary type, and termed feeding roller.

Rotary feeders

(a) *Roller on the bagasse.* This is the device most widely used with intermediate carriers of slat type; it is located above the feed plate and driven from the top roller of the mill, sometimes through an intermediate shaft near the head of the carrier (Fig. 8.18). The best system is one where its shaft and the two radius rods which connect it to the driving shaft form a rigid framework which can oscillate about the centre line of the drive shaft, which is placed sufficiently high for the roller to be easily lifted by an increase in the thickness of the bagasse layer. In this way it rests on the bagasse, which it compresses slightly while lifting if the bagasse layer becomes thicker. The surface of the feeding roller is grooved longitudinally, or fitted with small angle-pieces. Its external diameter is generally smaller than that of the mill rollers and equal to two-thirds, three-quarters or four-fifths of the latter. The sprockets driving it should be calculated so that its peripheral speed v' is greater than that of the mill rollers. v :

$$v' = 1.1 - 1.7 v \quad (8.12)$$

In the case of an apron carrier, this speed v' should be equal to that of the carrier (cf. eqn. (8.4)). The feed roller above the bagasse, though popular for some time, is now generally abandoned in favour of the following:

(b) *Under-feed roller.* Here the feeding rollers are located below the bagasse layer (Fig. 8.19). These are given a diameter approximately one-third to two-thirds that of the mill rollers. It is of advantage to make the diameter of this magnitude, as the surface of contact and the

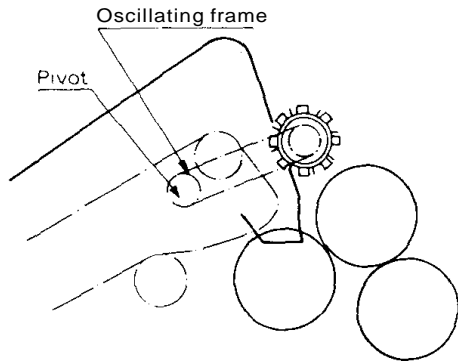


Fig. 8.18. Feeding roller above the bagasse.

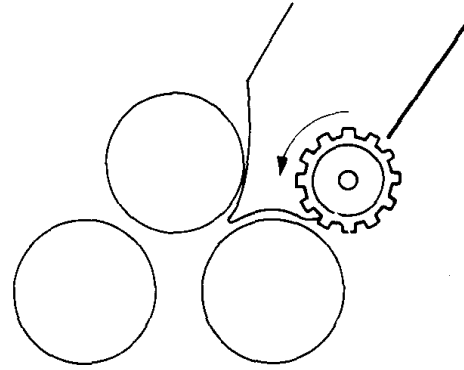


Fig. 8.19. Feeding roller below the bagasse.

grip of the roller, and hence the compaction effect on the bagasse, are improved accordingly. This effect can be improved by providing the roller with grooving, for example of 10–13 mm ($\frac{3}{8}$ – $\frac{1}{2}$ in.) pitch.

It is sometimes provided with a small trashplate connecting it to the feed roller; however, this is sometimes not easily adjusted, and may be omitted, thus permitting the roller to be set closer to the feed roller of the mill. The feeding roller will be arranged for a peripheral speed v' slightly higher than that of the mill rollers, v :

$$v' = 1.08 - 1.10 v \quad (8.13)$$

The setting c should be as small as possible as long as the bagasse does not overflow above the top roller. With good mill settings, this condition is obtained when $c =$ about 6 times the delivery opening e_A of the mill while operating¹⁰ (cf. p. 139). If this delivery operating, for example, is 25 mm (1 in.), the clearance between mean diameters of the feeding roller and the top roller would be approximately 15 cm (6 in.).

Van Hengel¹¹ gives $c = 7 e_A$ for a first mill, dropping to $c = 5 e_A$ for a last mill.

To permit of adjustment, the manufacturer designs the equipment to permit of a horizontal displacement of the roller of about 12.5–15 cm (5–6 in.) ($C' - C''$).

The clearance allowed between the feeding roller and the feed roller of the mill is generally 5 mm ($\frac{1}{4}$ in.). It should be as small as possible. The firm of Cail insist that the centre line of the feeding roller should be as low as possible relative to that of the top roller of the mill, and state that feeding would become defective if it were above that of the mill roller.

A properly set under-feed roller often permits of a reduction in ratio of setting of front and back rollers. This ratio for example may be reduced from 2 to 1.8. Other things being equal, such reduction always means an improvement in extraction.

Power. The feeding roller requires 2.5–7.5 kW (3–9 h.p.), generally 5 kW (6 h.p.).

Gain in capacity. The system improves the capacity of the mill by 10–20%. Any mill working at high capacity or near its limit should be fitted with an under-feed roller, which is more effective than the roller above the bagasse.

Continuous pressure feeder

This device, developed in Australia by the Colonial Sugar Refining Company, is widely used in Australia and South Africa. It was developed as a means of overcoming the feeding difficulties experienced with fine bagasse and hot maceration. It consists (Fig. 8.20) of a pair

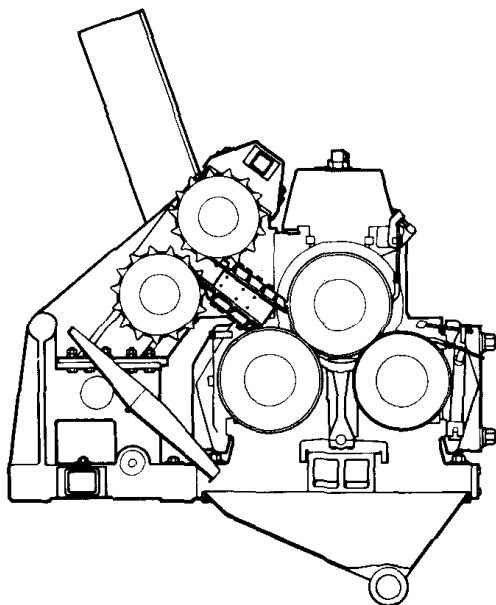


Fig. 8.20. Pressure feeder.

of rollers ahead of the mill, with an enclosed chute, diverging slightly, which gives the bagasse a positive movement from these rollers to the feed opening of the mill. The pressure feeder is driven by the mill engine, generally by means of an additional spur wheel and tail bar. The feeder rolls extract a high proportion of thin juice, thus reducing the excess moisture in the sloppy feed, and presenting the mill rollers with a comparatively dry feed supplied under pressure. The action of the feeder is continuous and uniform, unlike the intermittent action of pushers. It is much more forceful than the slight compression exerted by ordinary feeding rollers. Thus it gives a definite improvement in crushing rate or in lower moisture content of bagasse, or both. Hollywood¹² reports an increase in crushing rate from 77.1 to 81.1 t.c.h. (about 5%) with a decrease in final bagasse moisture from 49.6 to 46.0%, where pushers were replaced by a pressure feeder on the final mill. It is considered in Australia that the pressure feeder is the only device enabling high capacity to be maintained with heavy imbibition.

Pressure feeders are generally installed at the last mill, sometimes on intermediate mills and even on the first mill. One installation in Australia¹³ has given a first bagasse moisture below 52%.

1. Diameter. The roller diameter varies from 60 to 100% of the diameter of the mill rollers, being generally about 80% of that figure.

2. Surface. The surface of these rollers may be smooth, but preferably grooved; it may

be provided with chevrons. Both upper and lower rollers are furnished with Messchaert grooves (*cf.* p. 107).

3. Arrangement. The axial plane of the two rollers is generally at 45° to the horizontal. The centre line of the enclosed feed chute between pressure feeder rollers and mill is as closely as possible normal to the axial plane of these rollers and to that of the upper and feed rollers of the mill.

This chute is lined on the inside with a sheet of stainless steel, in order to reduce the friction of the bagasse and to facilitate its movement, while reducing power requirements and wear. The pressure developed in this chute is of the order of $0.07 - 0.10 \text{ kg/cm}^2$ ($1 - 1\frac{1}{2} \text{ lb./sq.in.}$) but may sometimes attain 1.0 kg/cm^2 (15 lb./sq.in.). The entry end of the chute should be located on a generatrix of the rollers at $15 - 20^\circ$ from their axial plane.

The upper and lower plates should diverge by at least 3.5° , or essentially 7%¹⁴. Murry and Shann^{14a} recommend 1 : 16 or 6.25%.

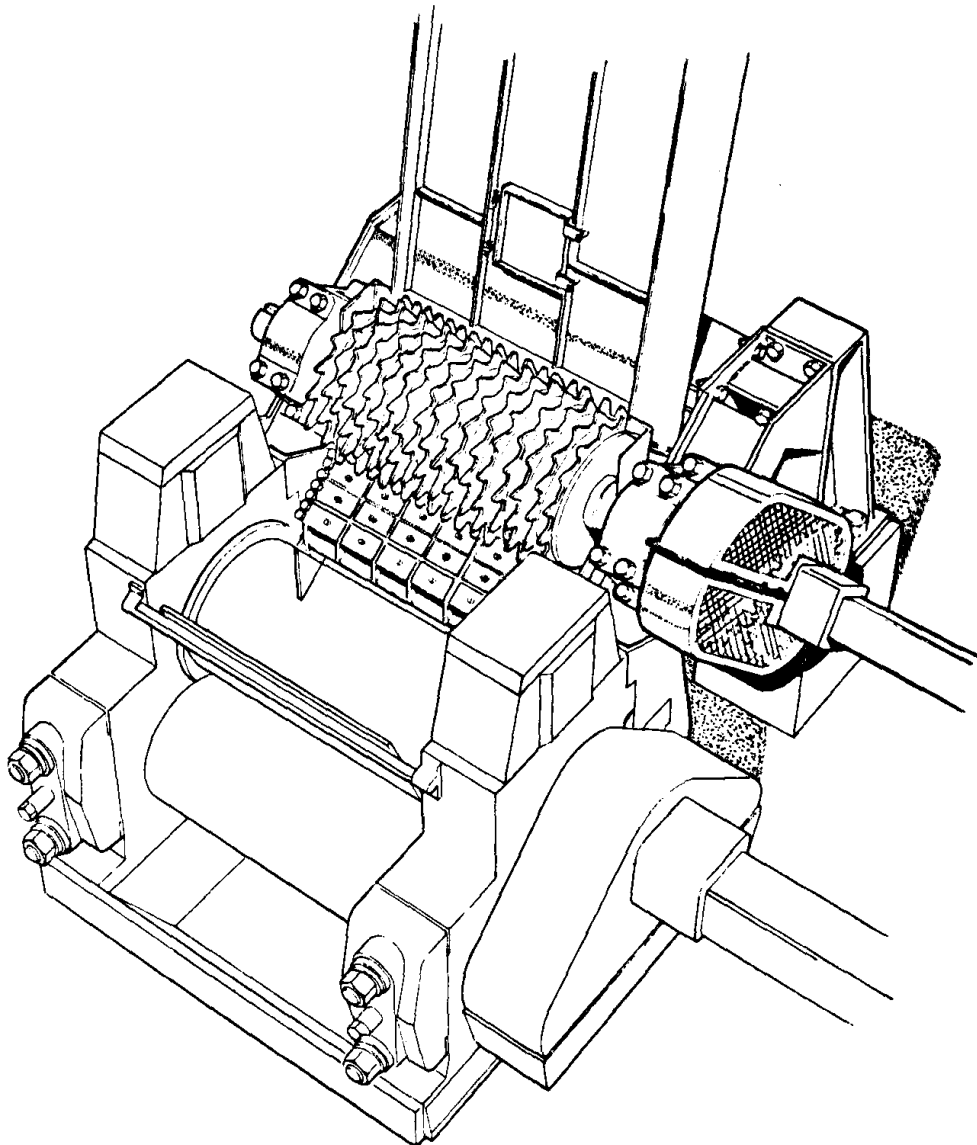


Fig. 8.21. Pressure feeder (Fletcher and Stewart).

The chute should meet the top and delivery rollers on a generatrix, making an angle not greater than 40° with their axial plane.

The compaction coefficient at the discharge end of the chute, *i.e.* at the entry to the mill, should be at least 80–95 or preferably 130–145 kg fibre/m³ (5–6 or 8-9 lb./cu.ft.). It has almost the same value at the entry to the chute. We may take for example:

	Coefficient of compaction			
	Entry	Discharge	Entry	Discharge
1st mill	104 kg/m ³	96 kg/m ³	6.5 lb./cu.ft.	6.0 lb./cu.ft.
2nd mill	112 kg/m ³	104 kg/m ³	7.0 lb./cu.ft.	6.5 lb./cu.ft.
3rd mill	120 kg/m ³	112 kg/m ³	7.5 lb./cu.ft.	7.0 lb./cu.ft.
4th mill	128 kg/m ³	120 kg/m ³	8.0 lb./cu.ft.	7.5 lb./cu.ft.
5th mill	136 kg/m ³	128 kg/m ³	8.5 lb./cu.ft.	8.0 lb./cu.ft.

It is recommended^{14a} that coarse grooving should be used on both rollers, for example, 50-mm (2-in.) pitch and with an angle of 34-35°.

4. Speed. The speed of the rollers must bear a fixed ratio to that of the mill rollers. This is easily arranged, since they are driven by a train of gears from the same prime mover as the mill.

Normally, these rollers are given a peripheral speed of 1.5 times that of the mill rollers. In Australia¹⁵ this ratio varies from 1.3 to 1.7.

5. Setting. Donnelly⁶ recommends that the opening between pressure feeder rollers should be set so as to give a generated volume (generally known as "escribed volume") of 1.5 times that of the feed and top rollers of the mill. If the mill has a ratio of settings (when operating) of 2, this will then give an overall ratio of 3 between the pressure feeder and the delivery opening of the mill (we may comment that, if the roller speed is 1.5 times that of the mill, the pressure feeder opening will be equal to the feed opening of the mill). With this setting, Donnelly has recorded bagasse moistures of 41.4% as an average, sometimes as low as 39.8%.

The moisture of the bagasse in the pressure chute should be 3 per unit of fibre, according to Crawford¹⁶, who recommends a pressure feeder setting such that the volume escribed by the rollers will be given by:

$$V = 3.8 Af \quad (V = 136 Af) \quad (8.14)$$

V = escribed volume of the pressure feeder rollers, in m³/h (cu.ft./h)

A = weight of cane treated, in t.c.h.

f = fibre content per unit of cane.

The object (or result) of the pressure feed rollers is thus to increase the compaction coefficient (*cf.* eqn. (10.20)) from the 80-95 kg/m³ (5-6 lb./cu.ft.) of a Donnelly chute to a higher figure in the pressure feed chute, increasing progressively from about 95 (6) for a first mill to 120, 130, 145 and even 160 (7.5, 8, 9, 10) for a final mill.²²

This corresponds to a weight of fibre per unit escribed volume ("fibre index") of 265 kg/m³ (16.5 lb./cu.ft.) for the pressure feeder, with approximately 450 (28) at the feed opening of the mill, and 880 (55) for the delivery opening.

In any case, the ratios of escribed volume of 1.5 and 3 as recommended by Donnelly should not exceed maximum values of 2 and 4 respectively, or there will be a risk of breakage of the pressure chute. The present trend is even to reduce these two ratios to 1.3–1.4 and 1.7–1.8 respectively.

6. Power consumption. Pressure feeders demand a fairly high power, which Scriven¹⁷ estimates at 35–45 kW (40-60 h.p.) for a mill of 1 x 2 m (36 x 72 in.).

7. Relative power consumption by pressure feeders and mill. Tests at Sezela²³ in South Africa give the following results:

Torque consumed by feeding rollers:

5 of total torque (rollers plus mill) in first mill
 12 of total torque (rollers plus mill) in fifth mill

The power developed was:

105 kW (140 h.p.) for feed rollers and 516 kW (690 h.p.) for the first mill
 25 kW (33.5 h.p.) for feed rollers and 291 kW (390 h.p.) for the fifth mill

8. Application. Pressure feeders are expensive and are often difficult to fit into the tandem, but they give an improvement in capacity and extraction, particularly where it is desired to use high imbibition at high temperatures. It should be mentioned, however, that in South Africa¹⁸ their results have sometimes been disappointing. On the other hand, the popularity of this method of feeding in Australia is such that there is practically no mill in that country without pressure feeders. Such a result can only speak strongly in its favour. The mean extraction of Australian mill tandems is slightly inferior to that of South African tandems, but it is obtained with an appreciably smaller number of mills per tandem (4–6 as against 6–7).

Following Walkers and Fletcher²⁴, Smith and Mirrlees have fabricated in cooperation with Hulett's some 6-roller mills designed to reduce the moisture of diffusion megasse from 85% to 48% (performance guaranteed).

9. Imbibition. The pol percent of the juice from the pressure feed rollers is appreciably lower than the pol of residual juice in the bagasse from the preceding mill. Under these conditions, it is of interest to use the pressure feeder juice as imbibition of the bagasse from the preceding mill (Fig. 8.22), and to use the imbibition normally used at that point, at a point further back in the train.²⁶

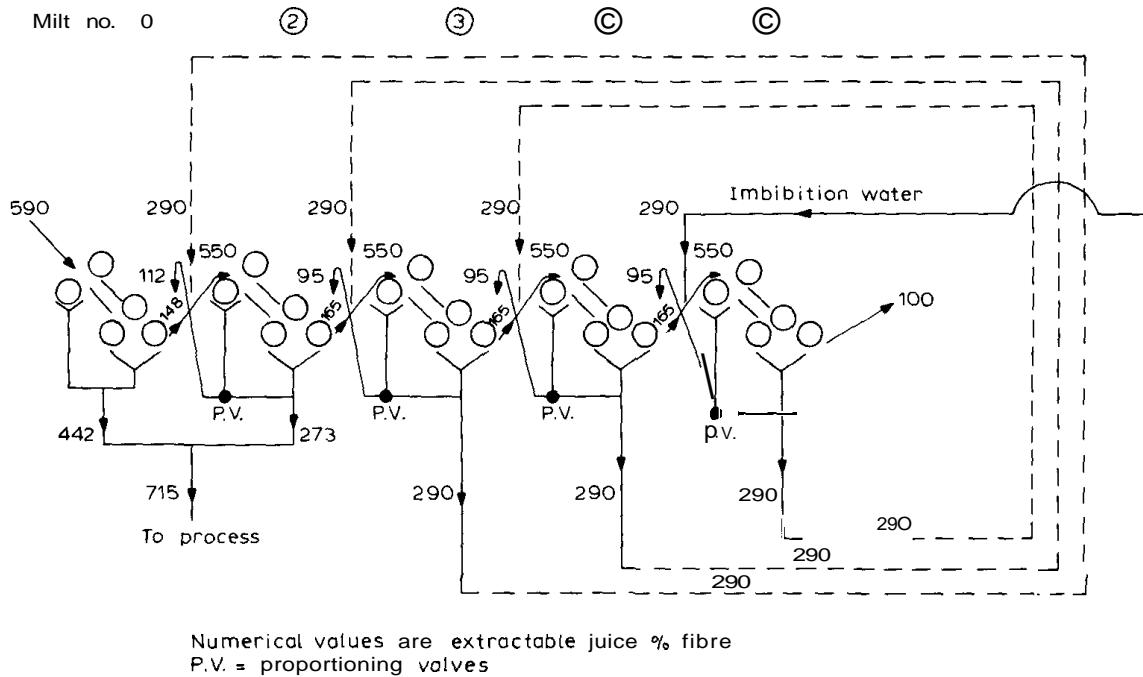


Fig. 8.22. Pressure feeder. Imbibition circuit.

BAGASSE CONVEYORS

The bagasse leaving the last mill should be elevated for distribution to the boiler furnaces. It is picked up by a bagasse elevator, which drops it onto a horizontal conveyor; this distributes it along the length of the boiler station to the furnaces (Fig. 8.23). If the last mill and the bagasse loft are nearly in a straight line, the same conveyor can serve at the same time as elevator and conveyor (Fig. 8.24). These conveyors are generally of the scraper type. When they are not also acting as distributors, they can well be of belt type.

For apparent density of the bagasse transported one can take 160 kg/m^3 (10 lb./cu.ft.).

Speed. They are generally driven at a speed equal to two or three times the peripheral speed of the last mill, or about 20–30 m/min (60–100 ft./min).

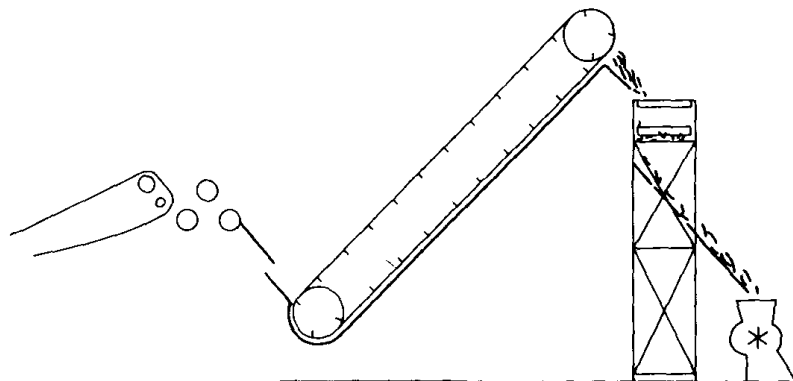


Fig. 8.23. Bagasse elevator.

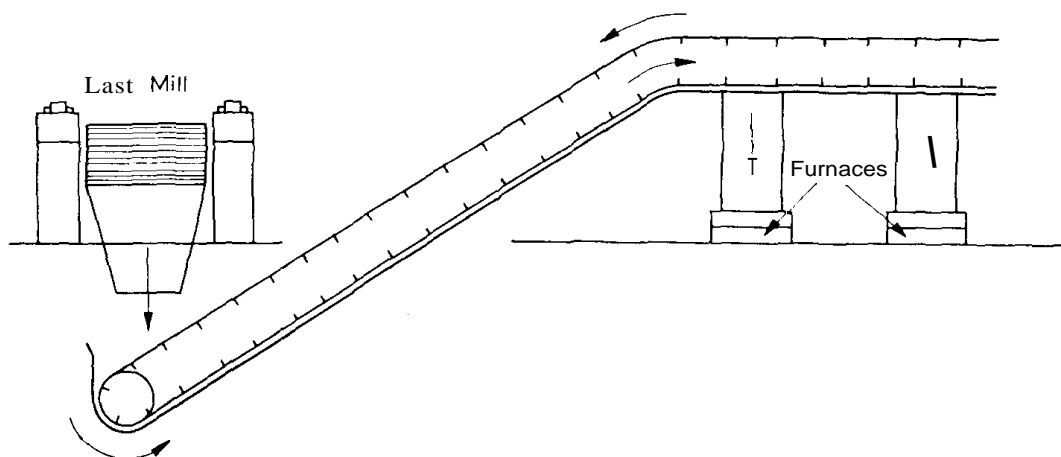


Fig. 8.24. Single conveyor for elevating and distributing bagasse.

In South Africa²⁵ a speed of 30-36 m/min (100-120 ft./min) is recommended for scraper conveyors and 90 m/min (295 ft./min) for belt conveyors, but this can reach 140 m/min (460 ft./min).

Belt conveyors can operate at 25° slope, 22½° being recommended. For scraper conveyors, 50° is appropriate.

Width. For transport on rubber conveyors, the following values are recommended:

- 25 – 50 t/h of bagasse: 6 t/h per 100 mm width (18 t/h per ft.)
- 50 – 100 t/h of bagasse: 7 t/h per 100 mm width (21 t/h per ft.)
- 100 t/h of bagasse: 8 t/h per 100 mm width (24 t/h per ft.)

For scraper conveyors:

6.5 t/h per 100 mm (20 t/h per ft.)

For scraper conveyors, the cost is higher than that for belt conveyors by a ratio of 1.5 or 2.0.

Power. For a factory of average size, an average figure may be taken of 1 kW for every 5 m total length of bagasse conveyor (*i.e.* about twice the actual length of the carrier = upper run + lower run) (1 h.p. for every 20 ft.).

COEFFICIENT OF FRICTION FOR CANE AND BAGASSE

Tests carried out in Queensland to determine this coefficient have given¹⁹:

	<i>Polished metal</i>	<i>Unpolished metal</i>
Cast iron	0.40	0.44-0.50
Steel	0.38	0.40-0.50

This coefficient²⁰ depends mainly on the grooving. The influence of moisture is less than would be supposed. The grooving, however, is more effective with lower moistures and with higher pressures.

Fineness of preparation has a slight favourable influence.

The coefficient of friction increases at first with pressure, passes through a maximum for a rather low pressure, then decreases in proportion as the pressure increases.

It decreases uniformly with speed, and Australian tests have led to the suggested formula:

$$\phi = 0.43 - 0.007 v \quad (\phi = 0.43 - 0.002 v) \quad (8.15)$$

ϕ = coefficient of friction of the cane or bagasse on the metal

v = peripheral speed of the roller, in m/min (ft./min).

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9. Roller Grooving

OBJECT

Originally, mill rollers were smooth. However, the capacity of a mill with smooth rollers is much less than that of a mill of the same dimensions and the same speed, but with grooved rollers. Further, the grooved rollers break up the bagasse more completely, and thus facilitate the extraction of the juice by the following mills.

Hence, ungrooved rollers are no longer made.

CIRCUMFERENTIAL GROOVES

The universal type of grooving is the circumferential groove. It is formed by grooving the roller with notches describing complete circles, in planes perpendicular to its axis. These are distributed regularly over the length of the roller, and a longitudinal section thus gives a serrated outline (Fig. 9.1).

The section of these grooves forms an isosceles triangle with an apical angle of about $55-60^\circ$. To avoid leaving sharp and fragile edges, the exterior point of the triangles is cut off, in such a way as to form a landing or flat of width equal to about $\frac{1}{10}$ of the pitch. In the same way, a landing of the same width is left at the bottom of the groove (Fig. 9.2).

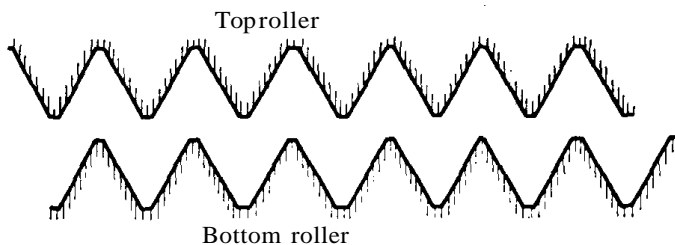


Fig. 9.1. Circumferential grooving.

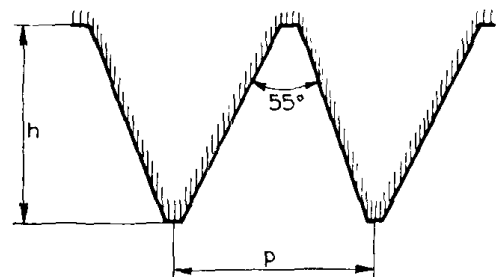


Fig. 9.2. Detail of grooves.

The grooving is specified by its depth h and its pitch p . It is described by these two figures, quoted in the order given: "grooving of 10 x 13 mm", for example, which means: grooving of 10 mm ($\frac{3}{8}$ in.) deep and 13 mm ($\frac{1}{2}$ in.) pitch. The pitch is the distance which separates the axes of two successive teeth. The depth, or height, is the distance from the top to the bottom of the teeth, measured perpendicular to the axis of the roller.

The grooving of the three rollers of a mill is arranged in such a way that the teeth of the top roller mesh with the grooves of the two bottom rollers.

Dimensions of grooving

Many different sizes of grooving are used. They are distinguished by the angle of the groove and by the pitch and depth.

The angles most often used are 45° , 50° and 55° . When it is desired to run the rollers without cleaning the bagasse from them (*cf.* p. 106) the angle may be as low as 30 or 35° . French manufacturers have adopted 50° .

The most frequent dimensions are:

(a) Angle of 50° ; pitch of 25, 30, 40, 45, 50, 60 and 80 mm (1 - 3 in.). The depth is thus approximately equal to the pitch: 22.5 x 25, 28 x 30, 39 x 40, 44 x 45, 49 x 50, 60 x 60 and 81.5 x 80 mm. Thus we often speak of 40 x 40 instead of 39 x 40, for example. The pitches of 60 or 80 mm (2 $\frac{1}{2}$ and 3 $\frac{1}{8}$ in.) are in general reserved for first mill-crushers.

(b) Angle of 55° ; 20 x 26, 40 x 52 mm (1 and 2 in.). Final mills were often of smaller pitch than the earlier mills in the train. For example:

First mill grooving:	38 X 50 mm (1 $\frac{1}{2}$ x 2 in.)
Second mill:	20 x 26 mm ($\frac{3}{4}$ x 1 in.)
Third and following:	10 x 13 mm ($\frac{1}{2}$ x $\frac{1}{2}$ in.)

The extraction was considered better with mills of fine grooving. Grooving even of 5 x 6.5 mm ($\frac{1}{4}$ x $\frac{1}{4}$ in.) pitch has been used in Hawaii. However, it is difficult to verify the assertion of better extraction, and the present trend is to reduce the number of different groovings in a tandem, so as to require a smaller number of spare rollers and give improved interchangeability. Thus a single grooving is generally adopted at least from the second to the final mill.

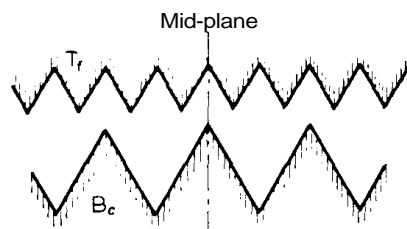


Fig. 9.3.

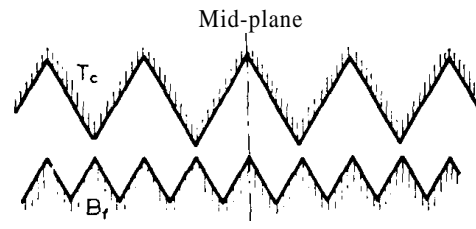


Fig. 9.4.

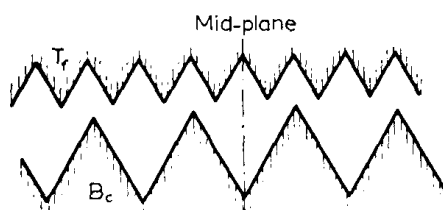


Fig. 9.5.

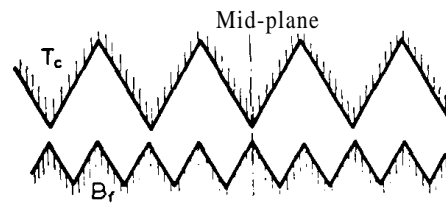


Fig. 9.6.

Fig. 9.3 – 9.6. Meshing of rollers with grooving of multiple dimensions.

Meshing of rollers with grooving of multiple dimensions

Unfortunately, the meshing of a roller with one having grooves of double the pitch is not as simple as might be imagined.

Let us consider rollers of two different pitches, one coarse (20 x 26 mm, for example) and one fine (10 x 13 mm, for example).

(7) If, in the bottom roller B_c with coarse grooving, a tooth occurs in the median plane, it will be necessary that the top roller T_f with fine grooving, should have a groove in the median plane (Fig. 9.3). The roller T_f then must have an even number of teeth, since there must be the same number on each side of the median plane, which is a plane of symmetry.

Consequently, a bottom roller B_f with fine grooving, which could mesh with the roller T_f , cannot mesh with a top roller T_c corresponding to B_c (Fig. 9.4); on the contrary, the teeth will come exactly opposite each other.

(2) If, in the bottom roller B_c with coarse grooving, a groove comes in the median plane, it will be seen (Fig. 9.5) that it will still be necessary for a groove to fall in the median plane of T_f , and that the rollers B_c and T_c corresponding respectively to T_f and B_c still will not mesh (Fig. 9.6).

In other words, for any roller, and especially for a bottom roller of coarse grooving B_c , considerations of symmetry require that the median plane will pass either through a ridge (Fig. 9.3), or through a groove (Fig. 9.5). In both cases, as may be seen from the figures, the roller of fine grooving T_f meshing with B_c should have a groove in the middle; all the ridges of T_f will then correspond with the middle of the flank of the grooves of B_c .

Hence the rules:

Rule 1. In order that a top roller of fine grooving should mesh with a bottom roller of double pitch, it is necessary that the top roller of fine pitch have a groove in its median plane, and consequently an even number of teeth. (Conversely, the condition that a top roller with coarse grooving should mesh with bottom rollers of half its pitch is that the top roller with fine grooving corresponding to the latter should have a tooth at its mid-plane, and consequently an odd number of teeth.)

Rule 2. To allow two rollers to mesh when one has a pitch double that of the other, it is necessary to choose from two solutions which are incompatible with each other:

- (a) T_f should mesh with B_c (and then T_c will not mesh with B_f) or
- (b) T_c should mesh with B_f (and then T_f will not mesh with B_c).

The advantages and disadvantages are strictly symmetrical. From the point of view of replacing rollers, for example, and if, normally, rollers of the same grooving are used in the same mill:

- | | |
|------------------------------------|--------------------------------------|
| (a) If T_f meshes with B_c : | (b) If T_c meshes with B_f : |
| T_c may be replaced by T_f | T_c may not be replaced by T_f |
| B_c may not be replaced by B_f | B_c may be replaced by B^* |
| T_f may not be replaced by T_c | T_f may be replaced by T_c |
| B_f may be replaced by B_c | B_f may not be replaced by B_c . |

We shall see, however (p. 105), that combination (a) of Fig. 9.3 is the only interesting one, as combination (b) is never met with in the sugar factory. Accordingly, it is preferable that T_f should mesh with B_c . In other words, the grooving should always be arranged so that the top rollers have a groove in the middle, *i.e.* have an even number of teeth.

If necessary, the sets of rollers having the coarsest grooving of the tandem could be an exception to this rule (since there will be no rollers of double pitch with which they would have to mesh). But, not knowing what may be done in the future, one would always be wiser to apply the rule just enunciated to all the rollers of the tandem.

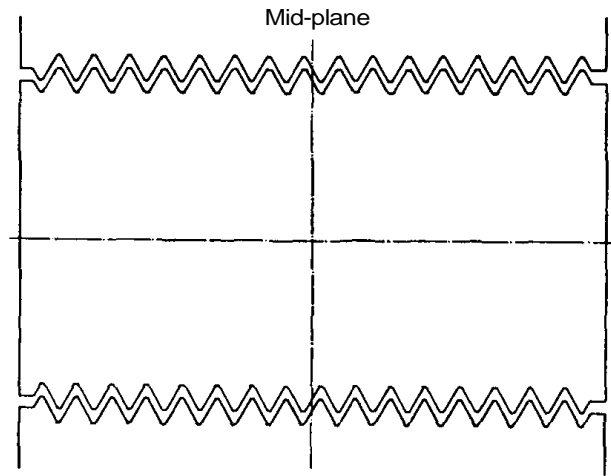


Fig. 9.7. Universal rollers.

Universal rollers

Rollers termed "universal" are also manufactured which may be used equally well as top or bottom rollers (Fig. 9.7).

With these, the idea of symmetry relative to the median plane is abandoned. Their grooving is devised in such a manner that it is the middle of the flank of a tooth which falls in the midplane of the roller. Consequently, a top roller becomes a bottom roller, or *vice versa*, by simple reversal end for end. This system is favoured in Argentina. It offers the advantage that, if necessary, one roller is sufficient for replacement and provision for breakdowns. But it has the disadvantages that:

(7) It is useful only for the chosen dimension of grooving. If, for example, it is 20 x 26 mm ($\frac{3}{4}$ x 1 in.), the grooving of 10 x 13 mm ($\frac{3}{8}$ x 7 in.) which would mesh with the 20 x 26 mm universal roller would be a symmetrical grooving (as may be readily seen from Fig. 9.3) with an odd number of teeth, and so would not be able to mesh with a smaller grooving of 5 x 6.5 mm ($\frac{3}{16}$ x $\frac{1}{4}$ in.). A grooving of 40 x 52 mm ($1\frac{1}{2}$ x 2 in.) meshing with the 20 x 26 mm universal grooving would be entirely **unsymmetrical**; the median plane would pass one-quarter of the way down the side of a tooth. It would be necessary then to abandon the idea of interchangeability of the 20 x 26 mm and the 40 x 52 mm grooving. Alternatively, the universal grooving could be applied to the 40 x 52 mm, and it would then be necessary to abandon interchangeability of 20 x 26 and 10 x 13 mm. We lose in interchangeability the advantage gained by the universal roller in its own grooving.

(2) In order to profit from the advantage offered, it is necessary to construct the roller with the shaft the same at each end, so that the free end carries a pinion landing and a squared end, the same as the driving end. Such a roller is more expensive and is awkward in appearance.

This disadvantage is less marked if rollers with alternate pinions are employed (*cf.* p. 167).

(3) In case of a breakdown, it may be necessary to change the pinion - an operation long and sometimes difficult, and liable to cause delay.

The universal roller is not attractive and is applicable only to tandems utilising only one pitch of grooving in a given mill; the advantage applies only to the one pitch.

Variation of grooving along the tandem

The size of the grooves generally decreased from the first to the last mill. There are two principal reasons for this:

(7) The bagasse becomes finer and finer as it proceeds from the crusher to the last mill. It was logical to proportion the grooving to the size of the bagasse particles which it was designed to handle.

(2) One of the objects of grooving is to break up the bagasse to facilitate the extraction of juice. It is in the early mills, therefore, that the greatest disintegrating effect must be sought (hence the coarsest grooving), in order that the later mills should profit from it.

This variation in grooving along the tandem has the disadvantage of leading to rollers of different surfaces, a disadvantage which is tempered by the adoption of grooving with one pitch a multiple of the others. In this way certain rollers may, in emergency, be replaced by others of double or half the pitch, as we have just seen.

To avoid breaking the uniformity of the tandem, certain factories, in Cuba, Queensland and Egypt for example, tended to adopt a standard grooving by continuing through to the last mill the coarse grooving adopted for the early mills. The difference in extraction is not appreciable if it can be arranged that the rollers of the last two mills are always new or in good condition. This solution was thus not strictly contra-indicated. It has the advantage, moreover, of removing one of the causes of choking of the later mills (poorer feeding with fine grooving), and it is now general.

Disposition of grooving. As example of previous practice in the disposition of grooving along the length of the tandem is given in Table 9.1 for the principal types of tandem.

Double pitch grooving in feed roller

Table 9.1 assumes that the three rollers of each mill have grooving of the same dimensions. However, it is sometimes advantageous to group in the one mill rollers with grooving of different dimensions, provided that they mesh with each other; and the only rational solution is to have the grooving of one roller meshing with grooving of double the pitch.

It is necessary moreover that the top roller and the delivery roller should have the same grooving, otherwise certain parts of the bagasse will not be subject to adequate pressure, resulting in some loss of extraction. For the same reason, these two rollers should have the smaller grooving. Under these conditions, there is only one combination possible; the feed roller must have a grooving double that of the other two.

This was the grooving adopted in our factories. Table 9.2 for example gives the grooving of Savanna in Reunion, the tandem then consisting of six mills of 725 X 1,370 mm (28 X 54 in.) preceded by two sets of knives and a shredder. This tandem has been replaced by one of 860 X 1,700 mm (34 x 67 in.) as the first mill with following mills of 810 x 1,675 mm (32 X 66 in.). The first mill grooving is 60 X 60 mm ($2\frac{3}{8}$ x $2\frac{3}{8}$ in.), the second 40 X 52 mm ($1\frac{1}{2}$ X 2 in.), the third 40 X 52 mm ($1\frac{1}{2}$ in.) for the feed roller and 20 X 26 mm ($\frac{1}{4}$ x 1 in.) for top and delivery. The later mills have grooving of 20 x 26 mm ($\frac{3}{4}$ x 1 in.) at all rollers.

This system had the advantage of assisting mill feeding owing to the space left free at the feed roller (Fig. 9.3), while maintaining a full pressure at the delivery. It allows for and accentuates the role assigned to each roller, as the feed roller has primarily the object of forcing the bagasse to enter the mill, and the delivery roller that of extracting as much juice as possible.

TABLE 9.1

SIZES OF GROOVING IN DIFFERENT TANDEM (in mm)

	<i>Crusher</i>	<i>1st Mill</i>	<i>2nd Mill</i>	<i>3rd Mill</i>	<i>4th Mill</i>	<i>5th Mill</i>
12-roller tandem	40 x 52	20 x 26	10 x 13	10 x 13		
14-roller tandem	40 x 52	20 x 26	20 x 26	10 x 13	10 x 13	
15-roller tandem	40 X 52	20 x 26	20 x 26	10 x 13	10 x 13	
18-roller tandem	40 x 52	20 x 26	20 x 26	10 x 13	10 x 13	10 X 13

TABLE 9.2

PITCH OF GROOVING (in mm), SAVANNA (mills 725 x 1,370 mm)

	<i>1st Mill</i>	<i>2nd Mill</i>	<i>3rd Mill</i>	<i>4th Mill</i>	<i>5th Mill</i>	<i>6th Mill</i>
Feed roller	57 x 52	37 x 52	57 x 52	28.5 x 26	28.5 x 26	28.5 x 26
Top roller	57 x 52	28.5 X 26	28.5 X 26	14.25 x 13	14.25 x 13	14.25 x 13
Delivery roller	57 x 52	28.5 x 26	28.5 x 26	14.25 x 13	14.25 x 13	14.25 x 13

Angle of grooving

Wear and breakages of the teeth increase when the angle of grooving is decreased, but to a smaller extent than was supposed until 1955. When there is no reason for adopting a more acute angle, a value of about 55° is employed. American factories sometimes adopt $45 - 50^\circ$, a value which may be recommended. However, it is the Australians who in about 1956 — 58 adopted the greatest innovations in this matter. While they had already long used an angle of 35° , they found, when aiming to allow the grooving of the top rollers to fill with bagasse ("dirty top roller"), that the angle of the grooving had a marked influence on the packing of bagasse in the roller grooving. At the feed roller, Donnelly¹ specifies 30° , this angle being sufficiently acute to prevent the bagasse from penetrating to the bottom of the grooves, under the low pressure at the feed roller; the groove thus acts as its own Messchaert groove for draining the juice.

For the top roller, *he* recommends $30 - 35^\circ$, with the object of allowing the grooves to fill with bagasse. For this it is necessary to place the top roller scraper at 75 mm (3 in.) (for a first mill) to 40 mm ($1\frac{1}{2}$ in.) (last mill) from the surface of the roller; or, better, to replace it by a straight plate, fixed at 6.5 – 20 mm ($\frac{1}{4} - \frac{3}{4}$ in.) from the tips of the teeth of the roller.

If it is not desired to allow the top roller to pack with bagasse, Donnelly recommends the adoption of a very wide angle, $45 - 60^\circ$, to ensure that the bagasse will pack to the bottom of the groove under pressure, so that the juice cannot flow back towards the entry to the mill and thus be lost as far as extraction is concerned.

For the delivery roller, he recommends 45° , since a more acute angle will hold the bagasse too strongly and the scraper will not be able to remove it.

These considerations are summarised, with our recommendations, in Table 9.3.

One may ask what is the best method for meshing of grooving of different angles. We consider that the best solution is to adopt the same height and the same pitch (or multiples or sub-multiples of these) and to vary the width of the flats. Take, for example, a mill with top roller of grooving 28.5 X 26 mm ($1\frac{1}{8}$ x 1 in.) and feed roller of double the pitch. Geometrical considerations show that meshing presents no problem (Fig. 9.8). It is necessary only to avoid the setting described as "metal to metal" and to allow always, in the setting of the mill at rest, at least 0.75 mm ($\frac{1}{32}$ in.) between the closest parts of the two rollers; this corresponds to 6.5 mm ($\frac{1}{4}$ in.) between the tops of the teeth and the bottom of the grooving, and to about 4.75 mm ($\frac{3}{16}$ in.) of average thickness of bagasse layer between the two rollers.

TABLE 9.3

Roller	ANGLE OF GROOVING (degrees)					
	Top roller dirty			Top roller clean		
	Feed	Top	Delivery	Feed	Top	Delivery
Normal values				50	50	50
Values recommended by Donnelly	30	35	45	30	50	45
If disregarding the Messchaert effect at feed roller	45	35	45	45	45	45
Recommended values	42	35	42	50	50	50

MESSCHAERT GROOVES

Messchaert grooves, or "Messchaerts" (also known as "juice grooves"), were invented and applied in Hawaii. They have only one objective: improvement of the extraction. They unquestionably render good service from this point of view, and their adoption today is almost universal.

At the feed roller, it is in the zone *AB* (Fig. 9.9) that the bagasse undergoes the maximum pressure, and the juice is extracted. But this juice has only two ways of escape: forward with the roller, beyond the trashplate, or backward in the direction of the point C. In either direction, it encounters a thick layer of bagasse, already compressed, moving in the same direction as the juice at *BD*, but in the opposite direction at *AC*.

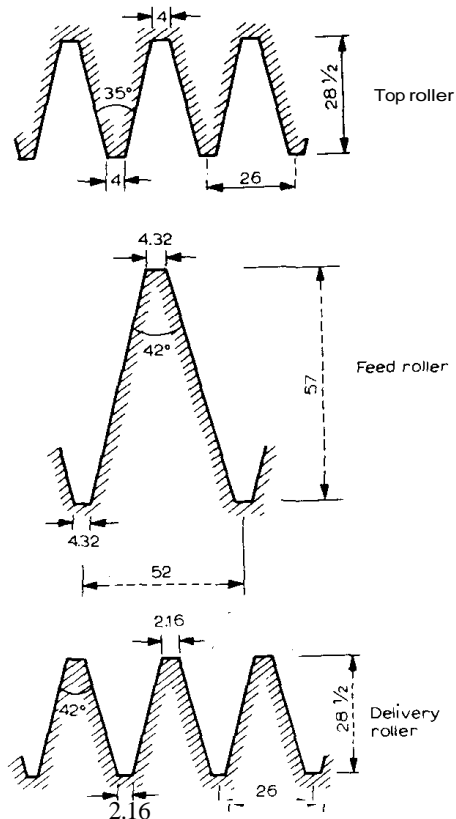


Fig. 9.8. Meshing of rollers of different groove angles (dimensions in mm).

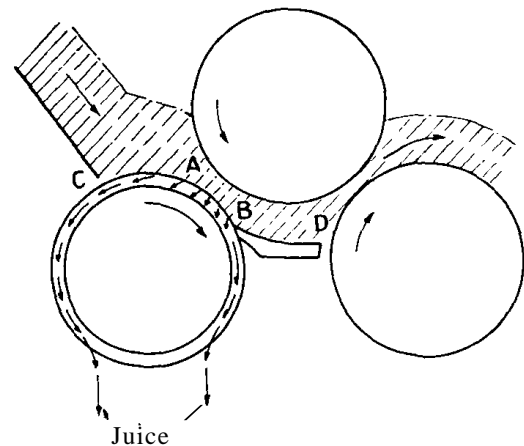


Fig. 9.9. Action of Messchaerts at the feed roller.

Messchaert had the simple but fruitful idea of providing an outlet for the juice by cutting circumferential grooves around the feed roller at intervals, by means of which, at all points in the zone *AB*, the juice can find an immediate way out, and can drain without hindrance from any part of the roller.

We may comment that the juice which flows from *B* comes mainly from the region *A*, since at *B* the point of maximal pressure has already been passed and the juice has in principle already been extracted; the portion of juice which remains is absorbed by the bagasse as it re-expands.

The same problem is encountered at the delivery roller, but here it is less serious, since gravity assists the juice to escape from the zone of high pressure, whereas at the feed roller it has to climb over the roller to pass through the layer *AC*.

Messchaerts may be provided in two ways:

- (1) By eliminating a tooth and locating the Messchaert in the position of the tooth eliminated (Fig. 9.10)
- (2) By leaving all the teeth, and cutting the Messchaert between two teeth (Fig. 9.11).

The latter system has the advantage that it does not lose a tooth, and consequently avoids creating a zone of lower pressure in the layer of compressed bagasse. On the other hand, lateral drainage of the juice is not so easy, and the teeth next to the Messchaerts have a greater tendency to fracture along the line *ab*. Thus the second system is reserved for coarse

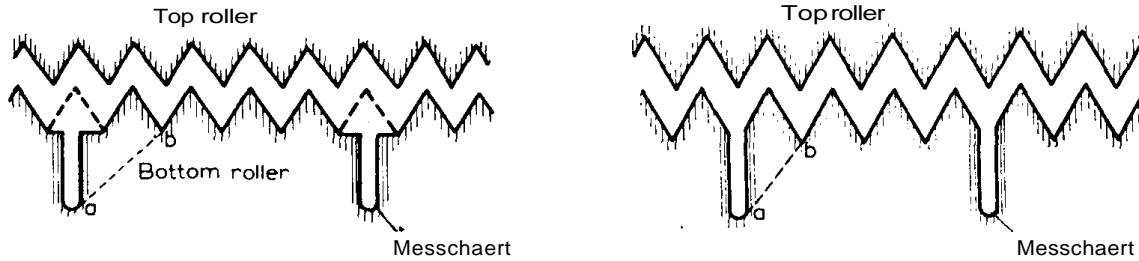


Fig. 9.10. Formation of a Messchaert by *elimination* of a tooth.

Fig. 9.11. Formation of a Messchaert without elimination of a tooth.

grooving in which the width of the tooth compensates for the proximity of the Messchaert (*cf.* Figs. 9.10 and 9.11).

Dimensions of Messchaerts

In the feed roller, the Messchaerts are given a width of 3 - 6.5 mm ($\frac{1}{8}$ - $\frac{1}{4}$ in.), generally 5 mm ($\frac{3}{16}$ in.). There is no advantage to be obtained by making them wider; the pressure succeeds in forcing bagasse into Messchaerts which are too wide, and drainage is less effective.

The depth of the Messchaerts depends on the quantity of juice which they have to handle. A depth of at least 20 mm ($\frac{3}{4}$ in.) is necessary, measured from the base of the grooves. In general, a depth of 25 mm (1 in.) is provided (Fig. 9.12). However, with very high crushing rates and heavy imbibition, when the Messchaerts are widely spaced, this depth is not sufficient to drain the juice from the feed roller. It is then necessary to make the Messchaerts either more numerous, or deeper. In Hawaii they are taken to a depth of 40 - 50 mm ($1\frac{1}{2}$ - 2 in.).

The flow of juice in the Messchaerts depends on their pitch, which is the distance between successive Messchaerts. This must obviously be a multiple of the pitch of the main grooving. For grooving of 13 mm ($\frac{1}{2}$ in.) pitch Messchaerts of 52 or 78 mm (2 or 3 in.) pitch are adopted, or 52 - 104 mm (2 - 4 in.) pitch for main grooving of 52 mm (2 in.) pitch.

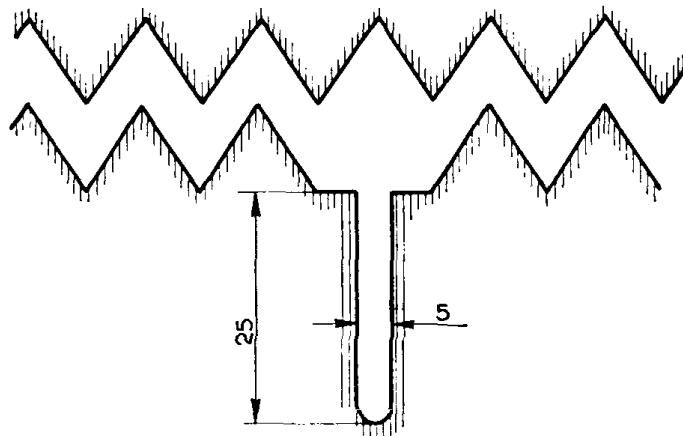


Fig. 9.12. Dimensions of Messchaerts (mm).

Pitch and symmetry of Messchaerts

As the pitch of the Messchaerts must necessarily be a multiple of that of the main grooving, we shall have 1, 2, . . . , 6 teeth between adjacent Messchaerts. With medium or coarse grooving working at high tonnages, it may be of advantage to have a Messchaert groove every two grooves rather than one every three grooves. However, this is not always possible if symmetry is to be maintained. Messchaerts are applicable only for bottom rollers, and we have seen (p. 103) that top rollers should preferably have a groove in the middle; hence the bottom rollers of the same grooving must have a tooth in the middle. Thus, for a symmetrical arrangement of Messchaerts relative to the median plane of the roller, they should have between them an odd number of teeth: 1, 3, 5, etc., as is easily seen by considering a bottom-roller grooving such as that of Fig. 9.4 or 9.6. An interval of one tooth, *i.e.* a Messchaert every groove, is always possible, though it may be desirable only with certain coarse groovings, for example, in the feed roller of the first mill; but it may often be found that an interval of three teeth is too much and that it is necessary to have a Messchaert every two teeth. In this case there are only two possible solutions:

- (1) To accept a non-symmetrical arrangement of the Messchaerts, which has moreover no disadvantage for performance but increases the risk of errors in the workshop, or
- (2) To locate the Messchaerts so that the median interval is 3 teeth, and all the others 2 teeth. This is the better solution.

This problem arises only for an interval of 2 teeth. For an interval of 4, there is scarcely any disadvantage in adopting 3 or 5 and thus retaining an odd-number spacing.

Delivery roller

The above figures for depth and pitch are for feed rollers. When Messchaerts are placed in delivery rollers, the depth does not exceed 20 mm (f in.), because:

- (1) This would not be necessary on account of the low output of juice
- (2) The pressure is much higher at the delivery roller: the risk of breakage of the teeth next to the Messchaerts would be much greater, and the consequent damage would be correspondingly more serious if the Messchaerts were made deeper.

In the same way, and for the same reasons, a narrower width is adopted, 3 mm ($\frac{1}{8}$ in.) instead of 5 mm ($\frac{3}{16}$ in.), and a substantially greater pitch. For grooving of the family 10 x 13 mm (l x 5 in.), the pitch varies from 78 mm (3 in.) to 130 mm (5 in.).

An attempt to standardise the rollers by making the Messchaerts the same for feed and delivery rollers would have more disadvantages than otherwise. The difference between the two rollers is unfortunate but necessary.

Scrapers

If suitable precautions were not taken, the Messchaerts would become filled with bagasse and would soon cease to be effective. It is thus necessary to install below the rollers special scrapers or "combs". These consist of steel bars, carrying, opposite each Messchaert, an angle-iron, one wing of which is extended to the bottom of the Messchaert (Fig. 9.13).

The length of these angle-irons or "fingers" is calculated so that the scraping edge makes an angle of about 30° with the roller surface.

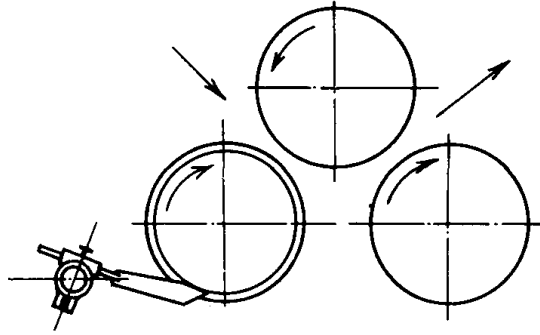


Fig. 9.13. Messchaert scraper.

Wear. It is important that the scraping fingers be firmly attached to the supporting bar, and set so as to pass accurately in the centre of the Messchaerts. Otherwise they will rub against the sides of the grooves, wear rapidly, enlarge the Messchaerts and completely destroy their efficiency.

Advantages and disadvantages

Rollers provided with Messchaerts suffer more damage from tramp iron passing through the mill, and wear more rapidly. But they are of great benefit at the feed roller:

(7) They improve the capacity of the mill by permitting it to deal with a quantity of juice which otherwise would cause choking. When the mill is not working at its maximum capacity, they render possible the use of a quantity of imbibition which otherwise would be prohibitive.

(2) They especially improve the extraction by augmenting the proportion of juice removed at the feed roller, thus relieving and facilitating the task of the delivery roller.

At the delivery roller, on the other hand, their usefulness is more debatable. The advantages are less, because there is less juice and it flows away more readily. The disadvantages are more serious, because the pressure is much greater, and because it is difficult to devise a satisfactory method for cleaning the Messchaerts. Furthermore, in the case of Messchaerts formed by elimination of one tooth (Fig. 9.10), the bagasse alongside the Messchaerts, at the delivery, is not compressed: its moisture and pol content are certainly higher than those of the bagasse coming from the full grooves. At the last mill especially, such loss cannot be recovered.

It may be added that the experiments of Bullock² may well be interpreted as indicating that Messchaert grooves in the delivery roller are not effective.

To sum up, while opinions are almost unanimous on the value of Messchaerts in the feed roller, opinions are divided as regards the delivery side. Our opinion is that Messchaerts in the delivery roller involve a definite increase in expense, while offering only a doubtful gain, or perhaps a loss. We do not recommend them.

CHEVRONS

Unlike Messchaerts, which have as their special object the improvement of extraction, chevrons have the sole object of improving the feeding of the bagasse.

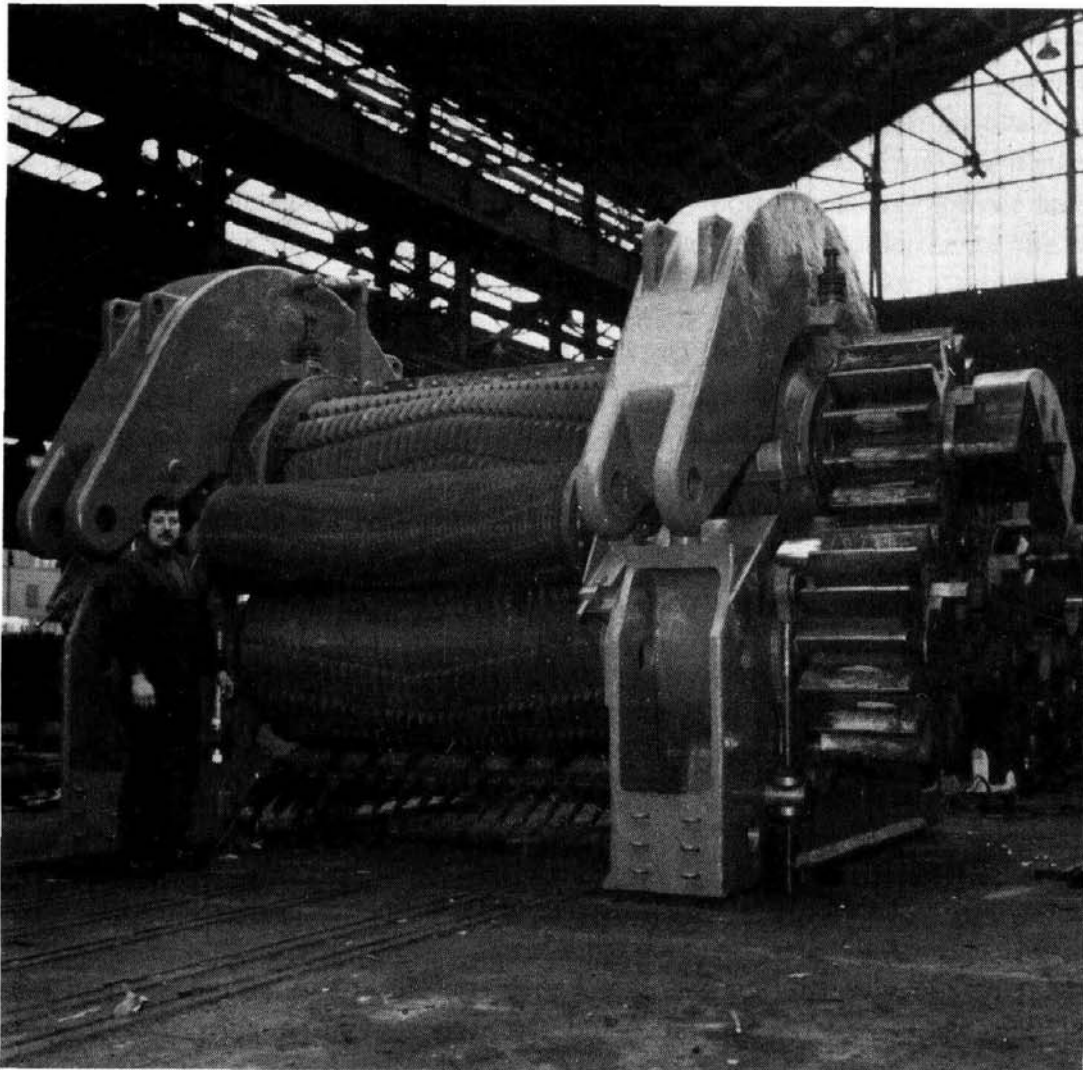


Fig. 9.14. Chevrons. Feeder roller. Messchaert scraper (hydraulic plungers removed).

These are notches (Fig. 9.14) cut in the teeth, in a sequence which describes a helix from the centre to one end of the roller, the other half of the roller carrying a helix symmetrical with the first with respect to the median plane of the roller. This is exactly the same design as that on the Fulton crushers, and uses the same form of notch (*cf.* Fig. 5.4). The same remarks on the depth and dispositions of the chevrons apply to mills (*cf.* p. 53).

The angle of the chevrons with a longitudinal line on the roller varies from 10 to 20° and is generally 18°. The pitch is of the order of 20 cm (8 in.).

Chevrons are placed only on the two feeding rollers: the top and feed rollers. They are never placed on the delivery rollers, because:

- (1) They would not be of any use, the delivery roller being in a way force-fed by the trashplate and the feed roller
- (2) The bagasse near the chevrons would not be adequately compressed.

For the latter reason, we consider that it is equally advantageous to omit the chevrons from

the top roller, the pressure at the delivery opening being exerted as much by the top roller as by the delivery roller.

Where slip is liable to occur between the roller surface and the bagasse, an acceptable compromise would be to cut chevrons spaced 25 cm (10 in.) apart (or better $\frac{1}{10}$ th of the circumference, *i.e.* 10 chevrons on the roller) and of depth equal to one-third the depth of the main grooving.

Moreover, chevrons are useful on the feed roller only in the case where its grooving is the same as that of the top roller. If the system described previously (Fig. 9.3) is adopted, they generally become superfluous; if not, the same compromise can be adopted in all cases as for the top roller. At the feed roller, moreover, chevrons have the disadvantage of increasing the proportion of the bagasse passing into the juice, on account of the small quantity which lodges in the chevrons and which is not scraped out by the toe of the trashplate.

Conclusion

In certain cases, chevrons are of real assistance to the grip of the rollers on the bagasse. They are not indispensable, except when approaching the extreme capacity of the mill, and if they are adopted it is preferable to restrict them to the feed roller and to reduce their depth to one-third the depth of the grooving.

KAY GROOVING

This system, devised by Kay, engineer at Aguirre, Puerto Rico, consists of cutting, in the top roller, helicoidal grooves of rectangular section, of a depth greater than that of the circumferential grooving (Figs. 9.15 and 9.16).

These grooves become filled with bagasse and the pads of bagasse so formed facilitate the feeding of the entering bagasse, without detriment to the continuity of the pressure. However, they have a tendency to lift the scraper plate and render it ineffective.

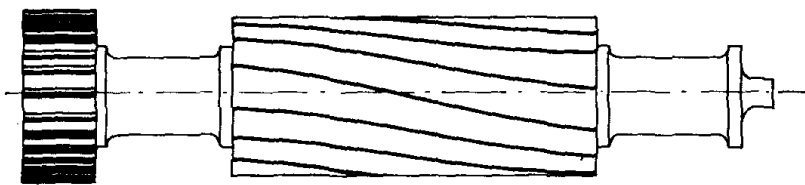


Fig. 9.15. Kay grooving.

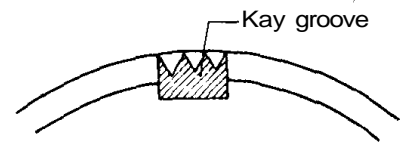


Fig. 9.16. Transverse section of roller with Kay grooving.

WEAR OF ROLLERS

Causes of wear

Wear of the rollers occurs owing to four causes:

- (1) Corrosion of the metal by the acid juice
- (2) Friction of scrapers and trashplate

- (3) Friction of the cane or the bagasse, which always slips on the roller to some extent
- (4) Passage of tramp iron, which crushes the metal or breaks the teeth
- (5) Necessity for turning the roller down during the slack season, after 2 or 3 campaigns, to restore its proper cylindrical form.

The wear is generally greater at the centre than at the ends. Turning the shell also permits of reduction of irregularities due to broken teeth.

Extent of wear

In mills of standard design, the maximum wear permissible in a roller is approximately 4–5%. In other words, a roller of 1 m in mean diameter when new has to be discarded when its diameter has been reduced to 950–960 mm. In mills of more modern design, wear may be allowed to go somewhat beyond 5%, exceptionally to 10% with suitable design.

In Java, from the viewpoint of mill performance, the diameters of different rollers of the same mill were not permitted to vary between themselves by more than 3%. When the wear of certain rollers of a tandem exceeded this figure, they were re-assembled together in the one mill. If one mill of the tandem thus had to work with smaller rollers than the others, it should not be the first, nor the last, nor, if possible (that is, in tandems of more than 11 rollers), the second. Generally, then, it was the third mill.

If the 3 rollers of a mill have different mean diameters, they should, as far as possible, be arranged: the smallest as feed roller, the largest as delivery, and the intermediate one as top roller.

This point of view, however, is contested by Mackay, in Australia, who asserts³ that it is the top roller which should have the lowest peripheral speed. He even goes as far as to recommend giving it if necessary a pinion having one tooth more than those of the lower rollers, so as to give it a peripheral speed of 75–150 mm (3–6 in.) lower per revolution. We have never tried such an unorthodox arrangement.

Tromp⁴ gives 6.5–13 mm ($\frac{1}{4}$ – $\frac{1}{2}$ in.) on diameter as the amount of wear per 100,000 tonnes of cane treated, or a life of about 3 years.

In South Africa, wear is reckoned at 1.5 mm ($\frac{1}{16}$ in.) of metal (3 mm ($\frac{1}{8}$ in.) on diameter) for:

16,000 t.c. for the top roller
25,000 t.c. for the feed roller
15,000 t.c. for the delivery roller

Actually, large rollers in the course of their lives handle a much greater tonnage of cane than small rollers, and it is more logical to reckon the life of a roller in hours of work. The life of 3 years indicated above would correspond fairly closely to average campaigns of 100–120 days of 24 hours for rollers of all dimensions. Rollers running at the highest speeds are evidently those which wear most rapidly.

Crusher rollers have a longer life, generally 9–12 years.

In Java, the extent of wear is reckoned as follows, in thickness of metal removed per 100 days of continuous crushing:

Rollers with ordinary grooving:	1.8 mm (0.07 in.)
Rollers with Messchaerts:	2 mm (0.08 in.)
Increased width of Messchaerts:	1 mm (0.04 in.)

It would be more accurate to distinguish between the feed roller and the two others, since it wears less rapidly.

The following are some measured values: (a) at Puunene in Hawaii⁵, (b) at Ambilobe in Madagascar, (c) at la Mare and Savanna in Reunion (mean over 5 seasons 1975 – 79), with the wear expressed as percentage of that of the top roller:

<i>Roller</i>	<i>Puunene</i> (per month)	<i>Ambilobe</i> (per 1000 h)	<i>Réunion</i> (per 100,000 t.c.)	<i>Prop.</i> <i>FCB</i>
Feed	0.53 mm (54%)	3.7 mm (71%)	2.1 mm (68%)	60%
Top	0.98 mm (100%)	5.2 mm	3.1 mm	100%
Delivery	0.85 mm (87%)	4.9 mm (94%)	2.9 mm (92%)	90%

The values for Ambilobe correspond to 1,000 hours of operation. The last column gives the proportions accepted by Fives Cail – Babcock.

Reshelling of rollers

When a roller is worn, it may be re-shelled by breaking off the cast-iron shell and returning the shaft to the manufacturers, who return it fitted with a new shell. Many factories are today equipped themselves for carrying out this rather critical operation. It can be done only once or twice at the most, on account of wear of the journals, which in turn limits the life of the shaft. Some factories, however, build up the worn or damaged journals by welding, and restore them to the original diameter.

We may comment in passing that the shrink fit of the shell on the shaft should be approximately 1 per 1,000. In other words, if the diameter of the shaft is 0.5 m, the bore of the shell should be 0.5 mm smaller than the diameter, or 499.5 mm.

Roller shafts may be checked by ultrasonic tests, to detect those which are cracked or which have incipient cracks, and are liable to break in the course of the campaign. The crack is located and its approximate magnitude determined. A South African organisation visits the factories in Natal, Mauritius, Reunion and Madagascar to conduct such tests.

To break the shell, two main methods are employed:

- (1) Drill holes every 10 cm (4 in.) along the length of the roller, to 90% of the thickness of the roller shell. Place in three of these holes plugs of dynamite, and explode them simultaneously.
- (2) Drop a monkey or "skull-cracker" (an old tail-bar, for example) on the roller from a height of 5 -6 m (15 -20 ft.), by means of a crane.

When both shaft and shell are worn, a valuable part of an old roller may still be recovered by making from its material a roller for a smaller mill (Fig. 9.17). The necessary condition for this is that the profile of the small roller should fit within that of the large one, and that

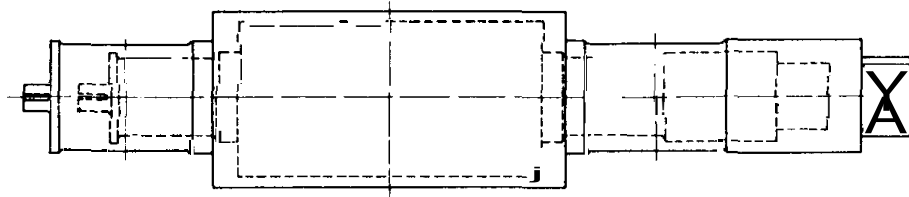


Fig. 9.17. Turning a roller from a larger roller.

after turning down there should remain a sufficient thickness of shell for strength of the roller, say at least 10% of the diameter below the bottom of the grooves. Rollers obtained in this way have a surface as hard as that of new rollers and a resistance to wear entirely comparable.

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- 2 K. J. BULLOCK, An investigation into the physical properties of sugar cane and bagasse. *Thesis*, University of Queensland, 1957, p. 329.
- 3 H. MACKAY, *Proc. 28th Conf. QSSCT, 1961*, p. 79.
- 4 L. A. TROMP, *Machinery and Equipment of the Sugar Cane Factory*, Norman Rodger, London, 1936, p. 186.

10. Pressures in Milling

Originally, the three rollers of a mill were fixed relative to each other, and their positions in the housing were adjusted at the start of the season or of the week with the aid of steel plates or wedges.

The pressure was then determined by the layer of bagasse: it increased when the tonnage was increased, and dropped when it was decreased. The milling results varied accordingly. The greatest disadvantage was associated with the passage of foreign bodies: if the piece of iron or steel was too big or too hard, the housing (or the roller) gave way.

This led to a search for some more flexible arrangement; hence the development of mills with pressure applied by springs (Fig. 10.1). This is the system still adopted today in very small mills.

HYDRAULIC PRESSURE

The use of hydraulic pressure was the next development; this had the advantage of maintaining a constant pressure, independent of the lift of the roller.

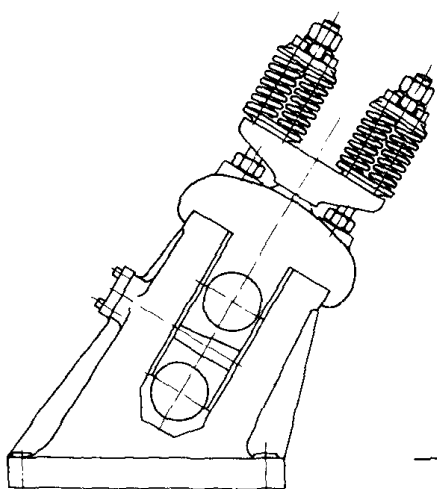


Fig. 10.1. Spring-loaded roller.

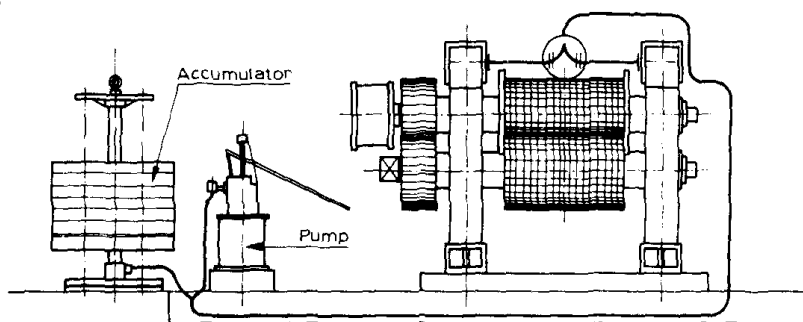


Fig. 10.2. Hydraulic pressure system.

The upper bearings of the top roller are arranged to slide in the gaps of the housing. They receive, with or without the interposition of an intermediate block, the pressure exerted by a hydraulic piston (Fig. 10.3) sliding in the cap. The hydraulic pressure is transmitted to the top of the piston by a system of piping containing oil under pressure (Fig. 10.2), the pressure being produced by an accumulator.

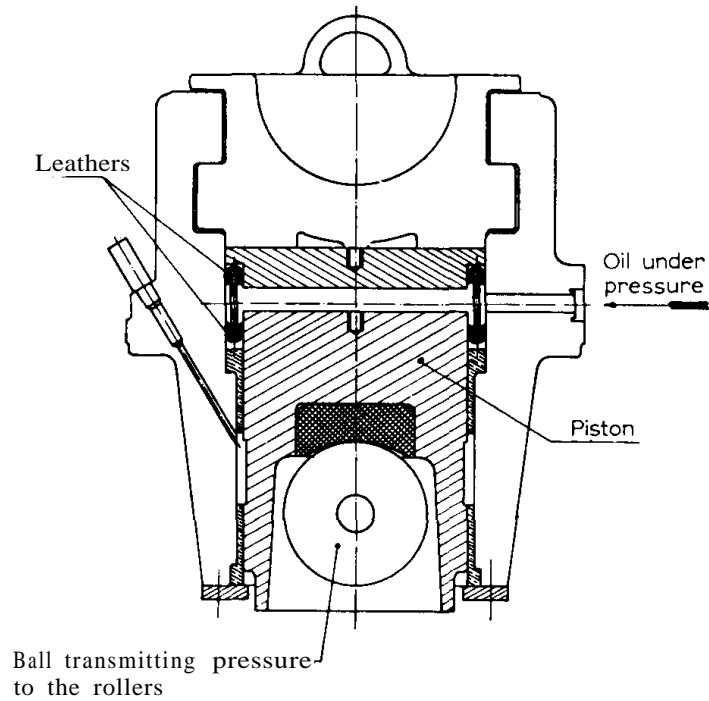


Fig. 10.3. Section of a hydraulic cap.

Accumulators

There are various types of accumulator, the principal ones being:

- (1) The deadweight accumulator (Fig. 10.2), now of historical interest only
- (2) The air - oil accumulator with air reservoir (little used - see second edition)
- (3) The individual air - oil accumulator.

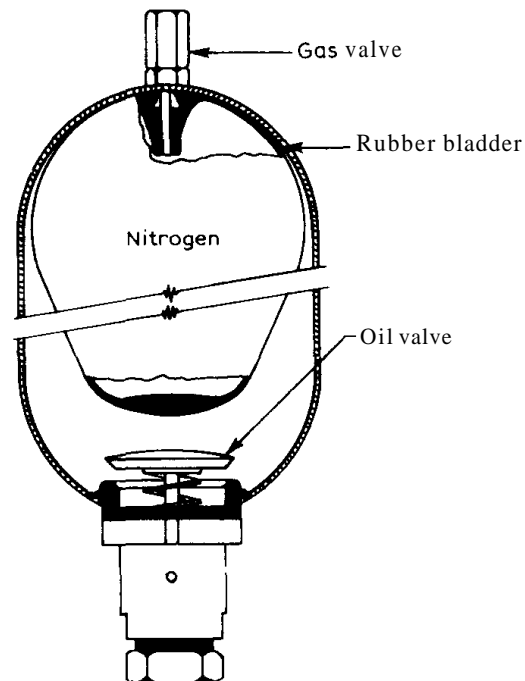


Fig. 10.4. Edwards accumulator with rubber bladder (Edwards Eng. Corp.).

Individual air – oil accumulators. The first accumulators using hydraulic pressure were placed alongside the mills, to which they were connected by piping. Edwards, in America, had the happy idea of placing the air chamber close to the mill cap, the accumulator consisting of a bladder filled with nitrogen which becomes compressed or dilated when the hydraulic ram rises or falls (Fig. 10.4). Several manufacturers have followed this example. Fives Lille – Cail deliver their mills with individual pressure bottles without a bladder, where the air is enclosed in the space at the top of the bottle, which is necessarily very airtight.

Oil is readily added to these bottles by means of a common oil pump placed close to the mill tandem; thus the pressure in them may be regulated at will; they are very free from leaks and rarely require recharging. They have, moreover, the advantage over the previous type of avoiding long and heavy hydraulic piping with a consequent pressure drop and the resulting slow response of the top roller. They thus act rapidly, are more flexible and also less expensive.

The interior diameter of the pressure bottles of Fives Lille - Cail is 20 cm (8 in.), those of Edwards are 22.5 cm (9 in.). They are then only approximately 1.2 – 2.5 times smaller than the hydraulic plunger of the mill, or 1.5 – 6 times in area. Their pistons, when they are provided, accordingly rise by 1.5 – 6 times as much as the roller.

Since the volume of the bottle is limited (of the order of 30 dm³ (1 cu.ft.)), the pressure

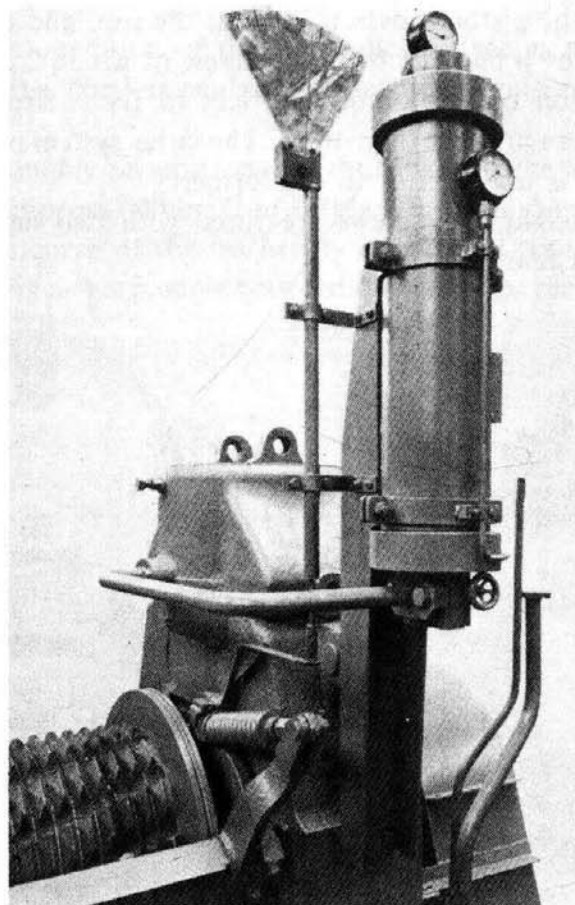


Fig. 10.5. Unilateral air – oil accumulator with lift indicator (Fives Cail – Babcock).

will accordingly increase somewhat as the roller lifts; this may be considered an advantage over other accumulators, where the pressure is constant or practically so.

Accumulators of this type are always provided with direct indicators of lift (Fig. 10.5), which add to the preceding advantages a complete control of the roller movements, and which, moreover, can be mounted on any type of mill, independently of the type of accumulator adopted. The use of these indicators can be strongly recommended. Their cost is insignificant and the services obtained are considerable.

These individual accumulators permit the pressure at each end of the roller to be regulated independently in such a manner that the lift is balanced and proportioned at each end, to the layer of bagasse which passes on that side. It often happens that one side of a mill, or of a tandem of mills, regularly receives a greater quantity of cane than the other side. This phenomenon is generally due to an *unsymmetrical* feeding of the carrier, which is frequently the case, for example, in factories possessing only one lateral table. In such a case, it is important to guard against aiming at an equal mean lift for the two sides: the side which is more lightly loaded should furnish a mean lift slightly less than that of the other. The two values so obtained for the lift on the left and the right are sometimes very different, and the higher pressure is not always found on the pinion side of the roller, as one would expect (*cf.* p. 167).

Caps and hydraulic pistons

When a hydraulic leather or packing fails, quick replacement should be possible. The cylindrical chamber in which the piston moves is open at the top, and closure is effected either by an ordinary screw, or by a bayonet fitting or screw of which 2 or 3 segments have been removed so that a rotation of 90° or 60° suffices to fix it firmly, as with the breech mechanism of much modern artillery (Fig. 10.6). The latter system is much quicker in operation, and its oil-tightness is not inferior to the former.

On account of their weight, the screws are fitted with two lugs which facilitate their unscrewing, handling and removal.

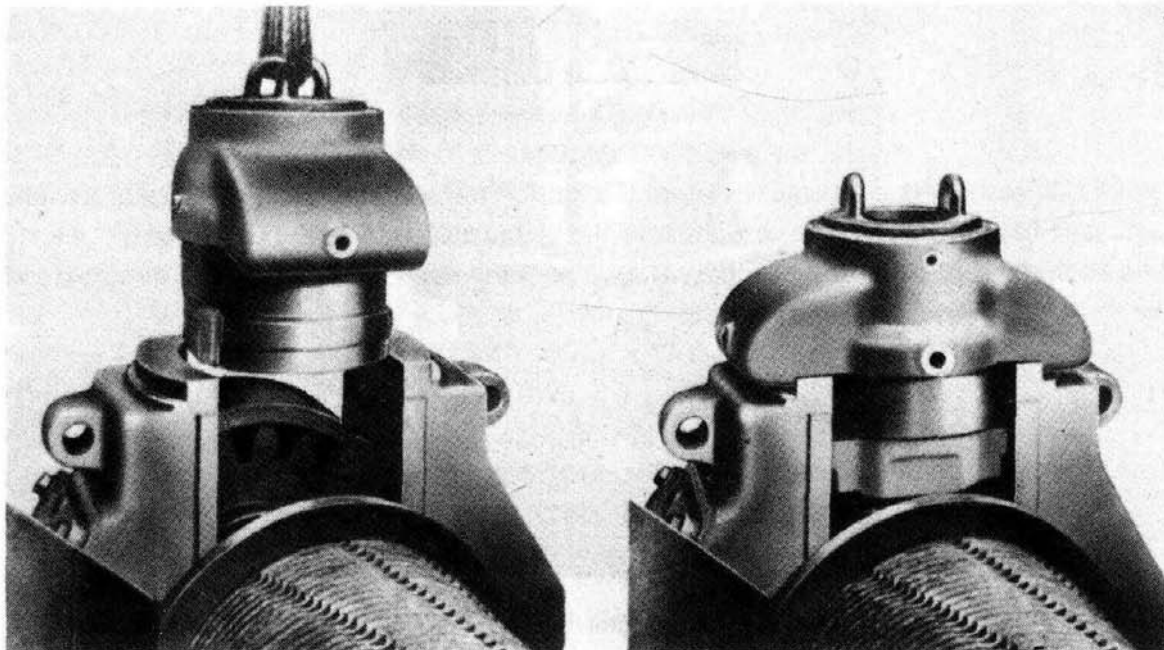


Fig. 10.6. Bayonet-type cap (Fives Cail - Babcock).

The hydraulic piston, sometimes called a "cheese" on account of its shape, is simply a cylinder with its edges rounded.

Leathers and packings. On account of the very high pressure prevailing in the space where the hydraulic pressure acts, the question of preventing serious leakage is important. For the piston, the question is complicated by the fact that it is mobile.

The problem is most often solved by closing the pressure chamber of the cap by a leather bent in the form of a U or preferably by a moulded metallo-plastic or synthetic rubber packing. One of these is placed against the screw cap and one at the upper portion of the piston (Fig. 10.3).

The pressed leather, working under high pressure, must be of very good quality. Mineral-tanned leathers last better than vegetable-tanned, even if the latter are oak-tanned.

Possible causes of poor behaviour of leathers. It happens sometimes that one of the leathers of a mill may "squirt" at any moment. If this trouble recurs frequently, it is most probably due to the absence of a flexible joint between the leather and the metal parts against which it presses. It is then necessary, either to interpose a rubber ring between the curve of the leather and the small metallic ring against which it is supported, or to place a wad of cotton in the inside of the leather, between the edge of the steel ring which fits inside the U and the bottom of the leather.

Friction. To make the operation of the hydraulics as free as possible, it is necessary to reduce to a minimum the friction caused by the movements of the piston and the accumulator.

Now, the friction is roughly proportional to the height of the leather (approx. 25 mm (1 in.)) or of the packing (approx. 20 mm ($\frac{3}{4}$ in.)) (Fig. 10.7). In the case of a leather, it is also beneficial to protect the curve of the leather by adopting a rounded form for the seating which receives it, avoiding a sharp angle between the piston or plate and the cylindrical wall of the chamber.



Fig. 10.7. Diametrical section of hydraulic leather.

The coefficient of friction f of the hydraulic packing against the walls has an approximate value:

$$f = (0.5 \text{ to } 0.6) \frac{h}{d} \quad (10.1)$$

f = coefficient of friction

h = height of leather, in mm (in.)

d = diameter of the hydraulic piston, in mm (in.).

The force necessary to overcome this friction is therefore:

$$\phi = fF \quad (10.2)$$

ϕ = force necessary to overcome the resistance of the hydraulic packing, in tonnes

F = hydraulic pressure acting on the piston, expressed in tonnes.

For the commonest dimensions, $f = 0.03 - 0.05$, and it will be seen that in these conditions, $\phi = 3 - 5\%$ of F . For a mill with its 2 packings receiving a total pressure of 300 tonnes, 9 - 15 tonnes are therefore required to "unstick" the pistons.

Maximum lift. The mill caps are generally constructed so as to permit of a maximum lift of the top roller of:

4-6 cm ($1\frac{1}{2} - 2\frac{1}{2}$ in.) for crushers

2-4 cm ($\frac{3}{4} - 1\frac{1}{2}$ in.) for mills.

Play in the operation of accumulators

There always exists some play or slack in the working of the hydraulic system, *i.e.* the top roller will always lift a certain amount before the pressure is fully effective on the bagasse; in the case of the old deadweight accumulators, before the accumulator begins to rise. In this case, the extent to which the roller rises from its stationary position, before the accumulator moves, gives a measure of this play.

This was one advantage of the deadweight accumulator, or the plunger-type air - oil accumulator, over the individual air - oil type; these indicated the instant when the bagasse is subjected to the full pressure, while the lift indicator shows the lift of the roller from its position of rest without showing whether the bagasse is receiving full pressure or not.

Causes of play. This play is due to several causes:

- (a) Play of the bearings in the housings
- (b) Mill caps badly adjusted
- (c) Adjusting-bolts of the delivery roller too short or fitted badly
- (d) Play between bolts and bolt-holes of the caps
- (e) Shafts badly adjusted.

Importance of play. Mills giving good extractions are always those in which the play is slight:

A play of 1 — 2 mm ($\frac{1}{32} - \frac{3}{32}$ in.) is normal

A value of 5-6 mm ($\frac{3}{16} - \frac{1}{4}$ in.) is excessive and harmful.

The existence of this play means that the bagasse is not submitted to the full pressure until the upper roller has lifted by an amount corresponding to the play: it has lifted by $A h_0$. The delivery opening accordingly is increased by $A e_0$, and we have in mills with normal housing (Fig. 13.4):

$$A e_0 = A h_0 \cos \frac{\alpha}{2}$$

a being the angle of construction of the mill (*cf.* p. 260). A e generally amounts to 3 - 6 mm ($\frac{1}{8}$ - $\frac{1}{4}$ in.), the highest values being found in older mills. It is thus necessary to be cautious of assuming that the full pressure is exerted as soon as the bagasse layer reaches a thickness equal to the delivery opening. The pressure depends only on the weight of the top roller and the items which it lifts with it (the coupling, part of the tail-bar, and in mills with fixed openings, the mill cap) until the moment when the opening attains e_0 .

True delivery opening of the rollers at rest

When the fabrication superintendent gives the "delivery setting", he indicates generally the measurement obtained by sliding a gauge, either between the two small cylindrical strips at the extremities of the rollers (*cf.* Fig. 13.2), or between the point of a tooth and the bottom of the groove with which it meshes, about one-quarter of the length along the roller, choosing for this an undamaged portion of the rollers.

If, for example, the thickest gauge which will pass between the rollers is 3 mm ($\frac{1}{8}$ in.), he will say, "My delivery setting is 3 mm".

It is very necessary to guard against reasoning as if the thickness of bagasse passing between the rollers when not lifting had the same value. This thickness is generally much greater than the "setting". The difference is due to a number of causes:

(a) Non-uniform wear of the rollers. If for example the setting has been taken between the cylindrical portions at the extremities of the rollers, the setting at the mid-point of the rollers will be higher, since the rollers wear more at the centre than at their extremities.

(b) Wear at the tips of the teeth.

(c) Destruction (more or less complete) of the teeth by the passage through the mill of pieces of iron or steel, breaking or crushing the teeth over part of their length and depth.

(d) The volume left free by chevrons and Messchaerts. As far as the latter are concerned, experiments made in Queensland have shown that it is suitable to take them into account, for the volume described by the opening, only to the extent of one-quarter of their depth (Bullock¹).

There are thus four measurements to be distinguished at the delivery of a mill:

- (1) The "delivery setting"
- (2) The true opening between the rollers when not lifting, taking into account the losses of metal (a), (b), (c), and the free volume (d)
- (3) The opening at commencement of lift of the accumulator
- (4) The delivery opening when operating.

The delivery setting is merely a theoretical figure, the only interest of which is that it is easy to measure. The true opening in the absence of lift corresponds to a true volume generated by the simultaneous rotation of the 2 rollers. When the mill is set in motion, it begins to fill with bagasse; the compression of the latter then increases to the point where it is sufficient to lift the weight of the top roller, or to force the bottom roller to the bottom of its bearings, or the bearings hard against the housing, etc. As the quantity of bagasse progressively increases, the various cases of play, (a) - (e) cited above, in the paragraph *Causes of play* (p. 122), come into operation successively, and the pressure increases; the volume generated thus attains the value corresponding to the opening at the commencement of lift; the pressure then increases more rapidly up to the limit corresponding to the commencement

of lift of the accumulator. Finally, the accumulator lifts, and the volume generated continues to increase in direct proportion to the lift, the pressure on the other hand remaining substantially constant.

We shall have for example:

Delivery setting:	1 mm (0.04 in.)
True opening in absence of lift:	4 mm (0.16 in.) (mean value)
Opening at commencement of lift:	6 mm (0.24 in.) (mean value)

In the case of a mill of 100 t.c.h. capacity, the delivery opening of which when working is 16 mm (0.64 in.) for example, the true opening in the absence of lift will be filled at a tonnage of less than 25 t.c.h. and the accumulator will not lift until the instantaneous rate corresponds to 50 t.c.h. Below this tonnage the pressure will not be fully effective.

Measurement of the true back roller opening in absence of roller lift. This determination is made very easily by the test with lead, on condition that a piece of lead is taken small enough to avoid displacing the rollers, for example, a piece 50 x 50 x 20 mm (2 x 2 X $\frac{3}{4}$ in.) will be suitable in general for mills of 914 x 1,828 mm (30 X 60 in.) or larger. This is made to pass through the mill while checking that neither the rollers nor the accumulator are displaced. The thickness of the piece of crushed lead is then determined in the following manner: it is placed on a piece of squared paper and its contour is drawn on the paper. The number of squares included on the paper is then counted, and the piece of lead is weighed. Let S be the area obtained for the piece and G its weight. We have:

$$e = \frac{G}{dS} \quad (10.3)$$

e = thickness sought, in cm (in.)

G = weight of lead, in g (lb.)

d = density of lead = 11.35 g/cm³ (0.41 lb./cu.in.)

S = area obtained, in cm² (sq.in.).

We thus obtain the mean thickness of the piece of lead with a high degree of precision. It corresponds to the true back opening between the rollers in the absence of roller lift. If the surface of the rollers is greatly damaged, the mean of several tests should be taken, allowing for the depth of a **Messchaert** groove as often as is necessary to represent its correct proportion of the surface of the roller.

This question of slack and of the mill openings at rest and at first application of pressure has been discussed at some length, but we consider that these details contribute greatly to a better understanding of what happens in the mill. Individual air – oil accumulators display a commencement of lift which does not correspond to the full pressure, on account of the causes of play indicated. Fortunately these are rather slight in modern well-constructed mills.

Maximum values

Hydraulic pressure. The accumulators, piping and mills are generally designed for oil pressures of 250 – 350 kg/cm² (3,500 – 5,000 p.s.i.). British firms have adopted as standard 280 kg/cm² (4,000 p.s.i.), while French manufacturers go up to 300-320 kg/cm² (4,250-4,500 p.s.i.).

This maximum, however, is not generally used, or is used only in the last mill. With the usual dimensions of hydraulic pistons, a value of 300 kg/cm² (4,250 p.s.i.) is sufficient to allow the optimal operating conditions to be attained, as we shall see later.

Total pressure. From the point of view of stresses in the roller material, the total pressure exerted on the top roller is generally measured by relating this pressure to the projected area (length X diameter) of the roller journals.

The pressure on the journals, so defined, generally does not exceed:

$$\begin{aligned} &70- 80 \text{ kg/cm}^2 (1,000 - 1,150 \text{ p.s.i.}) \text{ for crushers} \\ &100 - 110 \text{ kg/cm}^2 (1,400 - 1,600 \text{ p.s.i.}) \text{ for mills} \end{aligned} \quad (10.4)$$

Some manufacturers have gone up to 125 kg/cm² (1,800 p.s.i.) but, from the point of view of safety, and in order to maintain good lubrication, we consider it is not advisable to exceed 110 kg/cm² (1,600 p.s.i.).

PRESSURE CONSIDERED FROM THE OPERATING VIEWPOINT

Measure of the pressure on the bagasse

The indication of the total pressure exerted on the top roller gives little information as to the corresponding pressure undergone by the bagasse. If, for example, a pressure of 300 tonnes were applied to a mill of 1,097 x 2,560 mm (36 x 84 in.) it is evident that the effect on the bagasse will by no means be the same as if the 300 tonnes were applied to a mill of 914 X 1,828 mm (30 x 60 in.). In the larger mill, the total pressure will be distributed over a larger area of bagasse.

In the same way, the thickness of the bagasse layer has its effect.

In an attempt to take into account the dimensions of the mill, the British practice is to express the figure as pressure/unit length of roller. However, a pressure of 200 t/m (60 t./ft.) would give the same total pressure, *i.e.* 360 tonnes, on a roller of 863 x 1830 mm (34 x 72 in.) as on one of 810 x 1830 mm (32 x 72 in.), but the effect will be different.

If we consider (Fig. 10.8) a transverse section of a mill in operation, we can quite well picture this figure as representing, perhaps rollers of 50 cm (20 in.) diameter on a scale of $\frac{1}{20}$, perhaps rollers of 100 cm (40 in.) diameter on a scale of $\frac{1}{40}$, or rollers of any diameter whatever at the appropriate scale.

It will be seen then that the layer of bagasse indicated on the figure will likewise correspond to a thickness determined by the scale of the figure, but proportional to the diameter of the roller: 10 cm (4 in.) for rollers of 50 cm (20 in.), or 20 cm (8 in.) for rollers of 100 cm (40 in.), etc.

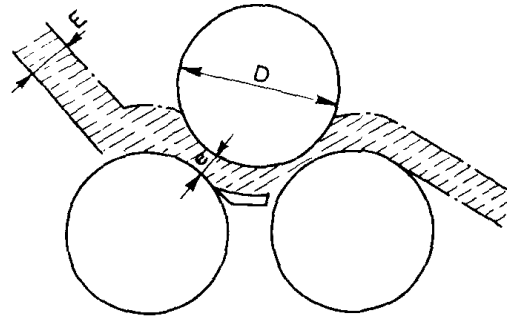


Fig. 10.8. The bagasse layer.

For the present, we propose to concern ourselves only with the pressure, and to disregard temporarily the more complex phenomena such as the flow of juice through the layer of bagasse, which, however, would not seriously affect our conclusions.

Other things being equal, it is obvious that, for a given diagram, *e.g.* for a given thickness of bagasse and at the roller setting shown, there will be an identical nip, degree of compression, ratio e/E and overall treatment of the bagasse, whatever the scale of the diagram. In other words, in the two cases below:

- | | |
|----------------------------------|-----------------------|
| (1) Diameter of rollers: | $D = 50$ cm (20 in.) |
| Thickness of loose bagasse: | $E = 10$ cm (4 in.) |
| Thickness of compressed bagasse: | $e = 1$ cm (0.4 in.) |
| | |
| (2) Diameter of rollers: | $D = 100$ cm (40 in.) |
| Thickness of loose bagasse: | $E = 20$ cm (8 in.) |
| Thickness of compressed bagasse: | $e = 2$ cm (0.8 in.) |

the result of the passage of the bagasse through the mill will be practically the same. But these two cases correspond to the same compression of the bagasse at homologous points in two geometrically similar figures (or to the same point in the same figure at two different scales), and this postulates that the pressure per unit area on the bagasse is the same at all corresponding points in its passage between the rollers. When the geometrical similarity is realised, the mechanical similarity will also be realised if the total pressure per unit length of the roller for the 100-cm (40-in.) mill is double that for the 50-cm (20-in.) mill. The length of the arcs in contact with the bagasse in the former case is effectively double that for the second case.

For a layer of bagasse of thickness equal to a given fraction of the diameter of the rollers (in the above example, 20% before pressing, 2% at delivery opening), the effect on the bagasse will be identical if the total pressure on the whole length of the roller is proportional to LD :

$$F = \lambda LD \quad (10.5)$$

- F = total hydraulic pressure acting on the top roller
 A = numerical coefficient
 L = length of rollers
 D — diameter of rollers.

Specific pressure

We now see that a logical way of taking into account the roller dimensions is to relate the total pressure, not to the length L of the rollers, but to the product LD of their length and diameter.

However, the pressure relative to LD does not represent anything tangible to the imagination. But it is found that, for a thickness of bagasse layer as ordinarily passed through the mills, the mean pressure exerted on the bagasse is approximately equal to that which would be exerted by the total pressure F if uniformly distributed on a plane surface of length L and breadth equal to one-tenth of the diameter, *i.e.* $D/10$.

We shall therefore define the *specific hydraulic pressure* of a mill, or s.h.p., by the quotient

$$P = \frac{F}{0.1 LD} = \frac{10 F}{LD} \quad (10.6)$$

p - specific hydraulic pressure, in tonnes/dm² (tons/sq.ft.) = "s.h.p."

F = total hydraulic pressure on the top roller, expressed in tonnes

L = length of the rollers, in dm (ft.)

D = diameter of the rollers, in dm (ft.).

This definition of s.h.p. is not closely related to the actual pressure exerted on the bagasse; it is merely the main factor in it, and gives an approximate figure of the order of magnitude of the mean pressure; it has the advantage of appealing to the imagination, as one can immediately picture a pressure, for example, of 10 t/dm² or 100 tons/sq.ft. of bagasse. It is also very useful, and will be used frequently in the following discussion. It should completely replace the figure of load per unit length of roller, which is illogical and liable to cause confusion.

Thickness of the bagasse layer. The s.h.p. permits of a comparison between the pressures of two different mills, but one must be cautious of using it as a measure of the pressure undergone by the bagasse. The s.h.p. would not provide a measure of this pressure unless the bagasse layer was always the same proportion of the roller diameter. This is not generally the case, either in two different mills, which may be under different control, or in the same mill, where the crushing rate may be increased or decreased.

Resolution of the pressures in the mill while operating

In an ordinary 2-roller crusher in operation, the reactions resulting from a load P exerted on the top roller are very simple. There is a reaction of F_s on the upper brasses of the top roller, and a reaction F_i on the bottom brasses of the bottom roller (Fig. 10.9), and each of these reactions is equal to P :

$$F_s = F_i = P$$

In a mill, the resolution is somewhat more complex. The reaction on the upper brasses of the top roller is still equal to P . If there were no trashplate, the load P would be resolved at the lower rollers into two reactions: F_1 at the feed and F_2 at the delivery roller (Fig. 10.10).

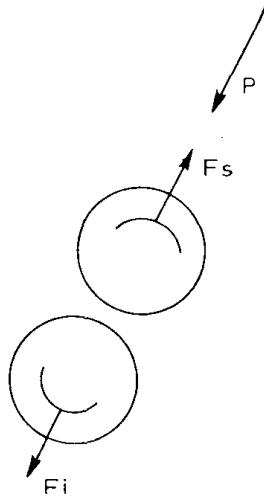


Fig. 10.9. Resolution of pressures in a crusher.

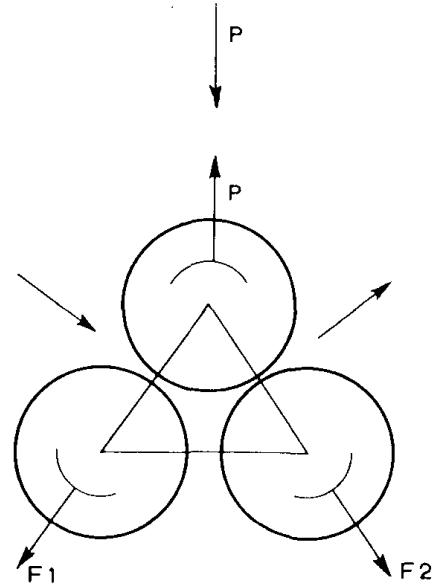


Fig. 10.10. Resolution of pressures in a mill.

The latter will always be greater, since the setting at the delivery is always closer than at the feed roller.

Influence of the trashplate. However, a certain portion of the pressure P is absorbed by the trashplate. The reaction of the latter is slightly displaced in front of the axial plane of the top roller. If we neglect this angle of displacement, which is small, we see that the reactions F_1 and F_2 will be produced only by the remaining fraction of P (Fig. 10.11).

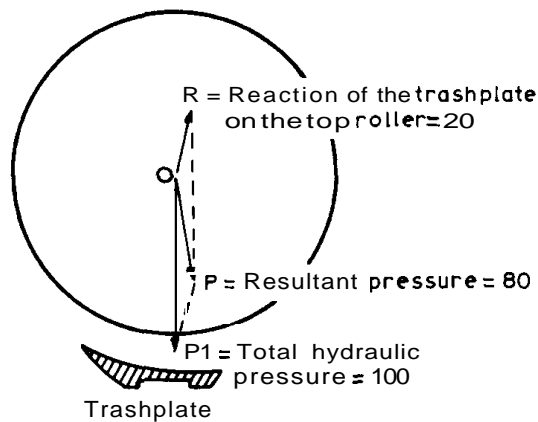


Fig. 10.11. Effect of trashplate reaction on the hydraulic load (relative pressures expressed as % total pressure).

What is the fraction of P absorbed by the trash plate? Lehky² estimates that the mean pressure on the trashplate is of the order of 3 kg/cm^2 (43 p.s.i.), which, for the usual proportions, would correspond to about 5% of P . This value appears to us to be very low. However, Russell^{2a} finds 1.8 kg/cm^2 (26 lb./sq. in.), corresponding to a total reaction $R = 0.05 P$. He finally proposes $R = 0.10 P$.

Munson³ has found normal values of $10 - 14 \text{ kg/cm}^2$ ($140 - 200 \text{ p.s.i.}$) but has stated that

this pressure often reaches 35 kg/cm² (500 p.s.i.) and can even exceed 60 kg/cm² (850 p.s.i.). On the other hand, some experiments conducted at the experimental factory at Audubon Park⁴ indicated that the reaction of the trashplate was proportional to the total pressure F , and approximately equal to half this total pressure. These results, as indeed their authors suggest, seem to err on the high side. Finally, some cross-checking, and particularly some tests by Varona⁵, which we interpret moreover in a manner slightly different from that of its author, and a test carried out at our factory of la Mare, lead us to the view that, in industrial mills, the proportion of the load absorbed by the trashplate would be in general of the order of 20 – 30% of the total hydraulic load. The Sugar Research Institute at Mackay⁶ agrees on this order of magnitude.

Munson, who suggested the tests at Audubon, considers that the setting of the trashplate has little effect on the value of this reaction, although the highest values are generally reported for very low settings; whether the plate be fixed high or low, the bagasse accumulates to the point where the thickness of the layer corresponds approximately to a certain constant reaction, which we have just estimated at 20 – 30% of the hydraulic pressure exerted on the top roller. We share this point of view provided, obviously, that the trashplate is not set *too* high or too low. If it is correct, there will be no need to consider the setting, which fortunately simplifies the problem.

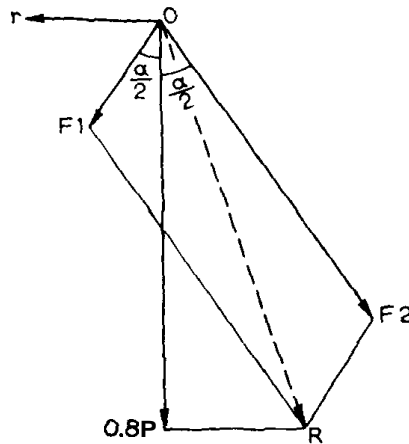


Fig. 10.12. Constancy of the sum of the two reactions: feed and delivery.

So then, 20 - 30% of the hydraulic pressure P being absorbed by the trashplate, there remains only some 75% to exert the pressures F_1 and F_2 at the entry and delivery openings. If we produce the lines of these forces to their point of intersection O , centre of the top roller, and project them on the vertical line (Fig. 10.12), we have:

$$F_1 \cos \frac{\alpha}{2} + F_2 \cos \frac{\alpha}{2} = 0.75 P$$

whence:

$$F_1 + F_2 = \frac{0.75 P}{\cos \alpha/2} \approx P \tag{10.7}$$

which shows that the sum of the reactions F_1 and F_2 remains constant even though their relative values may vary. (We shall see later (p. 164) a more accurate value for this sum.)

We can therefore enunciate the following principle:

Principle. The sum of the forces acting on the assembly of the working bearings of a mill has a constant value, which is approximately:

$$P + F_1 + F_2 = P + P = 2P \quad (10.8)$$

Crusher. For a crusher, the sum of the forces acting on the assembly of the 4 active bearings obviously has the value (Fig. 10.9):

$$F + F = 2P \quad (10.9)$$

We see then that for an equal pressure on the top roller, the forces acting on the group of bearings of a crusher are approximately equal to those acting on the bearings of a mill.

Definitions

We shall use henceforth the following nomenclature and abbreviations:

- t.h.p. = Total hydraulic pressure (or total hydraulic load) on the top roller. It is the number of tonnes obtained by multiplying the total area of the two pistons, in cm^2 (sq.in.), by the oil pressure expressed in tonnes/ cm^2 (tons/sq.in.)
- s.h.p. = Specific hydraulic pressure. This is the figure obtained by assuming the t.h.p. to be uniformly distributed over a rectangular plane surface of length equal to that of the rollers and of width equal to one-tenth of their diameter. It is expressed in tonnes/ dm^2 (tons/sq.ft.):

$$\text{s.h.p.} = \frac{\text{t.h.p.}}{0.1 LD} = \frac{10 (\text{t.h.p.})}{LD} \quad (10.10)$$

- r.h.p. = Resultant hydraulic pressure (or hydraulic load). This is the component, either on the feed side F_1 , or on the delivery side F_2 , corresponding to the t.h.p. In a crusher, r.h.p. = t.h.p.

- s.r.h.p. = Specific resultant hydraulic pressure:

$$\text{s.r.h.p.} = \frac{10}{LD} \quad (10.11)$$

PRESSURES IN MILLS

Phenomena influencing pressure in mills

It has long been believed that bagasse passing between mill rollers travels at a speed equal to the peripheral speed of the rollers which move it. The first edition of this manual was bas-

ed on that hypothesis; but more precise measurements have shown that this is not correct. As long ago as 1928, Egeter in Java had stated that the volume of the compressed bagasse, as it passes between the rollers, is greater than the volume generated by the rollers. He encountered much scepticism, as even technical people found it difficult to imagine a flow of material submitted to such pressures. More careful measurements in industrial mills, and laboratory experiments, particularly those of Bullock and Murry in Australia, have shown decisively that there is hardly ever equality between the volume of compressed bagasse and the volume generated by the rollers.

This phenomenon is still not very well understood, but may be explained in the following manner. Let us consider a layer *AB* of prepared cane or bagasse entering a mill (Fig. 10.13). Let us visualise vertical equidistant bands *AB*, *CD*, etc. as if they were marked in red, in this layer of bagasse. When these bands approach the rollers they will be disturbed by the compression of the bands preceding them; the material located on the surface and coming in contact with the rollers will be carried along by friction, but will drag the interior layers forward to a lesser extent only, particularly as they encounter greater and greater resistance due to the fact of the increased compression. Thus the bagasse has a tendency to flow back towards zones of lower pressure. These bands accordingly will at first bend backwards and, when the space between the rollers becomes very narrow, they will form pockets in the middle layer.

This material, however, is not solid like metallic fibre, nor fluid like steel on the point of melting; it consists of solid fibres and liquid juice. Under the influence of pressure, the cell walls burst, the juice tends to escape towards the zone of lower pressure and flows back through the fibrous mass towards the free exterior spaces *M* and *N*. At *N*, it escapes to the juice pan; at *M*, it cannot escape except by traversing the whole bagasse layer where it is not yet under pressure. It thus saturates the material and so contributes to increasing the proportion of juice therein and consequently the proportion which is trapped and carried on towards the opening.

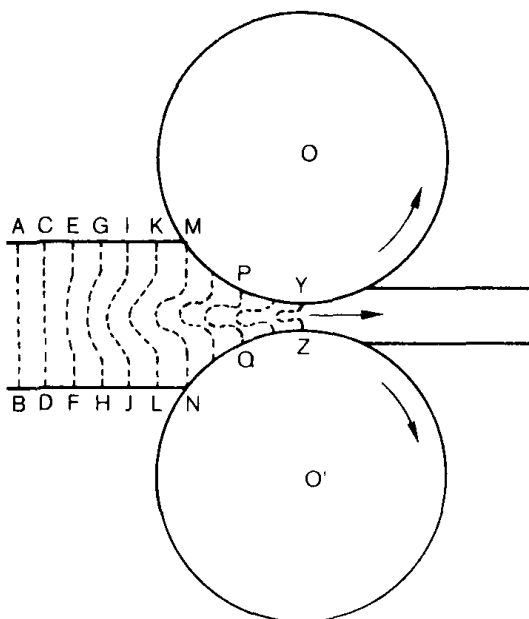


Fig. 10.13. Formation of semi-liquid pocket.

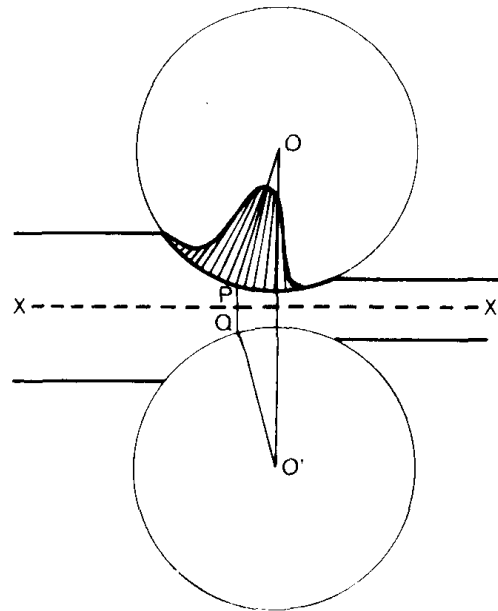


Fig. 10.14. Distribution of pressures on the roller.

A new factor now comes into play; the permeability of the fibrous mass. This permeability is very high in loose bagasse but drops abruptly when the pressure increases and becomes practically nil at very high pressures, as demonstrated by Bullock and Murry. The juice which has not been able to escape in the zone of high permeability, or low pressure, thus finds itself trapped. There arrives then at the zone of very high pressures, towards PQ , a material consisting of juice intimately intermingled with fibre. This material assumes under such pressures the state which Linley has described as "semi-liquid" (Murry⁷); it possesses peculiar properties, and the liquid shows a violent tendency to squirt through the first available opening, probably taking with it a certain proportion of fibre.

The juice which has thus been trapped in the middle of the bagasse layer forms a pocket which is well depicted by our distorted vertical bands (Fig. 10.13). We shall now consider what happens to this pocket of material: when it arrives at the axial plane OO' , it finds ahead of it a zone of decreasing pressures. The liquid will then obviously be thrown forward and will find in front of it a bagasse relatively dry and expanding like a sponge, in which it will immediately be re-absorbed. We see here one of the deficiencies of the crude system of extraction provided by the milling process; an enormous amount of energy is expended to release the juice, but a large fraction of the juice thus liberated is immediately lost again by re-absorption. This disadvantage is a matter of the nature of the materials and would be very difficult to overcome.

Consider again the pocket of juice: in the last inch preceding the axial plane OO' , the opening between the rollers is barely greater than that at the axial plane. If it is 0.80 in. at OO' , it would be 0.81 in. at $\frac{1}{2}$ in. before the axial plane. It is readily seen that, in the semi-fluid state described, squirting through the delivery opening will actually take place well ahead of the axial plane. However, as soon as squirting takes place, the pressure falls or at least ceases to rise. It follows that the maximum pressure takes place not at the axial plane but in an earlier plane, in the neighbourhood of PQ , called the "neutral plane".

The distribution of pressure in the mill is thus displaced towards the feed side (Fig. 10.14) relative to what is assumed in picturing the maximum as being at the axial plane. It is probably practically constant between the neutral plane and the axial plane, and falls off rapidly beyond the latter. We have represented these pressure changes by ordinates placed on the corresponding radii of the top roller.

The neutral plane PQ may be located at 25 - 50 mm (1 - 2 in.) before the axial plane. The squirting, or at least the flow at a speed above that of the rollers, commences at the neutral plane and finishes just after the axial plane.

A certain proportion of juice therefore passes through YZ (Fig. 10.13) at a higher speed. It is possible that this is a small proportion squirting at high velocity through the medium layer xx' (Fig. 10.14); it is possible also that practically all the juice passes through the fibrous mass at a speed only 10, 20 or 50% higher than that of the rollers. It is possible finally that the juice, being thrown forward in this way, entrains with it some solid material. The term "extrusion" is applicable to this latter hypothesis: there is an extrusion of the solid-liquid material under pressure through the delivery opening.

The term "re-absorption" is applicable in any case, since it refers to juice passing through the opening at increased velocity and becoming re-absorbed after the delivery opening in a mass of bagasse which has been deprived of its juice and is in a condition to absorb moisture eagerly.

Re-absorption factor or excess-speed coefficient

We may use the term "excess-speed coefficient" for the ratio r of the mean speed of the material through the opening to the peripheral velocity of the rollers. This ratio is also called the "re-absorption factor", or "coefficient of slip". The latter term is hardly appropriate, since one does not normally think of a negative slip at high pressures; "forward-slip factor" would perhaps be a more descriptive term.

This factor is easily calculated, when the volume of material is known as well as the volume generated by the rollers:

$$r = \frac{V_B}{V_E} \quad (10.12)$$

r = re-absorption factor

V_B = volume of the bagasse as it passes through the delivery opening

V_E = volume generated at the delivery opening by the rotation of the two rollers.

The ratio r increases with degree of compression. When it reaches unity, the phenomenon of re-absorption has actually already commenced. If we consider a mill with a very wide opening, the phenomena due to very high pressures and which we have described above will not be produced. Loose bagasse, however, except the layer in contact with the rollers, is not dragged forward bodily at the peripheral velocity. Slip therefore occurs and the ratio r is less than 1. The term "coefficient of slip" is then fully justified. It reaches unity only when, the pressure increasing or the mill being closed in, re-absorption commences and becomes sufficient to compensate the normal slip. It is no cause for surprise that the graph obtained in practice, giving the extraction as a function of increasing tonnage, for a mill with a given opening, crosses the theoretical curve (Fig. 10.15). (We understand by the theoretical curve the curve which would be obtained in the absence of re-absorption, *i.e.* if all the bagasse material was moved forward at the peripheral speed of the rollers.) There would otherwise be no explanation for the part of the graph corresponding to values of r less than 1.

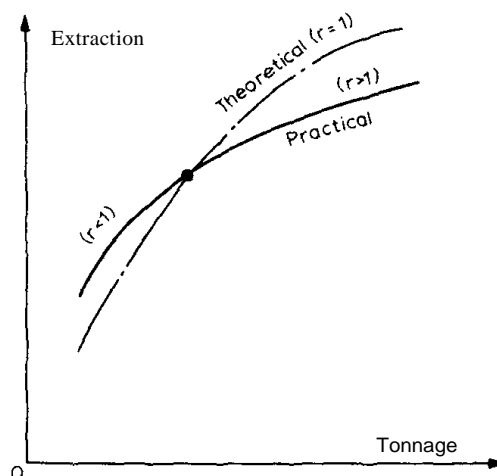


Fig. 10.15. Theoretical and practical extraction curves.

Densities of fibre, cane and bagasse

One of the first questions which arise is: how can the volume of the material under pressure be measured? It is apparently very variable, and depends on its fibre, its state of subdivision, the size and the shape of the pieces, on the thickness of the bagasse layer (a thick layer compacts more than a thin layer), on the brix of the juice, etc.

No-void volume. These variations and these sources of uncertainty are much reduced if, instead of the apparent volume, the no-void volume of the material is used. The material consists of two substances: fibre and juice.

A. Fibre. The density of fibre is now well known. The density of cellulose being 1.55, that of the fibre cannot differ greatly from this figure. The best determination which has been made is that of Pidduck⁸; this gives:

$$\delta = 1.512 + 0.000023 p \quad (\delta = 1.512 + 0.00000161 p) \quad (10.13)$$

δ = specific gravity of cane fibre under a pressure p

p = pressure to which the fibre is subjected, in kg/cm² (tons/sq.in.)

The value of 1.512 is given with a precision of ± 0.01 .

At:

Atmospheric pressure:	$\delta = 1.512$
100 kg/cm ² (1,400 p.s.i.):	$\delta = 1.514$
500 kg/cm ² (7,000 p.s.i.):	$\delta = 1.523$

There will be no significant error if we adopt a figure of 1.52 for fibre under pressure in a mill.

B. Juice. In Java it was considered that the liquid fraction of the cane consisted of (a) juice proper, of variable density; (b) water of constitution or hygroscopic water, which is practically pure water or at least of very low brix; this is closely associated with the fibrous material, and even the highest pressures cannot separate it from the fibre. It was estimated that this water of constitution amounted to 25% of the weight of fibre; others estimated 20%. Tests made by Van der Pol in South Africa had given approximately 30% but have later⁹ been corrected to 20%. This proportion, moreover, varies in the course of the season. If we adopt a value of 20%, the no-void volume of a kilogram of cane or bagasse would have the value:

$$V_0 = \frac{f}{1.52} + \frac{0.20 f}{1} + \frac{1 - 1.20 f}{d_J} \quad (10.14)$$

V_0 = specific no-void volume of cane or bagasse, in dm³/kg (for cu.ft./lb. x 0.016)

f = fibre per unit of cane or bagasse

d_J = density of juice (or of mixture of juice and imbibition water).

This may be written:

$$V_c = \frac{1}{d_J} - \left(\frac{1.20}{d_J} - 0.86 \right) f \quad (10.15)$$

The density of juice d_{J_0} under atmospheric pressure being known, its density d_J under a pressure p (kg/cm² or lb./sq.in.) has approximately the value:

$$d_J = d_{J_0} (1 + 0.00004 p) \quad (d_J = d_{J_0} (1 + 0.0000028 p)) \quad (10.16)$$

Or, with sufficient accuracy for juice under pressure in the mill:

$$d_J = 1.01 d_{J_0} \quad (10.17)$$

Consider first the case of cane: $d_{J_0} = 1.07-1.09$, say approximately 1.08; $f = 0.10-0.15$:

$$\text{Minimum: } V_0 = \frac{1}{1.09 \times 1.01} - \left(\frac{1.20}{1.09 \times 1.01} - 0.86 \right) 0.15 = 0.874 \quad \text{or: } d_c = 1.144$$

$$\text{Maximum: } V_0 = \frac{1}{1.07 \times 1.01} - \left(\frac{1.20}{1.07 \times 1.01} - 0.86 \right) 0.10 = 0.901 \quad \text{or: } d_c = 1.110$$

d_c = no-void density of cane under pressure: no significant error will be involved if we adopt $d_c = 1.12$ kg/dm³ (70 lb./cu.ft.).

Consider now the case of first bagasse: $d = 1.06- 1.08$; $f = 0.30-0.36$:

$$\text{Minimum: } V_0 = \frac{1}{1.08 \times 1.01} - \left(\frac{1.20}{1.08 \times 1.01} - 0.86 \right) 0.36 = 0.830 \quad \text{or: } d_B = 1.204$$

$$\text{Maximum: } V_0 = \frac{1}{1.06 \times 1.01} - \left(\frac{1.20}{1.06 \times 1.01} - 0.86 \right) 0.30 = 0.856 \quad \text{or: } d_B = 1.169$$

d_B = no-void density of bagasse under pressure.

For a final mill: $d = 1.01 - 1.02$; $f = 0.46 - 0.54$:

$$\text{Minimum: } V_0 = \frac{1}{1.02 \times 1.01} - \left(\frac{1.20}{1.02 \times 1.01} - 0.86 \right) 0.54 = 0.806 \quad \text{or: } d_B = 1.241$$

$$\text{Maximum: } V_0 = \frac{1}{1.01 \times 1.01} - \left(\frac{1.20}{1.01 \times 1.01} - 0.86 \right) 0.46 = 0.835 \quad \text{or: } d_B = 1.198$$

Note: The above densities are given in kg/dm³; for conversion to lb. per cu.ft. they should be multiplied by the density of water (62.4 lb./cu.ft.).

It is seen that the density of the bagasse is of the order of 1.20 kg/dm³ (75 lb./cu.ft.).

Fibraque

The Dutch workers in Java considered the sum of fibre and water of constitution as a com-

pound which could not be separated mechanically, since mills were not able to effect that separation. They called this combination "natural fibre". We shall at times adopt this concept but, to avoid any confusion, we shall give this combination the name of "fibrage", signifying fibre plus water.

As a general rule we shall suppose that the weight of fibrage represents 1.2 times the weight of dry fibre.

Fibre loading

We shall use the term "fibre loading" to designate the weight of fibre per unit area described by one of the mill rollers. We have:

$$q = \frac{Af}{S} = \frac{Af}{60 \pi nDL} \quad (10.18)$$

q = fibre loading, in kg/m² (lb./sq.ft.)

A = crushing rate, in kg/h (lb./h)

f = fibre per unit of cane

S = area described by a generatrix of the top roller in 1 hour, in m² (sq.ft.)

L = length of rollers, in m (ft.)

D = mean diameter of rollers, in m (ft.)

n = rotational speed of rollers, in r.p.m.

The fibre loading varies greatly. A mean order of magnitude would be, for example: $q = 12 \text{ kg/m}^2$ (2.5 lb./sq.ft.). However, as we have seen (p. 125), the layer of cane or of bagasse should, other things being equal, be proportional to the roller diameter. Since the fibre loading itself is proportional to the thickness of the bagasse layer, it follows that, in order to compare the loading of two mills of different roller diameters, that is to say the work which is required of them, it is necessary to relate the fibre loading to the diameter. Hence the concept of "specific fibre loading":

$$\text{Specific fibre loading: } r = \frac{q}{D} \quad (10.19)$$

T = specific fibre loading in kg/m²/m, i.e. in kg/m³ (lb./cu.ft.).

The specific fibre loading, obviously less variable than the fibre loading, is of the order of 16 kg/m³ (1 lb./cu.ft.).

Apparent volume

The apparent density of cane, including voids, corresponding to the apparent volume which it occupies on the carrier or at the entry to the mill, under zero pressure, depends on the degree of preparation. Murry^{9a} gives as a close approximation:

	<i>Fine</i>	<i>Medium</i>	<i>Coarse</i>	
d_0	0.660 (41.2)	0.634 (39.6)	0.548 (34.2)	kg/dm ² (lb./cu.ft.)

It is thus of the order of 0.6.

It is valuable to know the apparent volume of the cane or the bagasse as presented to the mill, since the lower this volume, the more the feeding of the mill is facilitated. In order to measure this apparent volume, we shall make use of the "compaction coefficient", which we shall define as follows:

$$\text{Compaction coefficient} = \frac{\text{weight of fibre}}{\text{total apparent volume of the material}} \quad (10.20)$$

expressed as kg/dm^3 (lb./cu.ft.).

This coefficient is of the order of 0.08–0.09 kg/dm^3 (5–6 lb./cu.ft.) for cane and for bagasse, in the absence of forced feeding. It is independent of the moisture content. It has been demonstrated in Australia¹⁰ that it remains practically unchanged whatever the imbibition, except at the first addition of water, which causes a slight compaction of 4%. With increasing imbibition it varies by less than 2% when the imbibition per cent fibre varies between limits as wide as 88 to 454, which are far below and above normal industrial values.

This coefficient could be defined as the ratio of volume of fibre to the total apparent volume, thus obtaining a dimensionless coefficient, which is desirable. This would be simply our coefficient t divided by the density of fibre; but the advantage of the coefficient weight/volume (eqn. (10.20)) is that the weight of fibre is readily obtained, whereas the volume would necessitate the additional step of division by 1.52 for metric units, 95.0 for British units; moreover, the weight of fibre represents approximately the volume of fibre plus water of constitution, or in other words, the volume of fibraque. It would represent this exactly if the water of constitution amounted to 34% of the fibre. Since the water of constitution cannot be extracted, this volume could be considered as the ideal minimum to which the mills should be able to reduce the material presented to them. Actually, this ideal is still beyond the capability of actual mills.

Thus in expressing t as weight per unit volume, we have in effect the practical equivalent of a dimensionless coefficient, representing in our opinion practically a ratio of volume per unit volume.

Fibre index

Another quantity necessary for the study of milling work is the quantity of fibre which can be made to pass through the delivery opening of a mill. For this purpose the weight of fibre passed through the opening in unit time is related to the volume generated by that opening when operating, in the same unit of time. This ratio thus bears a certain relationship to the compaction coefficient; both expressions have the same numerator, expressed in a ratio, in the former case to the volume before pressure is applied, and in the second case to the theoretical volume under pressure. As we have seen (p. 135) that fibre is **almost** incompressible, the quantity of fibre which can be passed through the mill will depend mainly on two factors: (1) the quantity of juice which will pass through with it; the less juice, the more space for fibre; (2) the re-absorption factor; the higher this factor, the more fibre will pass through a given opening.

The fibre index, or weight of fibre per unit escribed volume, is defined as:

$$\text{Fibre index} = \phi = \frac{\text{weight of fibre passed through the opening}}{\text{scribed volume of the opening}} \text{ (g/dm}^3\text{)} \quad (10.21)$$

In British units it is expressed as lb./cu.ft.

This coefficient, generally applied to the delivery opening of a mill, can equally well apply to the feed opening. At the delivery, it varies for example from 400 to 880 g/dm³ (25 to 55 lb./cu.ft.). It increases along the milling train, being minimal at the first mill and maximal at the last mill.

In order to show better the significance of this figure, we may recall that first-mill bagasse of 32% fibre contains $320 \div 0.86 = 372$ g of fibre per dm³ of no-void volume (23 lb./cu.ft.); and that a final bagasse of 55% fibre contains $500/0.82 = 610$ g of fibre per dm³ of no-void volume (38 lb./cu.ft.). The differences between the fibre index and these figures are due to the re-absorption factor.

When expressed, like the compaction coefficient, in kg/dm³, the fibre index can also, for the same reasons, be thought of as representing a ratio of volume to volume: the ratio of volume of fibraque to scribed volume. It is similar in this sense to a dimensionless coefficient, and thus represents the ratio of non-extractable volume to scribed volume and thus expresses the effort of compression exerted by the mill on the material. As it represents in this way a ratio of volume to volume, it is therefore in general lower than 1, except in the case of high re-absorption.

Compression ratio

The principal factor which determines the extraction in a mill is the degree of compression, *i.e.* the decrease in volume to which one subjects the material from which the juice is to be extracted. In the absence of a convenient means of measuring the re-absorption factor, the apparent decrease in volume is measured, and this is expressed by the "compression ratio" introduced by Bullock:

$$c = \frac{\text{no-void volume of the material entering the mill}}{\text{volume available for the compressed material}} \quad (10.22)$$

The compression ratio of a mill, or more precisely the compression ratio corresponding to the passage of the material between two rollers (for compression ratio may be applied either to the feed opening or the delivery opening), is the maximum value assumed by this ratio c in the course of the passage through the mill, *i.e.* the value corresponding to the opening in the axial plane. We shall designate this by c_A :

$$c_A = \frac{\text{no-void volume of material entering the mill}}{\text{volume generated by the opening between the rollers}} \quad (10.23)$$

The compression ratio is really of interest only for a first mill, which receives cane of a density which is well known. For the other mills, and even for the first, the fibre index is more useful since, even in a first mill, the performance and the delivery opening depend less on the volume of material presented than on the fibre.

Opening between rollers

There are two openings between mill rollers which are of interest; in other words, two spaces corresponding to particular generatrices of each of these two rollers:

(a) First, the opening in the axial plane of the two rollers, which we shall designate briefly as "axial opening". If this opening has a value e_A , the value which is interesting, and which will express whether the mill opening is large or small, is the ratio:

$$\epsilon_A = \frac{e_A}{D} \quad (10.24)$$

as will be immediately understood by a reference to Fig. 10.8 and to the considerations of p. 126. In other words, a mill with rollers 50 cm (20 in.) in diameter, for example, with an opening $e_A = 12.5$ mm (3 in.), is open to the same extent as a mill with 100-cm (40-in.) diameter rollers, with opening $e_A = 25$ mm (1 in.). Both of these have the same value $\epsilon_A = 0.025$.

We shall designate this ratio ϵ_A by the term "specific opening of the mill".

(b) Secondly, the opening in the vertical neutral plane (*cf.* p. 132). We shall designate this opening by e_N and we shall be interested in the same way or particularly in the value:

$$\epsilon_N = \frac{e_N}{D} \quad (10.25)$$

If we recall that the neutral plane is that in which the material has the same mean speed as the surface of the rollers, we see that it is the space e_N which permits us to calculate very simply the volume of cane or bagasse passing.

There is a very simple relation between these two openings:

$$e_N = r e_A \quad \text{or:} \quad \epsilon_N = r \epsilon_A \quad (10.26)$$

which is essentially the definition of r (*cf.* eqn. (10.12)).

Position of neutral plane

It is interesting to know the position of the neutral plane. We shall fix it by the angle ν between the corresponding radius of the roller and the axial plane (Fig. 10.16) and will adopt the method of calculation of Murry¹¹. Let us consider a point P of the top roller and the corresponding point Q on the lower roller. Let the angle $YOP = \theta$. The thickness $h (= PQ)$ of the layer of bagasse at this point has the value:

$$h = D (1 - \cos \theta) + e_A \quad (10.27)$$

or:

$$h = D (1 + \epsilon_A - \cos \theta) \quad (10.28)$$

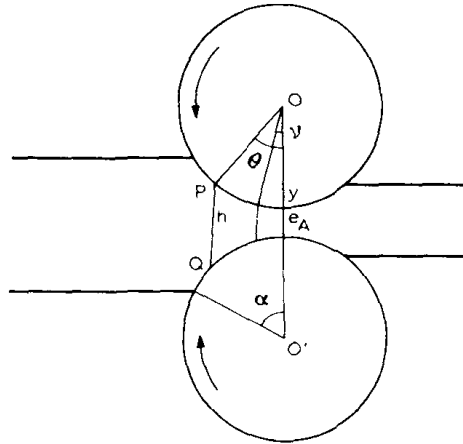


Fig. 10.16. Position of neutral plane.

Let v be the peripheral speed of the rollers and u the mean speed of the material passing through the mill, that is, such material as remains after the delivery, excluding the juice extracted by the two rollers which passes neither the neutral plane nor the axial plane. Let V_B be the volume of this material. At the point P , we have:

$$u = \frac{V_B}{Lh} = \frac{V_B}{LD(1 + \epsilon_A - \cos \theta)} \quad (10.29)$$

L = length of the rollers.

But:

$$V_B = r V_E \quad (\text{cf. eqn. (10.12)})$$

V_E = volume generated by the axial opening.

Or:

$$V_B = r e_A v L \quad (10.30)$$

Hence:

$$u = \frac{r e_A v L}{LD(1 + \epsilon_A - \cos \theta)} = \frac{r \epsilon_A v}{1 + \epsilon_A - \cos \theta} \quad (10.31)$$

The horizontal component of the roller speed has the value $v \cos \theta$. The position of the neutral plane will be given by the value of θ which we shall call ν at the neutral plane, for which:

$$v \cos \nu = u = \frac{r \epsilon_A v}{1 + \epsilon_A - \cos \nu}$$

or:

$$\cos \nu = \frac{r \epsilon_A}{1 + \epsilon_A - \cos \nu}$$

Hence:

$$\begin{aligned} \cos^2 \nu - (1 + \epsilon_A) \cos \nu + r \epsilon_A &= 0 \\ \cos \nu &= \frac{1 + \epsilon_A}{2} + \sqrt{\left(\frac{1 + \epsilon_A}{2}\right)^2 - r \epsilon_A} \end{aligned} \tag{10.32}$$

Giving typical values to ϵ_A and r , we obtain Table 10.1.

TABLE 10.1
VALUES OF $\cos \nu$ AND ν

r	ϵ_A			
	0	0.01	0.02	0.05
1	$\cos \nu = 1$ $\nu = 0$	$\cos \nu = 1$ $\nu = 0$	$\cos \nu = 1$ $\nu = 0$	$\cos \nu = 1$ $\nu = 0$
1.2	$\cos \nu = 1$ $\nu = 0$	$\cos \nu = 0.998$ $\nu = 3^\circ 39'$	$\cos \nu = 0.996$ $\nu = 5^\circ 11'$	$\cos \nu = 0.989$ $\nu = 8^\circ 21'$
1.5	$\cos \nu = 1$ $\nu = 0$	$\cos \nu = 0.995$ $\nu = 5^\circ 47'$	$\cos \nu = 0.990$ $\nu = 8^\circ 14'$	$\cos \nu = 0.973$ $\nu = 13^\circ 22'$

Compression (Hugot)

In our first edition, we had expressed the degree of compression effected by the mill by adopting the definition:

$$\text{"Compression"} \approx C = \frac{e_A}{H} \tag{10.33}$$

C = maximal theoretical compression undergone by the bagasse in its passage through the axial plane of the mill

e_A = opening between the rollers, in the axial plane

H = thickness of the layer of material, before entry to the mill.

We know now that the true compression is only:

$$C = \frac{r e_A}{H} \approx \frac{e_N}{H} \tag{10.34}$$

The graph of theoretical compression had been established from Noel Deerr's experiments on static compression of bagasse, based on a compaction coefficient of 0.07 kg/dm^3 (4.4 lb./cu.ft.).

This "compression" was, therefore, except for a numerical coefficient, the inverse of Bullock's "compression ratio". The numerical coefficient is involved because our figure was based on the loose bagasse, as the starting point for Deerr's experiments, corresponding to a pressure of 77 g/cm^2 (1.1 lb./sq.in.) and a compaction coefficient t of 0.07 kg/dm^3 (4.4 lb./cu.ft.), whereas the compression ratio is based on the no-void volume of the material ($t = 0.12 - 0.16 \text{ kg/dm}^3$ ($7 - 10 \text{ lb./cu.ft.}$) for cane ($0.40 - 0.60 \text{ kg/dm}^3$, ($25 - 38 \text{ lb./cu.ft.}$) for bagasse) (*cf.* p. 143).

Relationships between the various coefficients

1. **Fibre index ϕ .** We have (*cf.* eqns. (10.18) and (10.21)):

$$q = \frac{Af}{S} \quad \phi = \frac{Af}{S e_A}$$

whence:

$$\phi = \frac{q}{e_A} = \frac{\frac{q}{D}}{\frac{e_A}{D}} = \frac{\tau}{\epsilon_A} \quad (10.35)$$

ϕ = fibre index, in kg/m^3 (lb./cu.ft.)

q = fibre loading, in kg/m^2 (lb./sq.ft.)

T = specific fibre loading, in kg/m^3 (lb./cu.ft.)

e_A = opening between rollers, in m (ft.)

ϵ_A = specific opening = e_A/D

D = diameter of rollers, in m (ft.).

Let us follow the cane along the mills. Let B be the weight of bagasse leaving a mill and f' the fibre content of the bagasse. Writing that the weight of fibre remains constant, we have:

$$B f' = A f \quad B = A \frac{f}{f'} \quad (10.36)$$

B = weight of bagasse leaving one of the mills, in kg/h (lb./h)

f' = fibre per unit weight of this bagasse

A = crushing rate of the mills, in kg/h (cane) (lb./h)

f = fibre content per unit of cane.

But we have (eqn. (10.18)):

$$q = \frac{Af}{S}$$

whence:

$$Bf' = qS \quad B = \frac{qS}{f'} \quad (10.37)$$

Moreover, in the delivery (or feed) opening of the mill:

$$B = r V_E d_B = r e_A S d_B \quad (10.38)$$

r = re-absorption coefficient

V_E = volume generated per hour by the opening, in m³/h (cu.ft./h)

d_B = density of the compressed bagasse, in kg/m³ (lb./cu.ft.)

S = area described (scribed area) by the top roller, in mVh (sq.ft./h)

e_A = opening between rollers, in m (ft.).

Hence, by eqns. (10.38) and (10.37):

$$e_A = \frac{B}{r S d_B} = \frac{qS}{r f' S d_B} = \frac{q}{r f' d_B} \quad (10.39)$$

Hence from eqn. (10.35):

$$0 = r f' d_B \quad (10.40)$$

Equation (10.39) will be useful in problems of mill settings (*cf.* p. 208); it permits calculation of the opening e_A when the re-absorption coefficient is known, the three other quantities q , f' and d_B being given by the chemical control or being readily calculated.

It is seen by eqns. (10.35) and (10.39) that, for the same crushing rates of the mill, the axial opening e_A is inversely proportional to the fibre index ϕ :

$$e_A = \frac{q}{\phi} \quad (10.41)$$

2. Compression (EH). In the same way, we have (*cf.* eqns. (10.20) and (10.33)):

$$t = \frac{Af}{V_A} \quad C_{EH} = \frac{e_A}{H} = \frac{e_A S}{H S} \sim \frac{V_E}{V_A}$$

t = compaction coefficient, in kg/m³ (lb./cu.ft.)

V_A = total apparent volume of the material entering the mill (m³).

Hence:

$$\frac{t}{C_{EH}} = \frac{Af}{v_E} = \frac{Af}{s e_A} = \frac{q}{e_A} = \phi$$

It must be kept in mind that C_{EH} assumes $t = 0.07 \text{ kg/dm}^3$ (4.4 lb./cu.ft.). We have then*:

$$C_{EH} = \frac{0.07}{\phi} = \frac{0.07 e_A}{q} = 0.07 \frac{\epsilon_A}{T} \quad \left(\begin{array}{l} t \\ EH \end{array} = \frac{4.4}{\phi} = 4.4 \frac{e_A}{q} = \frac{\epsilon_A}{F} \right) \quad (10.42)$$

C_{EH} = "compression", starting from a material of compaction ratio $t = 0.07 \text{ kg/dm}^3$ (4.4 lb./cu.ft.).

3. Compression ratio C_A (Bullock). The compression ratio of Bullock is applicable mainly to cane, at the first mill. We have:

$$C_A = \frac{\text{no-void volume of material entering the mill}}{V_E} = \frac{\frac{A}{d_c}}{s e_A} = \frac{\frac{A}{d_c}}{\frac{AF}{Q} \rho_A}$$

or:

$$C_A = \frac{q}{d_c f e_A} \left(\approx \frac{0.9 q}{f e_A} \right) \quad (10.43)$$

or:

$$C_A = \frac{\phi}{f d_c} \left(\approx \frac{0.9 \phi}{f} \right) \quad ((10.44)$$

4. Fibre index and compaction coefficient. We have:

$$t = \frac{\text{weight of fibre}}{\text{total apparent volume of material}} \quad \rho = \frac{\text{weight of fibre}}{\text{scribed volume}}$$

hence:

$$\frac{\phi}{t} = \frac{\text{total apparent volume of material entering the mill}}{\text{scribed volume}} = \frac{V_A}{V_E} \quad (10.45)$$

* Nomenclature, p. 175.

Static compression

Let us consider a certain quantity of loose bagasse, *i.e.* not compressed and in the condition, for example, in which it arrives at one of the later mills (Fig. 10.17a). Its apparent density, *i.e.* the weight per unit volume in bulk of this material, does not have a definite value, because it depends on the quantity of water contained in the bagasse, or absorbed by it. If we add water, for example by applying it to the bagasse by means of a spray, the volume is hardly changed, but the weight will become increased by the quantity of water applied. The bagasse has, in effect, much like a sponge, a great ability to absorb water: about 5 – 10 times its dry weight.

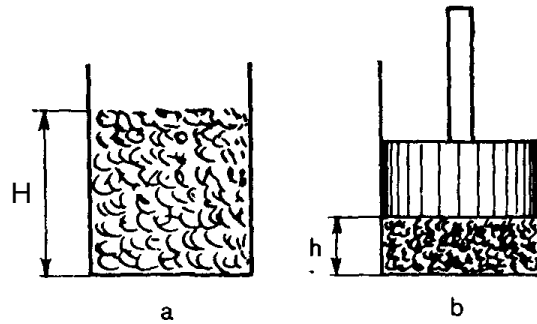


Fig. 10.17. Compression of the bagasse.

But there is one property which does not vary much in the loose bagasse, in its condition as it reaches the later mills. This is the weight of fibre which it contains per unit of volume. This weight is approximately 65 kg/m³ (4 lb./cu.ft.).

If, on this loose bagasse, we apply a gradually increasing pressure (Fig. 10.17b), its volume will decrease, rapidly at first, more slowly later. We use the term "compression" for the ratio:

$$c = \frac{h}{H} \tag{10.46}$$

of the volume of the compressed bagasse to that of the loose bagasse.

The volume of the compressed bagasse has a value which is rather more precise than that of the loose bagasse. All the reasoning which follows will, however, be based on this volume of loose bagasse, rather inaccurately measured, as at the starting point before pressure is applied: it will thus be easier to grasp, and less abstract. But it will be understood that, in practice, when it is necessary to transpose our arguments or our theoretical calculations into figures, this will always be done on the basis of the weight of fibre per unit volume, which is the only definite value.

The relationship moreover is simple. Let:

S = area of bagasse involved,

t = weight of fibre per unit volume of loose bagasse,

t' = weight of fibre per unit volume of compressed bagasse,

and remembering that the weight of fibre is the same before and after compression:

$$SH t = Sh t'$$

whence:

$$\frac{h}{H} = \frac{t}{t'}$$

We have then:

$$c = \frac{h}{H} = \frac{t}{t'} \quad (10.47)$$

In other words: the compression of the bagasse, or the ratio of the thickness of compressed bagasse to that of loose bagasse, may equally well be measured by the ratio of weight of fibre per unit volume of loose bagasse and compressed bagasse.

TABLE 10.2

EXPERIMENT BY **DEERR**: RELATION
BETWEEN THICKNESS OF A LAYER OF
BAGASSE AND THE PRESSURE EXERTED

<i>Pressure (kg/cm²)</i>	<i>Compression</i>
0.077	100
0.429	68.2
0.780	57.7
1.132	50.7
1.483	44.8
1.835	40.8
2.186	39.3
2.538	36.2
2.889	33.6
3.241	31.8
3.592	30.3
3.944	29
5.835	24.1
11.389	18.5
23.269	14.7
33.744	11.6
45	11.4
49.4	11
83.9	10.2
112	9.2
168	8.56
224	8.12
279.2	7.9
335.9	7.68
391.9	7.46
447.7	7.25
503.8	7.02
559.5	6.91
615.6	6.80
727.4	6.70
839.4	6.58

TABLE 10.3

VALUE OF THE EXPONENT IN EQN. (10.48) AND VALUE OF THE
"COMPRESSION" OF THE **BAGASSE**, AS FUNCTIONS OF THE
PRESSURE

<i>Pressure (kg/cm²)</i>	<i>Exponent</i>	<i>Compression</i>
1	2.51	0.5432
2	2.515	0.4116
4	2.54	0.3087
6	2.58	0.2591
8	2.64	0.2275
10	2.70	0.2059
12	2.78	0.1888
15	2.92	0.1697
20	3.18	0.1483
25	3.46	0.1347
30	3.73	0.1255
35	3.99	0.1190
40	4.22	0.1142
45	4.425	0.1105
50	4.605	0.1076
60	4.90	0.1032
70	5.115	0.1000
80	5.28	0.0975
90	5.405	0.09545
100	5.503	0.09373
120	5.638	0.0909
150	5.76	0.0876
200	5.86	0.0836
250	5.91	0.0806
300	5.937	0.0783
400	5.964	0.0747
500	5.977	0.072
600	5.984	0.070
700	5.988	0.0681
800	5.991	0.0666
900	5.993	0.0653
1.000	5.994	0.06417

Relation between pressure and compression

Evidently there exists a relationship between the compression of the bagasse and the pressure exerted to accomplish it. Noel Deerr made a study, now old, but so far a standard work, of this relationship. He has not made a complete mathematical study of it, though this is profitable. Bullock¹² has repeated these experiments in greater detail, and has verified his conclusions. These experiments of Noel Deerr have the disadvantage that they were made under static pressure conditions, by means of a piston acting on bagasse placed at the bottom of a cylinder; the juice thus could not escape in the same manner or in the same direction as with bagasse subjected to pressure in the nip of two rotating rollers. It is therefore necessary to relate the results of Deerr to those obtained in industrial practice; however, they do provide a useful basis from which to commence the study of pressures in milling.

We give in Table 10.2 a summary of the experiments of Noel Deerr. We have plotted the results on the graph (Fig. 10.18) where they are represented by the separate black points.

It is evident that at the beginning a light pressure is sufficient to reduce greatly the thickness of the bagasse. Then, in the neighbourhood of 20 kg/cm² (300 p.s.i.), the resistance increases quite abruptly, and higher and higher pressures are necessary to obtain a given degree of further compression. Above 50 kg/cm² (700 p.s.i.) the layer of bagasse becomes less and less compressible, and a pressure 10 times greater, say 500 kg/cm² (7,000 p.s.i.), reduces the thickness of the layer only in the ratio of 11 to 7.

Expressed as a formula, the relationship between the pressure exerted on the bagasse and the compression which results, may be written*.

$$P = \frac{70}{(10 c)^n} \quad \left(p = \frac{1,000}{(10 c)^n} \right) \quad (10.48)$$

where

$$n = 6 - \frac{5,800}{p^2 + 1,660} \quad \left(n = 6 - \frac{1.17 \times 10^6}{p^2 + 335 \times 10^3} \right)$$

p = pressure exerted on the bagasse, in kg/cm² (p.s.i.)

c = compression of the bagasse, defined by eqn. (10.46).

The exponent occurring in the denominator being somewhat complex, we give in Table 10.3 the values of this exponent and of c for various values of p . An excellent agreement will be noted between the values given by eqn. (10.48) and the results of Noel Deerr. The slight divergences are due to experimental errors, as is evident from the irregular nature of the broken line which would be necessary to join the points scattered on one side or the other of the curve in Fig. 10.18.

* Important equations are given in both metric and British units.

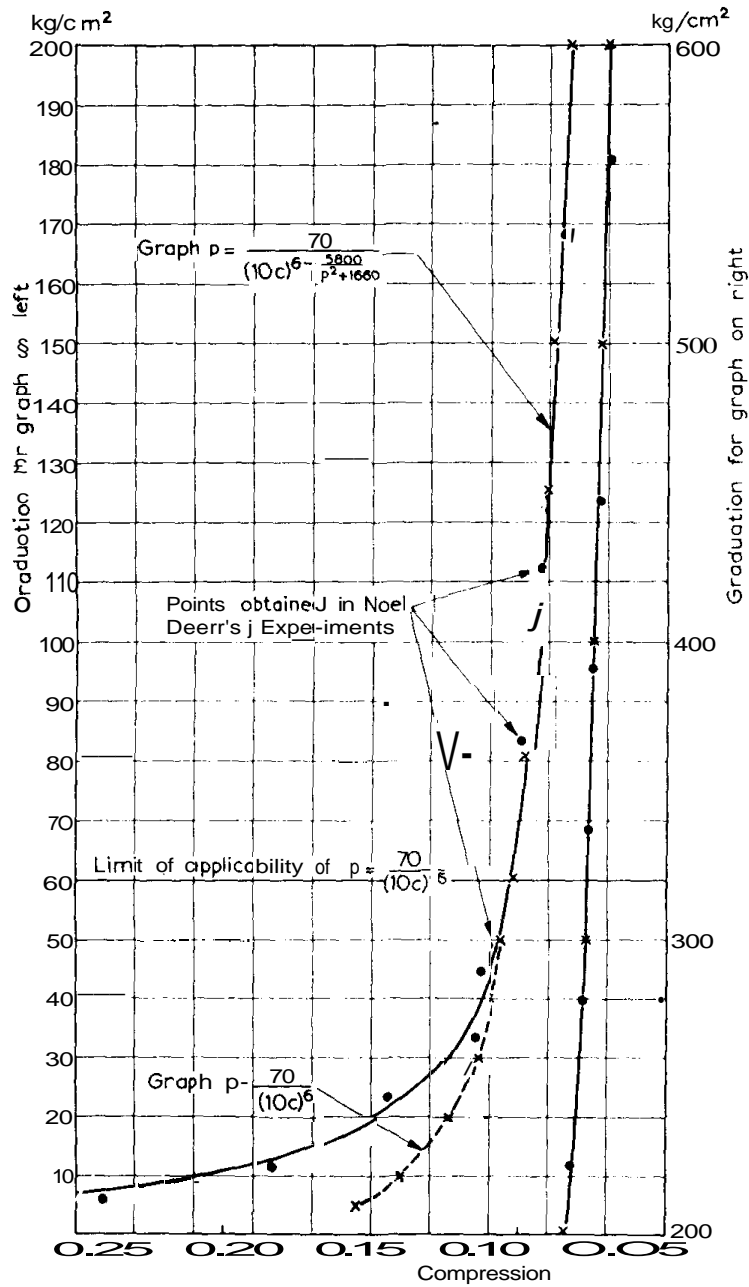


Fig. 10.18. Relation between pressure and compression of bagasse.

Equation (10.48) is interesting, but not easy to handle. Fortunately, for all values of $p \geq 50 \text{ kg/cm}^2$ (700 p.s.i.) it may be replaced with great accuracy by the simplified formula:

$$p = \frac{70}{(10 c)^6} \quad \left(p \approx \frac{1,000}{(10 c)^6} \right) \quad (10.49)$$

It is evident from the graph of Fig. 10.18 that the curves representing eqns. (10.48) and (10.49) practically coincide beyond $p = 50 \text{ kg/cm}^2$ (700 p.s.i.).

However, Noel Deerr's tests were done in the laboratory, under static conditions. Now, G. H. Jenkins has shown¹³ that the pressure required in a mill to obtain a given compression

c is appreciably greater than the static pressure recorded by Noel Deerr for the same compression. He explains this divergence by the additional pressure necessary to expel the juice through the layer of bagasse in the short interval of time between the moment when the juice begins to be expressed and the moment when it passes the axial plane of the cylinders.

As we are interested here only in the pressure in the mills, we shall make use of Jenkins' results to reconcile the results of Deerr with industrial practice. The ratio of the dynamic pressure in a mill and the static pressure corresponding to the same compression varies according to the state of subdivision of the bagasse and the speed of expulsion of the juice. Taking the average conditions, and working from the most representative points of Jenkins' experiments, we shall take for mill conditions:

$$P = \frac{88}{(10 c)^6} \quad \left(p = \frac{1,250}{(10 c)^6} \right) \quad (10.50)$$

Graph of pressure in the mill

We now know the law relating pressure and compression. It becomes of interest to study the behaviour in the mill.

We shall first disregard re-absorption and study the case of pressure between two rollers, assuming the re-absorption factor $r = 1$, *i.e.* the case where the neutral plane coincides with the axial plane.

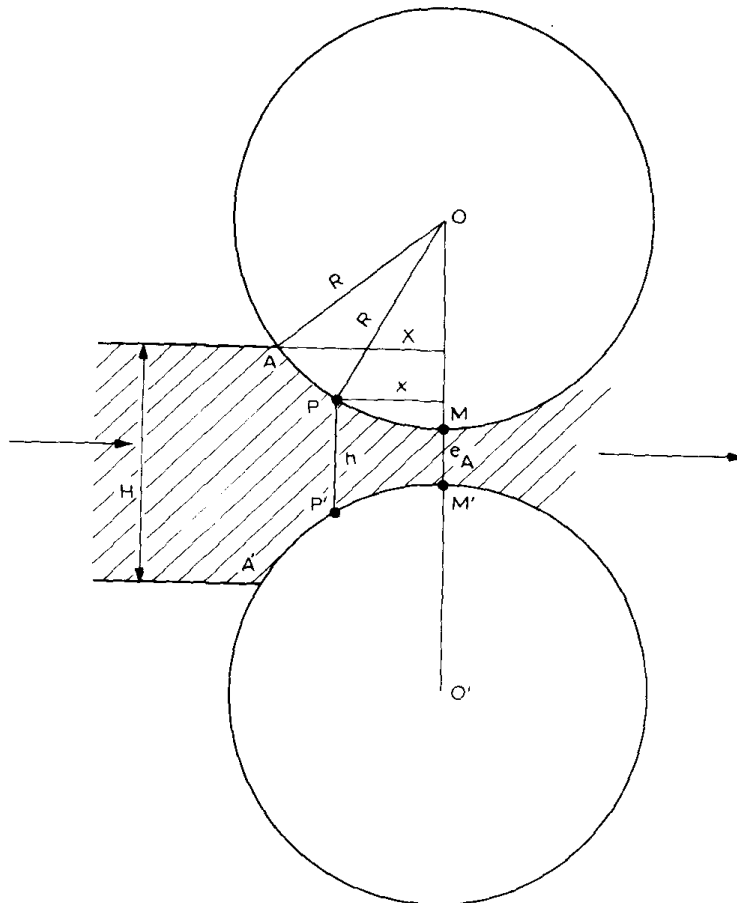


Fig. 10.19. Pressure in the mill.

Let us consider any two rollers, which could be, say, the rollers of a crusher, the two feed rollers, or the two delivery rollers of a mill.

Let

D = mean diameter of rollers, in m (ft.)

R = mean radius of rollers $\approx D/2$, in m (ft.)

H = thickness of the layer of loose bagasse entering the mill, in m (ft.)

e_A = distance between the mean surfaces of the two rollers, measured in their common axial plane, in m (ft.)

X - distance from the point where the layer of bagasse meets the roller to the axial plane of the two rollers (Fig. 10.19), in m (ft.).

The calculation of pressure was done in the second edition. It rests on the hypothesis that the layer of loose bagasse and the layer of compressed bagasse both move at the same velocity in a horizontal direction.

M. A. T. de Boer, engineer with Stork - Werkspoor, has commented that the true compression cannot be measured by the simple ratio h/H . In effect, in order to correspond to Fig. 10.18 and the experiments of Noel Deerr, it is necessary that the material before, during and after compression should correspond to the same bagasse, in other words, to the same weight of fibre.

Let v be the peripheral speed of the roller during unit time (e.g. 0.01 s), during which the surface advances by ds . Consider the vertical section of compressed bagasse between the element ds and the element ds' symmetrical with the roller O' (Fig. 10.19). The section PP' describes the volume $hLv \cos POM$ (L = length of roller). The compression at this point has the value:

$$C = \frac{hLv \cos POM}{HLv \cos AOM} \approx \frac{h \cos \beta}{H \cos \alpha} \quad (\text{and not } h/H) \quad (10.51)$$

designating the angles AOM and POM by α and β . The minimum compression thus has the value:

$$C = \frac{e_A}{H \cos \alpha} \quad (10.52)$$

We require to know:

- (1) The mode of variation of the pressure, from the point A , where it is zero, to the point M , where it is at a maximum
- (2) The value reached at M and corresponding to this maximum
- (3) The resultant of all pressures on all sections such as PP' , a resultant which obviously must be equal and opposite to the pressure F acting on the top roller. (In the case of a mill, the pressure F is obviously the r.h.p. on the side under consideration (cf. p. 130.)

(Actually, the pressure exerted by the bagasse is equal to the pressure P increased by a certain component of the weight of the roller; but we shall always neglect this component: (a) to avoid complicating the reasoning; (b) because this component is small relative to the hydraulic pressure; (c) because it is largely compensated by the reaction of the pinions, which

we shall discuss later, and which we shall neglect in the same way (*cf.* p. 166) and for the same reasons.)

Value of the pressure at any point. In Fig. 10.19, we have:

$$c = \frac{h \cos \beta}{H \cos a} = \frac{e_A + 2R - 2\sqrt{R^2 - x^2}}{H \cos a} \times \frac{\sqrt{R^2 - x^2}}{R} \quad (10.53)$$

$$c = \frac{1}{H \cos \alpha} \left[(e_A + 2R) \sqrt{1 - \frac{x^2}{R^2}} - \frac{2(R^2 - x^2)}{R} \right] \quad (10.54)$$

For all values of x which are low relative to R , we may replace the radical by its development as a series to its first two terms:

$$(1 - x)^m = 1 - \frac{m}{1} x + \frac{m(m - 1)}{1.2} x^2 - \dots$$

in which we put $m = \frac{1}{2}$ and replace x by $\frac{x^2}{R^2}$.

We then have:

$$c = \frac{1}{H \cos \alpha} \left[(e_A + 2R) \left(1 - \frac{x^2}{2R^2} \right) - 2R + \frac{2x^2}{R} \right]$$

$$c = \frac{1}{H \cos \alpha} \left[e_A \left(1 - \frac{x^2}{2R^2} \right) + \frac{x^2}{R} \right]$$

$$c = C \left(1 - \frac{x^2}{2R^2} \right) + \frac{x^2}{RH \cos a} = C \left(1 - \frac{x^2}{2R^2} + \frac{x^2}{Re_A} \right)$$

or:

$$c = C \left(1 + \frac{x^2}{\frac{D^2 e_A}{2(D - e_A)}} \right) \quad (10.55)$$

an expression easier to calculate than eqn. (10.54), and giving values practically identical in the region close to MM' , which moreover is the only one of interest.

This gives an excellent approximation for values of $x < R/2$. It is all the more admissible since the error which it involves becomes entirely negligible in regions far removed from the axial plane OO' , where it diverges most from the correct values, the corresponding pressures being infinitesimal compared with those obtaining in the region of the axial plane.

Substituting in eqn. (10.50), we have the pressure p at section PP' :

$$P = \frac{88}{(10 c)^6} = \frac{88}{(10 C)^6 \left[1 + \frac{2x^2 (D - e_A)}{D^2 e_A} \right]^6} \quad \text{(metric units)} \quad (10.56)$$

We show (Fig. 10.20) the graph of a mill operating under the following conditions:

- D = diameter of rollers: 1,065 mm (42 in.)
- H = thickness of bagasse layer at entry to rollers: 344 mm (13½ in.)
- K = working distance between rollers: 28 mm (1⅒ in.)
- q = fibre loading: 0.168 kg/dm² (3.5 lb./sq.ft.)

The pressure graph is that marked $r = 1$ on Fig. 10.20. These conditions correspond to

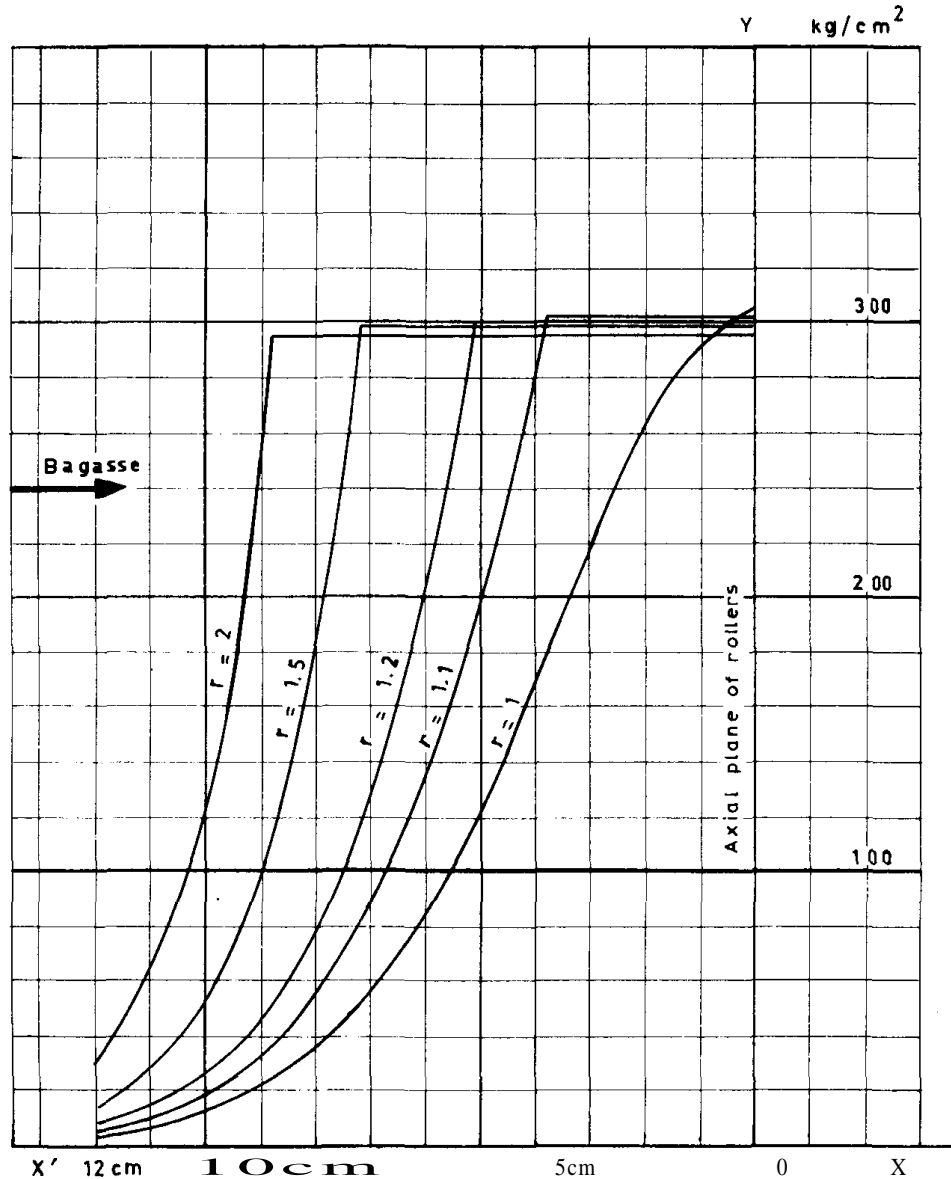


Fig. 10.20. Pressure between the rollers as a function of re-absorption coefficient r .

an s.r.h.p. (*cf.* p. 130) of 13.6 t/dm² (124 t./sq.ft.), which is low, but necessary to envisage zero extrusion.

The correction based on the tests by Jenkins consists of increasing the pressures in the ratio 88/70; this relates the static pressure to the total dynamic pressure in a mill, and arises partly from the resistance to flow of the juices in the closed cylinder of the static experiment. The angle a of which the cosine appears in the denominator of the fraction in eqn. (10.52) also plays a part: if it were the only cause, we would be able to obtain a rough idea of the practical value by calculating:

$$\cos^6 a = \frac{70}{88} = 0.79545 \quad \cos a = 0.96258 \quad a = 15 - 16^\circ$$

This angle a corresponds to the moment where there is no air remaining in the bagasse layer (density of no-void bagasse). It is obviously much lower than the apparent angle of contact.

The pressures developed at 10, 5, 3, 2 and 1 cm from the axial plane may be read off from the ordinates for the corresponding distances measured from the axial plane. It will be observed that practically all the pressure is exerted in the 5 cm (2 in.) preceding the axial plane.

The area included between the curve and the x axis represents the total pressure (r.h.p.) exerted by the top roller.

Delivery side. It is difficult to calculate and even to estimate the pressure on the delivery side, beyond the axial plane. The pressure curves obtained by Murry¹⁴ would give 15–40% of the pressure on the feed side. If we accept a mean value of 25%, it will be necessary to multiply the result for the feed side (*i.e.* before the axial plane) by 1.25 to give the total pressure. It is probable that this factor diminishes in proportion as the re-absorption increases: it may, for example, drop from 1.30 to 1.20 when the re-absorption factor increases from 1 to 1.25. In the absence of precise data on this point, we shall assume in all cases a value of 1.25 independent of the re-absorption factor.

Calculation of the resultant reaction (no re-absorption)

The r.h.p. determines the compression of the bagasse, this compression being such that the sum of the pressures in each section PP' corresponds to the r.h.p. As the calculation is rather long and complex we shall give it for simplicity in the metric system only.

As we have adopted the kg/cm² as unit of pressure, let us consider a section of thickness 1 cm measured along the length of the cylinder (like a slice of sausage). Referring this to Fig. 10.19, and plotting as ordinate (as we have just done in the previous paragraph) the pressure at each point P , the sum of all these pressures will represent the total reaction exerted by the bagasse on the roller, a reaction equal and opposite to the pressure exerted by the roller on the bagasse.

Since the relationship is irreversible on the delivery side, we shall limit ourselves for the present to the left-hand side of the figure from the entry of the bagasse up to its passage through the axial plane.

The total reaction or the total pressure p relative to the 1-cm section under consideration has the value

$$F_1 = \int_0^X p \, dx \quad (10.57)$$

p = pressure at the point P on an element of area dx (of length dx and depth 1 cm).
We have (in metric units):

$$\int_0^X p \, dx = \frac{88}{10^6} \int_0^X \frac{dx}{C^6 \left[1 + \frac{2x^2(D - e_A)}{D^2 e_A} \right]^6} \quad (10.58)$$

Value of X . We require the abscissa of the point corresponding to the entry of the bagasse. Equation (10.53) gives, for $c = 1$:

$$1 = C \quad H \quad \frac{2\sqrt{R^2 - X^2}}{H}$$

whence:

$$X = \sqrt{\frac{H(1-C)}{2} \left[D - \frac{H(1-C)}{2} \right]} \quad (10.59)$$

Calculation of the integral. Let

$$E = \frac{D^2 e_A}{2(D - e_A)} \quad z = \frac{x}{\sqrt{E}} \quad (10.60)$$

We have:

$$dz = \frac{dx}{\sqrt{E}}$$

and:

$$F_1 = \frac{88}{(10^6 C)^6} \int_0^X \frac{dx}{\left(1 + \frac{x^2}{E}\right)^6} = \frac{88\sqrt{E}}{(10^6 C)^6} \int_0^X \frac{dz}{(z^2 + 1)^6} \quad (10.61)$$

Let:

$$y = \int_0^X \frac{dz}{(z^2 + 1)^6} \quad (10.62)$$

The calculation of this integral was done in the second edition. It has the value:

$$y = \frac{9.7.5.3}{10.8.6.4.2} \cdot \frac{\pi}{2} = 0.3866$$

Substituting in eqn. (10.61) and increasing by 25% for the delivery side, we have (in metric units):

$$F_1 = 1.25 \times 0.3866 \times \frac{88 \sqrt{VE}}{(10C)^6} = 42.5 \frac{\sqrt{\frac{D^2 e_A}{2(D - e_A)}}}{(10C)^6} \quad (10.63)$$

This expression (F_1) applies to a 1-cm section of the roller. For the full width of the roller, the r.h.p. would be:

$$F = LF_1 = \frac{42.5 LD \sqrt{\frac{e_A}{2(D - e_A)}}}{(10C)^6} = 30 LD \frac{\sqrt{\frac{e_A}{D - e_A}}}{(10C)^6} \quad (10.64)$$

e_A being small relative to D , and this approximation being minor after what has just been applied for the delivery side, we may write:

$$F = \frac{30 LD \sqrt{e_A}}{(10C)^6} \quad \left(F = \frac{427 LD \sqrt{\epsilon_A}}{(10C)^6} \right) \quad (10.65)$$

F = total pressure exerted by the roller on the bagasse (r.h.p.), expressed in kg (lb.)

L - length of the rollers, in cm (in.)

D - mean diameter of the rollers, in cm (in.)

e_A - mean opening between the rollers when operating, in cm (in.)

ϵ_A = specific opening = e_A/D

C - compression of the bagasse in the axial plane of the cylinders = e_A/H .

It should be kept in mind that this equation gives the pressure between two rollers, in the absence of re-absorption.

Expression for the compression C for practical application

Equation (10.65) has been taken from the experiments of Noel Deerr. We have seen, however (p. 147), the weaknesses of these experiments; they apply to a certain condition of cane, and the compression obtained varies with the state of preparation of the cane or the fibre content of the bagasse. The influence of preparation is most evident at the first mill, but its effect is not great when it is a matter of standard preparation (e.g. two sets of knives and a shredder), in spite of variations in the effectiveness of these preparators according to their power usage, clearance, etc. On the other hand, the effect of fibre is very great, and this is clearly seen if eqn. (10.65) is applied from the first to the final mill of a tandem. By doing this for a number of tandems, and correcting in each case for re-absorption (since we are still considering the case of no re-absorption, where $r = 1$), it may be shown that the correction to be applied to eqn. (10.65) (taken from the experiments of Noel Deerr) to adapt it to the dynamic conditions of mills, consists of substituting for $(10C)$ the expression:

$$\frac{\epsilon_A d f'}{V_0}$$

and to modify consequently the coefficient of 30. On the other hand, it is not the force F which is of interest: it is easily determined. What is of interest is the specific opening when operating at ϵ_A (or the opening itself, e_A), which we cannot measure precisely. We profit from this to anticipate the following calculations and again alter the coefficient to obtain directly the delivery opening of a three-roller mill in operation, taking into account the normal re-absorption in mill tandems which serves as a basis. We thus write the equation in the form:

$$\epsilon_A^{5.5} = 0.34 \times 10^{-6} \cdot \left(\frac{V_0}{df'} \right)^6 \quad (10.66)$$

- ϵ_A = specific delivery opening of the mill = $\frac{e_A}{D}$
 L = width of the rollers, in dm
 D = mean diameter of the rollers, in dm
 F = total hydraulic load on the top roller, in kg
 V_0 = no-void volume of cane or bagasse, in dm^3/kg
 d = density of bagasse under pressure, in kg/dm^3
 f' = fibre of bagasse after pressure, relative to unity.

We do not give the formula in British units since this leads to prohibitive and unwieldy powers of the terms involved.

This calculation is equally precise as that given later (p. 208) for mill settings, and which dispenses with the figures for density of material. It has the advantage over that formula of taking into account the load F the influence of which, acting only to the root of 5.5, has been neglected for mill settings, but it has the disadvantage of bringing in unusual powers of quantities (6 and 5.5) and a number of decimals, which lead to risks of computational error.

Maximal pressure and loading on the roller. If we apply eqn. (10.50) to the delivery opening, it gives us the maximal pressure p_M at that point. Taking eqns. (10.50) and (10.65), and eliminating the compression C , we obtain the maximal pressure p_M :

$$p_M = \frac{3F}{L \sqrt{D e_A}} \approx \frac{3F}{LD \sqrt{\epsilon_A}} \quad (10.67)$$

- p_M = maximal pressure in the axial plane, in kg/cm^2 (p.s.i.)
 F = total load on the roller in the axial plane, in kg (lb.)
 L = length of rollers, in cm (in.)
 D = mean diameter of the rollers, in cm (in.)
 e_A = delivery opening, in cm (in.)
 $\epsilon_A = e_A/D$ when $r = 1$.

Since it applies only to the case of zero re-absorption, which does not occur in practice, this formula (10.67), being purely theoretical, would not serve for any practical determination.

Influence of re-absorption

We shall now consider the general case, where re-absorption takes place and the re-absorption coefficient r is greater than 1.

We shall consider two rollers crushing bagasse. To facilitate our reasoning, we shall assume that it is a case of the same two rollers as we have studied (p. 152), treating the same bagasse at the same fibre loading. Commencing from a condition of no re-absorption, we assume that conditions are altered so as to increase the re-absorption coefficient step by step without changing the fibre loading; in other words, that we progressively reduce the delivery opening. This may be done by using a closer setting in the case of fixed rollers or by increasing the hydraulic load where hydraulics are in use. Now, considering the instants at which the re-absorption coefficient reaches the values $r = 1.1, 1.2, 1.3, 1.5$ and 2 , we may calculate for each case the position of the neutral plane, by applying eqn. (10.32) (*cf.* p. 141). We assume now, as Murry¹⁵ has concluded from measurements of actual pressure, that the pressure increases (as in the case of no re-absorption) between the point of first contact and the neutral plane, that it then remains practically constant between the neutral plane and the axial plane,

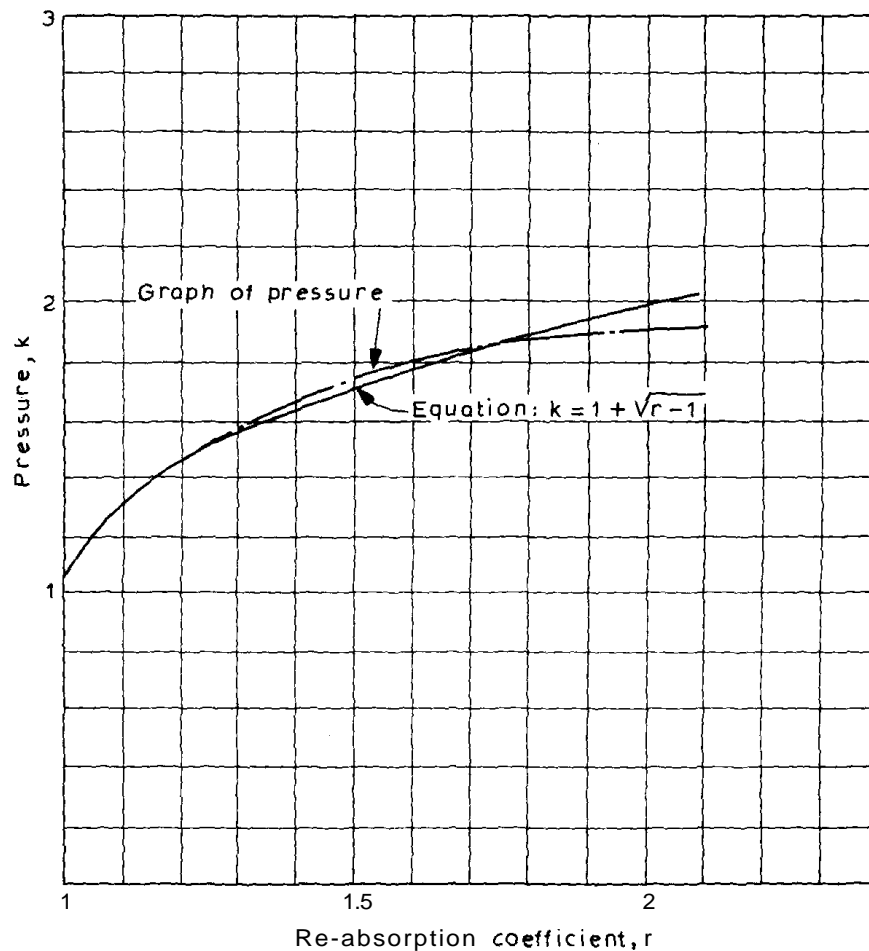


Fig. 10.21. Variation of the roller load as function of re-absorption coefficient r .

and finally drops soon after the axial plane, exerting in all cases after the axial plane the same fraction of the total pressure.

We obtain thus the graphs marked: $r = 1.1, 1.2$, etc. in Fig. 10.20. The area included between each of these graphs and the x and y axes thus gives a relative value for the total pressure in each case. Calculating these areas, and transferring their relative values (taking as unit value that for no re-absorption) to the graph of Fig. 10.21, we obtain the graph shown as a full line. We find that this curve for normal re-absorption is very close to that shown dotted:

$$k = 1 + \sqrt{r - 1} \quad (10.68)$$

We may deduce from this that when re-absorption varies (other conditions remaining the same), the total pressure increases, relative to that for no re-absorption, in the proportion given by eqn. (10.68).

General value for total pressure

Considering the case of re-absorption, we see that the expression for total pressure becomes:

$$F = 1,300 LD \sqrt{\epsilon_A} \left(\frac{q d_B}{re_A f'} \right)^6 (1 + \sqrt{r - 1}) \quad (\text{metric units}) \quad (10.69)$$

The relative opening ϵ_A occurs in effect at its real value, as is shown by the calculation on pp. 154–155, whereas it is the opening re_A which occurs within the brackets at the 6th power. The same calculation as before applies here, re_A having the value $e_N = V_B/S$ (cf. eqns. (10.12), (10.18), (10.26), p. 153). We have finally:

$$F = 1,300 LD \sqrt{\epsilon_A} \left(\frac{q d_B}{e_N f'} \right)^6 (1 + \sqrt{r - 1}) \quad (10.70)$$

$$\left(F = 0.16 LD \sqrt{\epsilon_A} \left(\frac{q d_B}{100 e_N f'} \right)^6 (1 + \sqrt{r - 1}) \right)$$

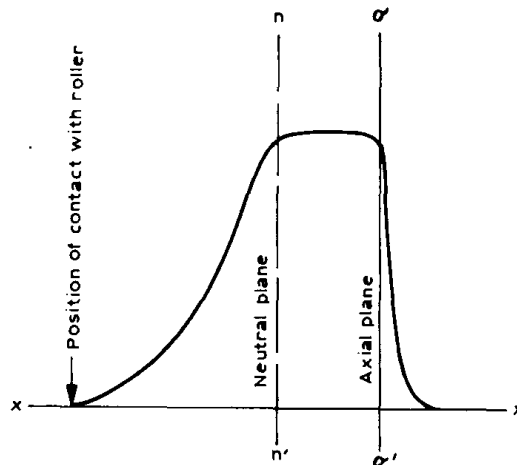


Fig. 10.22. Graph of pressure in the mill.

or (for high pressures):

$$F = 1,300 LD V \epsilon_A d_B^{1.2} (1 + \sqrt{r} - 1) \quad (\text{metric units}) \quad (10.71)$$

Empirical relations between load and maximum pressure

It is interesting to compare certain empirical values, obtained by experiment, with the preceding values deduced by calculation, such as that given by eqn. (10.67).

Holt¹⁶ has given a relation between the total pressure F exerted on the roller and the maximal pressure p_M applied to the bagasse in its passage between the neutral plane and the axial plane (Fig. 10.22):

$$p_M = \frac{344 F}{LD (\alpha + 3.5 v)} \quad (10.72)$$

p_M = maximal pressure applied to the bagasse in its passage through the opening in kg/dm² (p.s.i.)

F = axial component of the total hydraulic load acting on the top roller, in kg (lb.)

L = length of rollers, in dm (in.)

D = diameter of roller, in dm (in.)

α = angle between the axial plane and the position at which the bagasse meets the roller surface, in degrees

v = angle between the axial plane and the neutral plane, in degrees.

Normal values of re-absorption

Re-absorption plays an important role in the mill, and the re-absorption coefficient is determined or calculated only in a small number of factories; however, it is always possible to obtain a relatively accurate idea of its value, since it depends mainly on two factors which are much more readily known: speed and fibre index.

Murry¹⁷ gives, for a compression ratio of 3, the regression line:

$$r = 1.06 + 0.017 v \quad (r = 1.06 + 0.0052 v) \quad (10.73)$$

r = re-absorption factor

v = peripheral speed of the rollers, in m/min (ft./min).

Similarly¹⁸, for a speed of 9 m/min (30 ft./min), he gives the regression line:

$$r = 0.0912 c_A + 0.87 \quad (10.74)$$

which we shall write, making use of eqn. (10.44), and knowing that the fibre content of the cane in that experiment was 12.8:

$$c_A = \frac{\phi}{f d_c} = \frac{\phi}{0.141} \quad \left(c_A = \frac{\phi}{f d_c} - \frac{\phi}{9} \right) \quad (10.75)$$

$$r = 0.65 \phi + 0.87 \quad (r = 0.01 \phi + 0.87)$$

The two equations (10.73) and (10.75) may be combined in one:

$$r = 0.75 + 0.017 v + 0.65 \phi \quad (r = 0.75 + 0.0052 v + 0.01 \phi) \quad (10.76)$$

which gives r as a function of the two main factors determining this coefficient (see *Comment 1* below).

Example. For a mill with peripheral speed $v = 10$ m/min and fibre index at entry $\phi_e = 0.3$ kg/dm³, at delivery $\phi_s = 0.6$ kg/dm³, the re-absorption factor would be:

$$\begin{aligned} \text{At entry:} \quad r_e &= 0.75 + (0.017 \times 10) + (0.65 \times 0.3) = 1.11 \\ \text{At delivery:} \quad r_s &= 0.75 + (0.017 \times 10) + (0.65 \times 0.6) = 1.31 \end{aligned}$$

Comment 1. The two original formulae (10.73) and (10.74) being established for a first mill, the derived formula (10.76) is probably correct for first mills, but there is reason to suppose that the constant term 0.75 would drop for other mills in the tandem, and would fall to the neighbourhood of 0.60 for a final mill. In the absence of precise experimental data, we shall assume that this term has a value:

$$\begin{aligned} &0.75 \text{ for a first mill} \\ &0.65 \text{ for a second mill} \\ &0.60 \text{ for following mills} \end{aligned} \quad (10.77)$$

Comment 2. The foregoing equation would obviously give only an approximate figure. It does not take into account the effect of a third factor, which cannot be neglected: the state of preparation of the material entering the mill.

Ratio of pressures at feed and delivery rollers

It is useful to know the relative pressures exerted on the bagasse at entry and delivery rollers of a mill; in other words, the components at the feed and delivery rollers of the total load applied to the mill.

This ratio obviously varies greatly according to the condition of the bagasse entering the mill, and particularly the settings adopted and the fibre loading at any instant (relative to that for which the setting has been established). To obtain some idea of the figures, we shall take mean values. We recommend later (Table 13.6) for a third mill, a ratio of 1.8 between feed and delivery openings when operating.

We shall apply eqn. (10.69) and calculate the expressions involved in it: the terms $1,300 LD$ and q are the same at entry and delivery; using the ratio of settings $m = 1.8$, the values of ϵ_A and, to a close approximation, e_N at feed and delivery are in the ratio of 1.8. We require further f' , d_B and r .

(a) f' . Assuming a fibre in cane of 0.14, an imbibition $w = 0.36$, a juice extraction of $\frac{2}{3}$ at the feed and 3 at the delivery roller, giving a total $j = 0.36$ and leaving a bagasse $b = 0.30$; the fibre content of bagasse leaving the feed opening, and passing over the trashplate, will be:

$$f'_E = \frac{0.14}{0.30 + 0.12} = 0.333$$

and the fibre content of bagasse leaving the mill:

$$f'_S = \frac{0.14}{0.30} = 0.467$$

(b) d_B . The corresponding densities will be (cf. p. 135):

$$V_0(E) = \frac{1}{1.03 \times 1.01} - \left(\frac{1.20}{1.03 \times 1.01} - 0.86 \right) 0.333 = 0.864; d_B(E) = \frac{1}{0.864} = 1.158$$

$$V_0(S) = \frac{1}{1.03 \times 1.01} - \left(\frac{1.20}{1.03 \times 1.01} - 0.86 \right) 0.467 = 0.824; d_B(S) = \frac{1}{0.824} = 1.213$$

(c) r . We shall assume respective re-absorption coefficients as (cf. p. 160):

$$r_E = 1.11 \quad r_S = 1.31$$

We have then:

$$\frac{F_S}{F_E} = \sqrt{\frac{1}{1.8}} \times \left(\frac{1.213 \times 1.8 \times 0.333}{1.158 \times 1 \times 0.467} \right)^6 \left(\frac{1 + \sqrt{0.31}}{1 + \sqrt{0.11}} \right) = 0.745 \times 5.9 \times 1.169 = 5.1$$

Depending on the setting, the crushing rate, and the speed, this ratio may vary from 2 to 8, generally between 2 and 4. It is lower in the first mill, where the ratio between the two fibre contents is higher and the material entering the mill is still relatively coarse, than in a later mill, where the ratio between the two fibre contents is lower.

In a fairly accurate experiment, made with strain gauges and a cathode-ray oscillograph on the first mill at la Mare (Bourbon), the ratio F_S/F_E was found to vary between 1 and 2.5, the lower values corresponding to lower tonnages. In exceptional cases, the ratio fell below 1. In a similar experiment at another factory in Reunion, the ratio varied from 3.1 to 5.7 for the first mill, and 3.7 to 4.0 for the final mill.

Crawford¹⁹, in Australia, concluded that the ratio of delivery and feed roller pressures ranged from 2 to 4 for the first mill, from 4 to 5 for a third mill. These values obviously depend mainly on the ratio m between the feed and delivery openings of the mill while operating. They also depend on the place of the mill in the tandem.

Polygon of forces

To demonstrate better the relative importance of the various reactions acting on the top roller of a mill in operation, let us set them out on a diagram (Fig. 10.23).

We shall adopt the ratio of 5 between delivery and entry pressures, and trace the polygon

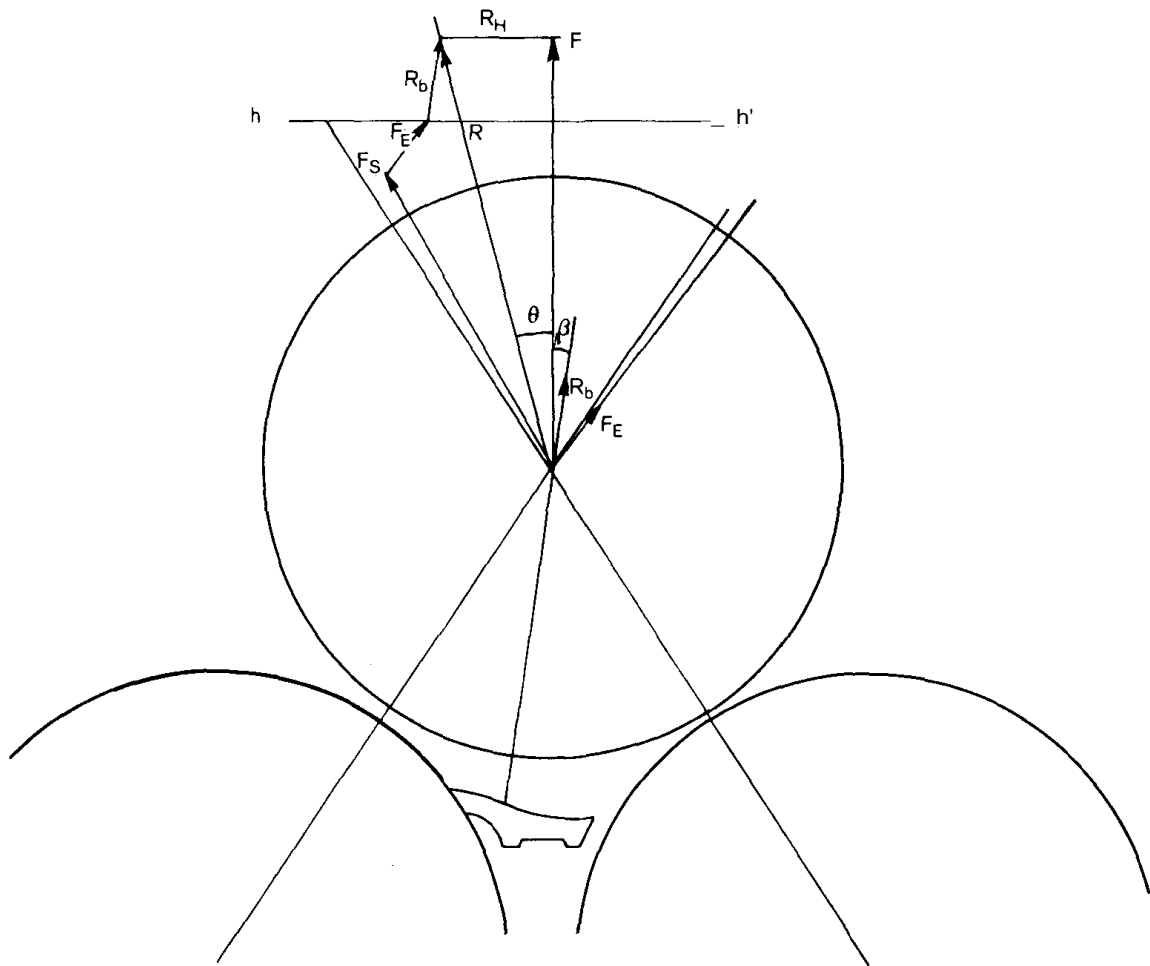


Fig. 10.23. Polygon of forces in the mill.

of forces acting on the top roller. Their directions are easily ascertained; the t.h.p., F , is vertical. The reaction of the trashplate, R_b , makes a small angle β with the vertical, of some 10° .

For F_S and F_E we see from Fig. 10.14 that the centre of pressure (that is, the centre of gravity of the area included between the curve and the circumference of the roller) occurs not in the axial plane but at $2-3^\circ$ ahead of it. By taking:

$$F_E = \frac{F_S}{5} \text{ and } R_b = 0.25 F$$

we obtain thus the polygon indicated in Fig. 10.23.

R is the resultant of the three reactions F_S , F_E and R_b ; and the projection of R on the vertical must be equal to P . It is seen that this resultant will act at about 15° ahead (on the feed side) of the vertical. This is the angle generally adopted by manufacturers who build mills with inclined headstocks and asymmetric housings.

We may comment that, by projecting on the vertical:

$$(F_S + F_E) \cos \frac{\alpha}{2} + R_b \cos 0 = F$$

Putting: $F_E = 0.2 F_S$, $R_b = 0.25 F$ and commenting that $\cos \beta \approx 1$, and $\cos a/2 = 0.8$, we have:

$$(1.2 F_S \times 0.8) + 0.25 F = F$$

Hence:

$$F_S \approx 0.8 F \tag{10.78}$$

which shows that the r.h.p. on the delivery side, F_S , is generally of the same order of magnitude as the t.h.p. acting on the top roller, F , but slightly lower.

This calculation is less accurate, if it is assumed that the reactions are in the axial planes. If we note on the diagram that the reaction R_b should remain on the horizontal line hh' , we shall have immediately, from the graph, all the values assumed by F_S , for each value of

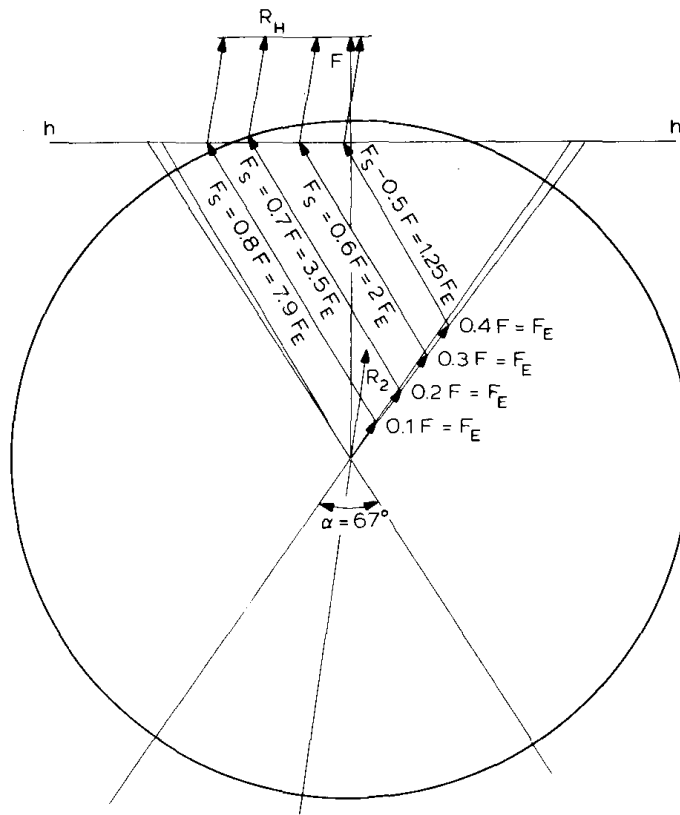


Fig. 10.24. Values of F_S relative to F_E .

F_E (Fig. 10.24). For a mill of angle $a = 67^\circ$ (and the result will hardly vary for a different construction angle) we find, in putting:

$$\phi_E = \frac{F_E}{F} \text{ and } \phi_S = \frac{F_S}{F}$$

that:

$$\begin{aligned}\phi_E &= 1 - 1.07 \phi_S \\ \phi_S &= 0.93 - 0.93 \phi_E\end{aligned}\tag{10.79}$$

This gives the corresponding values shown in Table 10.4.

TABLE 10.4

When $F_E = 0.10 F$	$F_S = 0.8 F = 8 F_E$	$F_E + F_S = 0.9 F$
When $F_E = 0.20 F$	$F_S = 0.7 F = 3.5 F_E$	$F_E + F_S = 0.9 F$
When $F_E = 0.30 F$	$F_S = 0.6 F = 2 F_E$	$F_E + F_S = 0.9 F$
When $F_E = 0.40 F$	$F_S = 0.5 F = 1.25 F_E$	$F_E + F_S = 0.9 F$

F_S then remains generally between 0.6 and 0.8 of F , most often about 0.7 F .

We may comment that the sum of $F_E + F_S$ is constant and equal to 0.9 F , and that the reaction R_H on the housing vanishes when $F_E = 0.4 F$.

Friction of brasses in housing

With housings of standard design, with vertical headstocks, it is immediately apparent from Fig. 10.23 that the inclination of the resultant R involves the existence of an important horizontal reaction R_H on the lateral face of the headstock, on the feed side:

$$R_H = F \tan \theta \approx 0.15-0.25 F\tag{10.80}$$

or approximately 80 tonnes when $F = 400$ tonnes.

The whole of this reaction acts between the feed-side faces of the upper brasses and the housing. The coefficient of friction between bronze and steel, without lubrication, is approximately 0.18–0.20, and this friction combines with that of the leather or metallo-plastic packing in the hydraulic cylinder to oppose the sliding movements of the bearing and the hydraulic piston, that is, to oppose the lift of the top roller.

The friction of brass on steel falls to about 0.10 if the surfaces in contact are lubricated. It is of advantage to ensure that this lubrication is effective in order to facilitate the proper functioning of the hydraulic pressure.

Certain manufacturers now use plates of synthetic material of high resistance to wear and very low coefficient of friction, placed between bearing and housing.

In the same way, and for the same reason, housings with inclined headstocks offer a great advantage, which may be further accentuated by lubrication or provision of a plate of synthetic material. If the inclination of the headstocks corresponds to the value of 15° generally adopted, it will be sufficient to provide lubrication, rollers, or plate on the feed-side face of the brass. If the inclination is 20° or greater, it will be of advantage to so equip both faces, feed and delivery sides, as the reaction can then change from one side to the other according to settings and conditions.

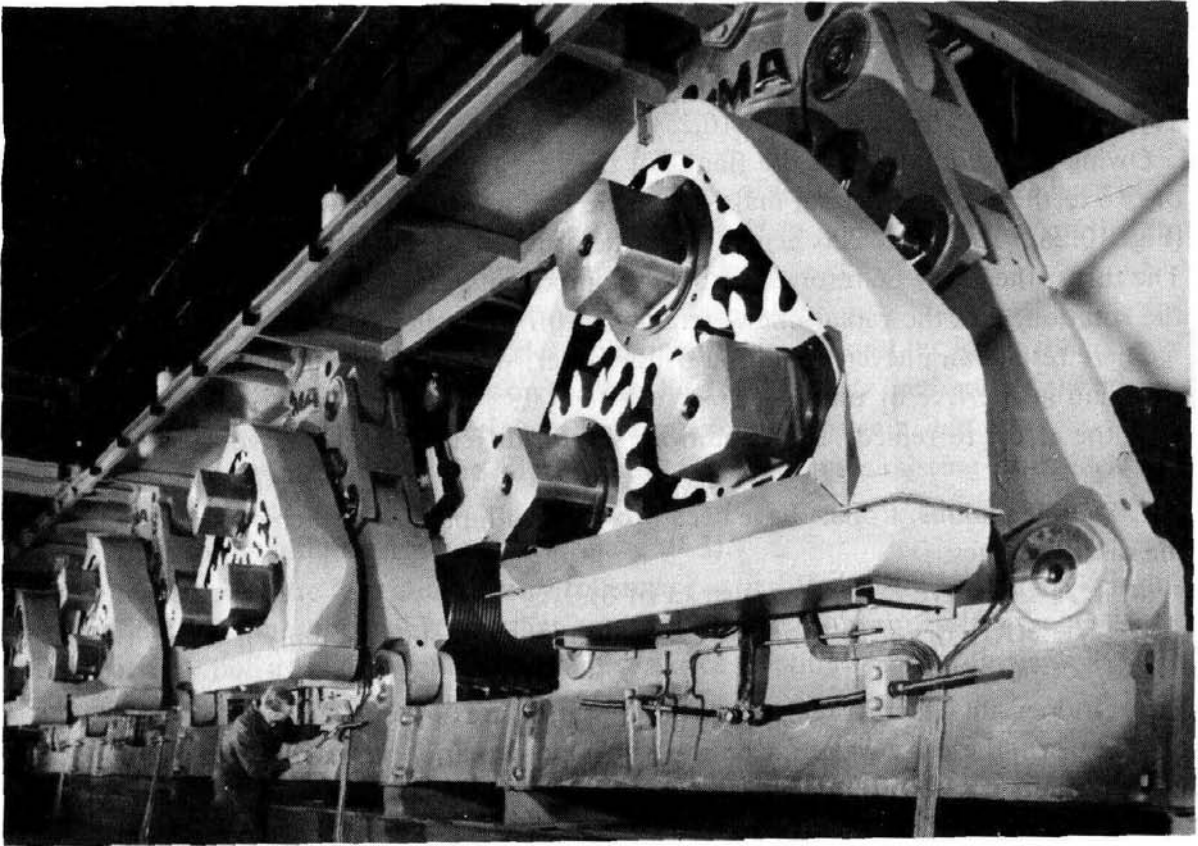


Fig. 10.25 Meshing of the three roller pinions

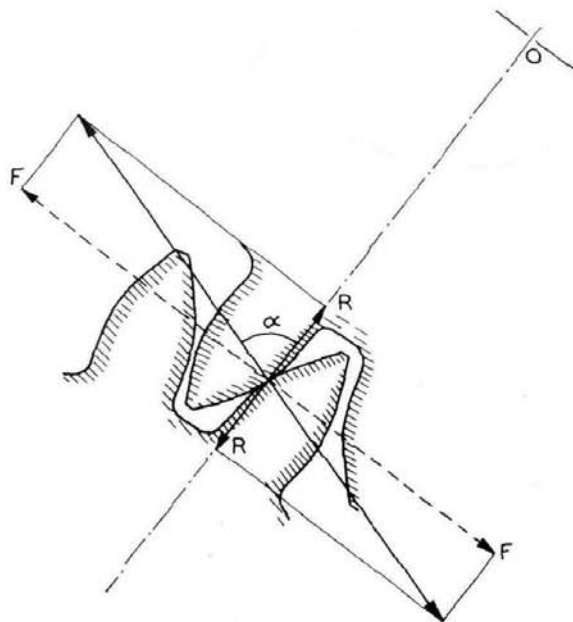


Fig. 10.26 Pinion reaction

Pinion reaction

In a mill, the power from the prime mover is generally transmitted by the gearing to the top roller. It is the latter which drives the two lower rollers by means of a group of 3 pinions keyed to the 3 roller shafts (Fig. 10.25).

Owing to the obliquity of the flanks of the gear teeth, the thrust at the point of contact of two teeth is not normal but makes an angle α of less than 90° with the radius at that point (Fig. 10.26). This angle α is generally of the order of 65° . Fives Cail-Babcock reckon 70° . The thrust then may be resolved into two components, the normal component F producing the rotation, and the radial one R tending to lift the roller.

There results an uneven condition of the roller, which will lift more readily at the pinion end than at the free or pintle end.

In the effort to remedy this condition, several different devices have been adopted:

1. Double pinions. Rollers have been provided with pinions at each end. This solution has the disadvantages:

- (a) It is expensive (6 pinions instead of 3).
- (b) The effect is not to suppress the pinion reaction, but to apply it at both ends. This would not be serious, but:
- (c) It does not ensure a real equalisation of pressure at the two sides. So long as the teeth are not worn, there is always one pinion which momentarily carries more load than the other, and produces a reaction.

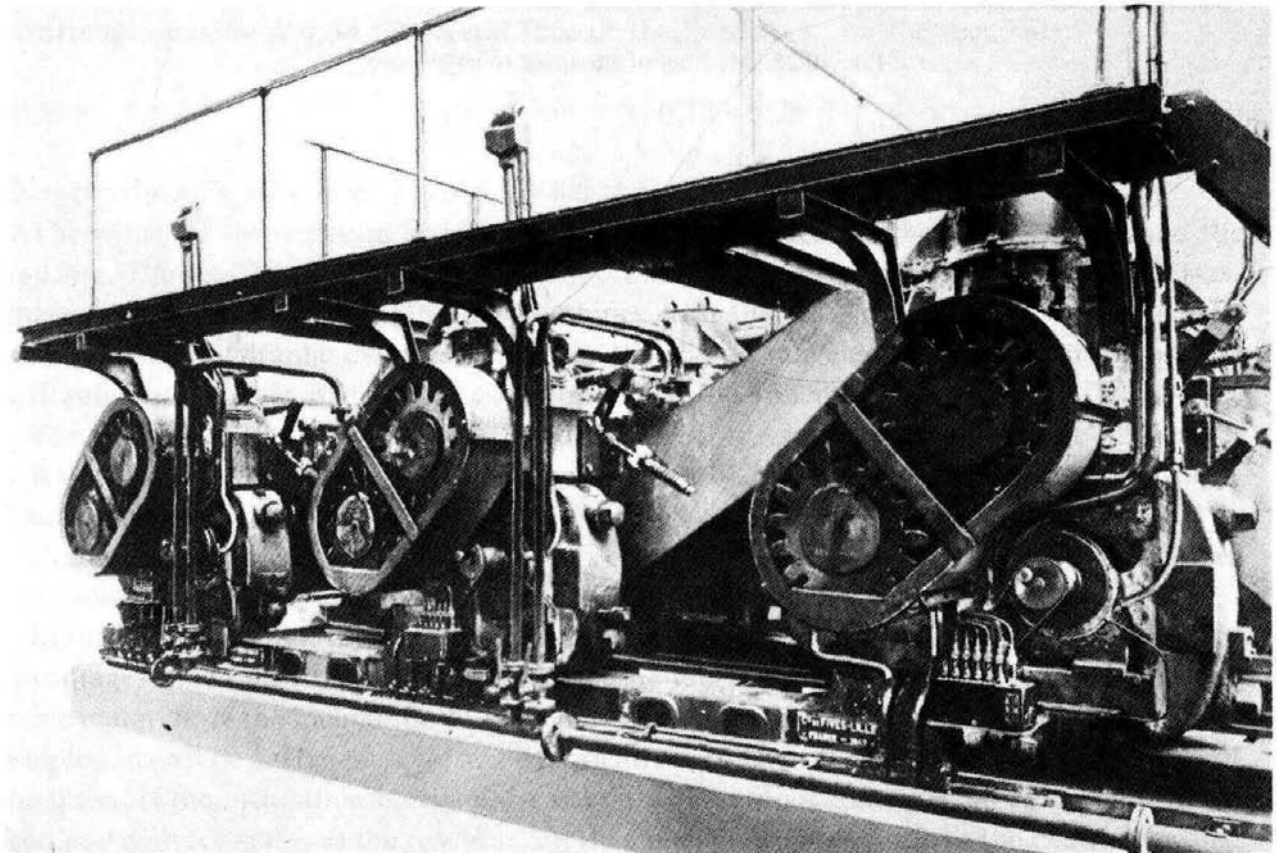


Fig. 10.27. Mills with alternate pinions.

(d) Even when the teeth are worn, the slightest difference in the thickness of the bagasse layer on the left and on the right at once destroys the equilibrium, alters the positions of contact of the teeth, loses contact at the side with the greater lift, and causes a reaction at the opposite side.

2. Difference in diameters of hydraulic pistons. Pistons of larger diameter have been placed on the pinion side, of such a size that their area is 10–20% greater than that on the opposite side. This is a barbarous solution, and most inadequate.

3. Alternate pinions. The top roller carries a pinion at each end, and the lower rollers have their pinions, one on the right, the other on the left (Fig. 10.27). This method had the advantage of permitting a substantial reduction in the top angle of the mill, but:

(a) It is scarcely elegant: it gives the impression that something is missing from each side of the mill.

(b) It requires one additional pinion (4 in place of 3).

(c) The reactions of the feed and delivery rollers are very different (*cf.* p. 161). Their difference is therefore not appreciably less than their sum ($6 - 1$ is not much less than $6 + 1$), and one corrects but a small fraction of the divergence existing in an ordinary mill. It is scarcely worth the trouble.

4. Separate sets of pinions. This solution consists of driving each roller by a separate tail-bar, the set of three pinions being arranged in the gear train in a special compartment called the "pinion housing" (Fig. 10.28).

It has some disadvantages:

(a) It is expensive (only 3 pinions, but 4 couplings, 2 tail-bars and the pinion housing extra). However, the individual tail-bars and couplings are lighter than the single tail-bar and the 2 couplings of the ordinary mill.

(b) It increases by 0.3–0.6 m (1–2 ft.) the overall width occupied by the gearing.

(c) It adds to the installation by 4 plumber blocks and lubricators per mill; hence a slight increase in consumption of power and oil. The increase in power is compensated many times by the elimination of pinion reaction at the rollers, but the slight increase in oil consumption remains.

(d) The usual length of the tail-bars does not allow sufficient range of movement at the couplings to permit of all possible adjustments of the lower rollers (cases of extreme settings, or of very worn rollers); in this case it is necessary to replace the ordinary couplings by more flexible devices (*e.g.* cardan shafts, *cf.* p. 256).

But it has some advantages:

(a) It completely solves the problem of pinion reaction. The top roller, entirely free, floats at ease.

(b) Like the preceding solution, it permits the top angle of the mill to be reduced to a minimum.

In fact, the pinions, being fixed in their housing, may be designed with the ordinary form of gear tooth, whereas pinions mounted on the rollers must be able to function with the distance between their centres varying greatly according to the settings adopted, the mean

diameter of rollers more or less worn, and the lift of the top roller in operation. In these conditions, one is compelled to provide these pinions with very long teeth, so that the pitch diameter is at $\frac{1}{3}$ of the length of the teeth, and to make the number of teeth as small as possible. Hence, the overall diameter of the pinion greatly exceeds the mean diameter of the rollers, necessitating a high minimum distance between the axes of the two bottom rollers, in order to allow clearance between their pinions.

With the separate set of pinions, on the other hand, one can adopt a normal tooth-form (pitch-circle diameter half-way up the teeth), a greater number of teeth (23 instead of 17, for example) and a very small clearance between the ends of the teeth of the two lower pinions. Hence it is possible to bring the two bottom rollers close together and in consequence to reduce the top angle of the mill to 67° , a value which could not be attained for rollers fitted with ordinary pinions, except by means of alternate pinions.

(c) When the rollers are replaced or re-shelled, it is no longer necessary to remove (or to re-key) the pinions, always a long and disagreeable task. The shafts are shorter, lighter, and cheaper.

(d) The pinions, being fixed, and working in much more favourable conditions, do not

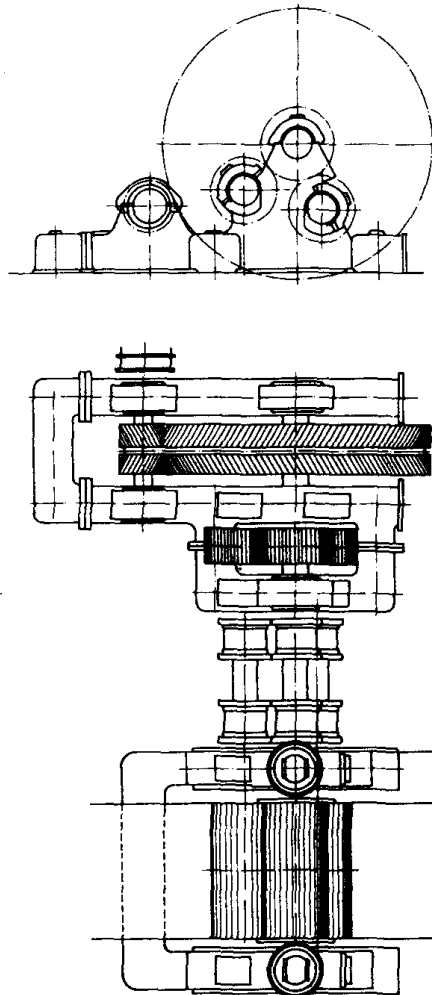


Fig. 10.28. Nest of pinions (Fives Cail - Babcock).

wear so quickly. Above all, one is not obliged to mutilate them by turning them down in order to reduce the length of the teeth and enable the mill to be tightened up, when the rollers have become worn.

(e) The installation cost is progressively recovered, when it comes to buying replacement rollers.

However, largely on account of the expense, this device is not widely used.

5. Individual accumulators. The limited adoption of separate pinions and tail-shafts is due probably to the fact that the general adoption of individual air-oil accumulators today allows the pressures on the two sides of the mill to be readily balanced. Thus the main disadvantages of pinion reaction may be completely avoided; and this is the solution generally adopted at the present day. It is not as complete as the separate pinions, but is simpler and less expensive. The latter is now regarded for this reason as a luxury; we consider, however, that with a tandem which is expected to last 20 years or longer, this luxury would become a paying proposition.

Value and distribution of pinion reaction. Consider the case of a mill where there is no correction for pinion reaction, *i.e.* which has no devices such as double pinions, unequal pistons or the other devices as just described. This is in fact the more general case. We require to know the value of the pinion reaction, and how it can be compensated by means of individual accumulators.

It will be seen (p. 234) that the power taken by a mill is of the order of $P = 0.15 - 0.23 FnD$ ($P = 0.65 - 1.0 FnD$), F being the hydraulic load in tonnes, n the speed in r.p.m. and D the mean diameter of the top roller in m (ft.), the power P being expressed in kW (h.p.).

It may be assumed that the power is distributed in the ratio of 50% to the top roller and 50% on the two lower rollers. Although the latter do different work and have an appreciably higher pressure at the delivery roller, tests have shown that the power is distributed approximately equally between them, *i.e.* in overall proportions of 25% and 25%. Taking a mean power $P = 0.25 FnD$ h.p., and giving F a normal value $F = 250 LD$ tonnes, $P = 0.25 \times 250 LD \times nD = 62.5 nLD^2$ h.p., or $P = 4.687 nLD^2$ kg m/s.

The corresponding couple exerted on the roller by the driving motor is C :

$$P = C\omega = \frac{2\pi nC}{60} = 0.105 Cn$$

whence:

$$C = \frac{P}{0.105 n} \text{ kg m}$$

The corresponding force at this couple is:

$$f^* = \frac{C}{r} \sim \frac{2C}{D} = \frac{2P}{0.105 nD} \text{ kg}$$

The loads exerted on the lower rollers will be:

$$f_1 = f_2 = \frac{0.25 \times 2P}{0.105 nD} = \frac{0.5 \times 4.687 nLD^2}{0.105 nD} = 22,380 LD \text{ kg}$$

If the angle of reaction is 20° and the angle of the mill is 80° , these two forces f_1 and f_2 , reduced to their radial components (Fig. 10.24), relative to the centre of the top roller, will give a resultant R :

$$\begin{aligned} R &= 22,380 LD \sin 20^\circ \cos 40^\circ \times 2 \\ R &= 44,760 LD \times 0.342 \times 0.766 = 11,726 LD \text{ kg} = 11.726 LD \text{ tonnes} \end{aligned}$$

Thus:

$$\frac{R}{F} = \frac{11.726 LD}{250 LD}$$

$$R = \frac{11.726 LD}{250 LD} F = 0.047 F \text{ tonnes}$$

Distribution of forces. The shaft of the top roller acts as a beam, supported at two points and carrying an overhung load, the reaction R of which is distributed between the two supports, *i.e.* on the two bearings, causing the reactions R_1 and R_2 ; R_1 on the driving side, directed downwards, and R_2 on the far side, directed upwards (Fig. 10.29).

If the distance between the centres of the two bearings is $d = 100$ cm, and in general d_1 is of the order of 23 cm, then d_2 is consequently 123 cm (d_1 being the distance between the centre of the bearing on the driving side and the centre of the pinion). The loads on the bearings are thus given by:

$$R = R_1 - R_2 \quad R_1 d_1 = R_2 d_2$$

whence:

$$R_1 = \frac{R d_2}{d_2 - d_1} = \frac{R d_2}{d} = 1.23 R$$

$$R_2 = \frac{R d_1}{d} = 0.23 R$$

whence:

$$R_1 = 0.047 F \times 1.23 = 0.058 F$$

$$R_2 = -0.047 F \times 0.23 = -0.011 F$$

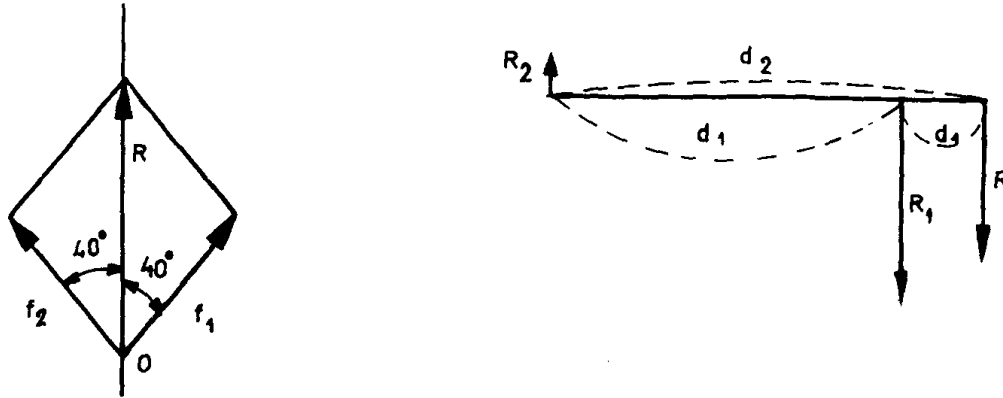


Fig. 10.29. Pressure repartition in the mill housings between drive and pintle sides.

If the load F is to be distributed equally between the two sides of the mill, it will be necessary, to compensate the pinion reaction, to exert forces of:

On the driving side:

$$f_1 = \frac{F}{2} + 0.058 F = 0.558 F$$

On the outer side:

$$f_2 = \frac{F}{2} - 0.011 F = 0.489 F \tag{10.81}$$

If the load is, for example, $F = 460$ tonnes, this will give $f_1 = 257$ tonnes and $f_2 = 225$ tonnes, or approximately 14% extra on the driving side.

Mean values of the t.h.p.

Thes.h.p. (*cf.* p. 130) in mills generally varies from a minimum of 12 t/dm^2 (110 tons/sq.ft.) to a maximum of 40 t/dm^2 (360 tons/sq.ft.). It is mostly maintained between 15 and 30 t/dm^2 (140 and 280 tons/sq.ft.), generally between 20 and 25 (180 and 220).

For the fibre loadings generally employed, we can then estimate a mean value of the t.h.p. as:

$$F = 250 LD \quad (F = 23 LD) \tag{10.82}$$

F = total hydraulic load (t.h.p.), in tonnes

L = length of rollers, in m (ft.)

D = diameter of rollers, in m (ft.).

Thence:

$$\text{s.h.p.} = \frac{2.5 LD}{0.1 LD} = 25 \text{ t/dm}^2 \text{ (230 tons/sq.ft.)} \tag{10.83}$$

Deadweight

The normal value indicated above, $F = 250 LD$, takes no account of the other components of load due to the weight of the top roller and accessory items which add their weight to the hydraulic pressure exerted. This deadweight is more important in the case of mills such as the self-setting mills of Fives Cail – Babcock, where the mill caps and the upper halves of the bearings also exert their weight upon the bagasse, as do the crown wheel, the coupling and part of the tail-bar in all models of mills (except those with the nest of pinions).

However, if the hydraulic pressure varies proportionally to LD , the weight of the roller and the deadweight in general will be proportional to LD^2 . If it is desired to take account of these factors, it may be stated that the pressure exerted on the bagasse is of the order of $(250 + 10 D) LD$ tonnes (L, D in m).

Comparison between crushers and mills. By reason of the distribution of the forces in a mill, it may appear *a priori* difficult to compare the pressure of a crusher with that in the mills. The t.h.p. indeed acts wholly on the bagasse in the crusher, while in the mill it is distributed between the trashplate and the two lower rollers.

Our comment (p. 163) and the calculations which precede it show, however, that for a given s.h.p. and a given t.h.p., the pressure exerted on the bagasse in a mill is slightly lower than, but of the same order as, that in a two-roller crusher.

Variation of extraction as a function of pressure limits

When the hydraulic pressure is increased, starting from a low value, the extraction increases rapidly at first, then, when the s.h.p. reaches about 15 t/dm^2 (140 tons/sq.ft.), more slowly. We show in Fig. 10.30 the changes in extraction in a test carried out at Savanna (Reunion) on a tandem of five mills, the first $870 \times 1,700 \text{ mm}$ ($34 \times 67 \text{ in.}$) and the others $810 \times 1,675 \text{ mm}$ ($32 \times 66 \text{ in.}$), fibre in cane 13.8%, and imbibition 1.6, with changes in hydraulic pressure.

Crawford²² estimates that an s.h.p. of 31.2 t/dm^2 (100 tons/ft. for a 42-in. roller) leaves only a small margin of safety, owing to the internal stresses already existing due to the operation of shrinking the roller shell onto the shaft.

Specific hydraulic pressure

It was estimated in Cuba that the gain in extraction obtained by the use of an s.h.p. greater than $18\text{-}22 \text{ t/dm}^2$ ($165\text{-}200 \text{ tons/sq.ft.}$) (say, above 20 t/dm^2 (180 tons/sq.ft.)) was not worth the expense and wear involved.

The firm of Farrel²⁰, in America, adopts a maximum pressure of 23.5 t/dm^2 (214 tons/sq.ft.) which, in American units, gives slightly more than 2 tons per foot of length and per inch of diameter of roller.

In Australia²¹ it is considered that the gain in extraction beyond $30\text{—}33 \text{ t/dm}^2$ ($270\text{—}300 \text{ tons/sq.ft.}$) becomes very slight, since the power expended increases rapidly, and that these figures form a practical limit which should not be exceeded.

For our part, we consider:

(7) That it is absolutely necessary to go up to 5 t/dm^2 (50 tons/sq.ft.) at the crusher and

16.5 t/dm² (150 tons/sq.ft.) at the mills, but that there is no need to exceed these values if considerations of strength of material or consumption of steam are unfavourable to it (as may be the case in some very old factories).

(2) That there is a very definite advantage in going up to 8 t/dm² (75 tons/sq.ft.) at the crusher and 20 t/dm² (180 tons/sq.ft.) at the mills, if the equipment and the thermal balance of the factory permit.

(3) That there is still an advantage in increasing to 16.5 t/dm² (150 tons/sq.ft.) at the crusher, 25 t/dm² (225 tons/sq.ft.) at the mills other than the last, and 28-30 t/dm² (255 – 275 tons/sq.ft.) at the last mill, if the plant has been designed accordingly and if (as is generally the case) one has a large margin of safety in the consumption of steam at the mills.

Sequence of pressures in the train

Is it desirable to increase the hydraulic pressure from the first mill to the last, or to reduce it?

It is generally the former method which is encountered in practice. It complies with the idea that a mill should press more heavily than the preceding mills in order to extract what they have not been able to extract ahead of it.

But the partisans of the second method point out that it is mainly the imbibition which is effective, much more than very high pressures; and that, to render it effective, it is necessary to prepare the cane to the maximum extent in the earlier units.

Table 10.5 gives some examples of the sequence of pressures in tandems giving very good extraction figures.

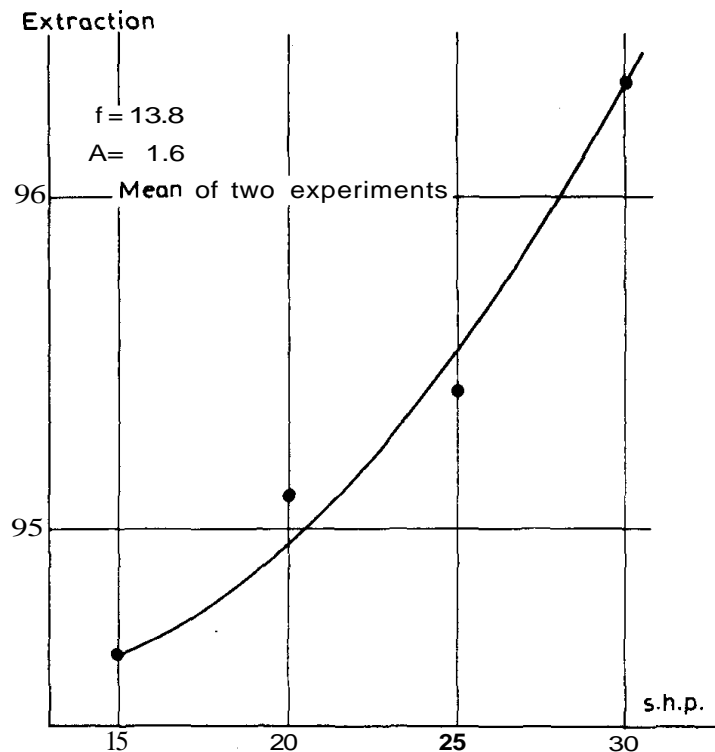


Fig. 10.30. Variation of extraction with s.h.p. (specific hydraulic pressure).

TABLE 10.5

SEQUENCE OF S.H.P. IN VARIOUS TANDEM, IN t/dm² (tons/sq.ft.)

Location	N	N ₁	n	Mill number						
				1	2	3	4	5	6	7
<i>Normal tandems</i>										
S. Africa	18	6	3	22 (261)	20.4 (187)	20.4 (187)	20.3 (186)	20.6 (188)	20.6 (188)	
	21	7	3	23.6 (216)	19.8 (181)	20.9 (187)	20.5 (187)	19.8 (181)	19.5 (178)	24.7 (226)
Mauritius	15	5	1	26.8 (245)	20.1 (184)	19.1 (175)	20.5 (187)	21.7 (198)		
	15	5	1	27.1 (248)	27.8 (254)	28.5 (261)	29.9 (273)	30.3 (277)		
<i>Tandems with pressure feeders on all mills</i>										
S. Africa	25	5	1	19.3 (176)	19.3 (176)	19.9 (182)	20.1 (184)	19.4 (177)		
Australia	20	4	2	21.1 (193)	22.9 (209)	23.6 (216)	21.4 (196)			
	30	6	2	16.7 (153)	17.3 (158)	17.3 (158)	17.8 (163)	19.6 (179)	19.7 (180)	

N = number of mills in tandem; N₁ = number of rollers; n = number of tandems on which average is based.

The following general comments may be made:

- (1) Short tandems are obliged to use higher pressures, while longer tandems may use pressures appreciably lower
- (2) Intermediate mills are generally more lightly loaded
- (3) The final mill often has the heaviest load.

When the tandem has a 2-roller crusher, it generally has an s.h.p. 50 - 60% of that of the first mill.

We would recommend the following scale of pressures, which may be understood also as giving relative values for factories which cannot attain the loading indicated:

TABLE 10.6

RECOMMENDED MEAN VALUES OF S.H.P., IN t/dm² (tons/sq.ft.)

Tandem	1st mill	2nd mill	3rd mill	4th mill	5th mill	6th mill	7th mill
12 rollers	26 (240)	24 (220)	27 (250)	30 (275)			
15 rollers	26 (240)	23 (210)	25 (230)	27 (250)	29 (265)		
18 rollers	25 (230)	22 (200)	24 (220)	25 (230)	26 (240)	28 (260)	
21 rollers	25 (230)	21 (190)	23 (210)	24 (220)	25 (230)	27 (250)	28 (260)

Variation of hydraulic pressure with length of the tandem. We have already remarked that short tandems are obliged to employ high hydraulic pressures, whereas long tandems may go more easily with their hydraulic pressures if they so desire.

There is indeed so much less to be gained by increasing the hydraulic pressure when the tandem is long, since the number of compressions intervenes to compensate rapidly for the loss suffered at each mill by insufficient s.h.p.

If, for one reason or another, the total power which can be supplied to the mills is limited, the mean t.h.p. may in fact be reduced in inverse ratio to the square root of the number N of rollers in the tandem:

$$\frac{P}{P'} = \frac{\sqrt{N'}}{\sqrt{N}} \quad (10.84)$$

This rule permits a certain parity to be maintained between the total power absorbed by the tandem and its capacity (*cf.* pp. 190 and 237). It is more or less instinctively followed by engineers in charge of milling plants, but many factories are now preoccupied with extraction and steam economy, and the maximum work is demanded from each mill; the total power absorbed by the tandem is in that case proportional to the number of rollers in the tandem.

NOMENCLATURE

The following nomenclature applies for milling calculations in Chapters 10–14, unless indicated otherwise in the text. Units, also subscripts and superscripts, are given in the text.

A - crushing rate, t.c.h.

B = weight of bagasse leaving mill per hour

b = "bagasse loading" = weight of bagasse per unit roller surface

C (or C_{EH}) = "compression" (Hugot) = e_A/H or h/H

c - compression ratio (Bullcock) = $\frac{\text{volume of material entering}}{\text{volume available for compressed material}}$

c_A - compression ratio at minimum opening

C_F = filling ratio

D = mean diameter of roller

d = density (d_B for compressed bagasse, d_c for cane, d_j for juice)

E = opening (feed) between rollers (E_A in axial plane)

e = delivery opening between rollers (e_A in axial plane, e_N at neutral plane)

F = total load on roller

f = coefficient of friction (subscripts as indicated)

f_c = fibre per unit weight of cane

f_b = fibre per unit weight of bagasse

H - thickness of layer of loose bagasse

h = thickness of layer of compressed bagasse

L = length of roller

- m = ratio of feed to delivery openings while operating
 N = number of rollers in tandem
 N_1 = number of mills in tandem
 N_2 = number of compressions in tandem
 n = speed of rollers, r.p.m.
 P = power consumption of mill (subscripts 1, 2, ..., E, S)
 p = specific hydraulic pressure on roller
 p = pressure on bagasse
 q = fibre loading = weight of fibre per unit roller surface
 R = mean radius of roller
 r = re-absorption factor = $\frac{\text{bagasse volume}}{\text{escribed volume}} = V_R/V_F$
 S = escribed area (surface)
 t = compaction coefficient = $\frac{\text{weight of fibre}}{\text{total apparent volume of material}}$
 V = peripheral speed of rollers
 V_0 = no-void volume of cane
 V_B = volume of bagasse
 V_E = escribed volume of roller opening
 a = top angle of mill
 a = angle of contact of feed with roller
 ρ = density of fibre
 e = specific opening between rollers = e/D
 B = angle between position of feed and axial plane
 X = imbibition per unit fibre
 μ = ratio of changes in feed and delivery opening = dE/de
 ν = angle between position of feed and neutral plane
 Q = mechanical efficiency of gearing
 T = specific fibre loading
 ϕ = fibre index = $\frac{\text{weight of fibre passing}}{\text{escribed volume}}$
 ψ = safety coefficient for mill settings

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11. Mill Speeds

LINEAR SPEED AND SPEED OF ROTATION

One often speaks loosely of the speed of mills, with consequent risk of confusing two different things. There are in fact two ways of reckoning this speed:

(a) As peripheral speed of the rollers, *i.e.* the linear speed of a point at the mean diameter of a roller. It is generally expressed in metres per minute (feet per minute).

(b) As speed of rotation of the rollers, *i.e.* the number of revolutions which they make in unit time. It is generally expressed in revolutions per minute (r.p.m.).

Relation between the two speeds

We have:

$$V = \pi Dn \quad (11.1)$$

V = peripheral speed, in m/min (ft./min)

D = diameter of rollers, in m (ft.)

n = speed of rotation, in r.p.m.

Whence:

$$n = \frac{V}{\pi D} \quad (11.2)$$

Use of the two speeds

When discussing speed of mills, should we use the peripheral speed V , or the speed of rotation n ?

The general practice is to use predominantly V , which gives a measure of the speed of movement of the cane on the carrier, and of the bagasse in the intermediate carriers and in the mills themselves. However, some authors attach more importance to the speed of rotation than to the peripheral speed. Which is the correct view?

This question is very far from being merely academic. It is of direct importance to both the manufacturer and the user of mills.

The question is important mainly on account of the effect of speed on extraction. This acts in two ways:

(1) On the theoretical plane, it would be valuable to know which factor is important, since the answer to this question will allow us to better penetrate the mysteries of the mill and to understand it better.

(2) On the practical plane, since all mill rollers do not have the same dimensions, if Paul

possesses a mill of 0.61 m (24 in.) roller diameter and Peter a mill of 1.09 m (42 in.), it is obvious that they will reach very different conclusions according as the limit for a good extraction is attained in the neighbourhood of 7 r.p.m. or in the neighbourhood of 20 m/min (65 ft./min):

If it is $n_M = 1$ r.p.m.	for Paul: $V = 13.4$ m/min (44 ft./min)
	for Peter: $V = 23.4$ m/min (77 ft./min)
If it is $V_M = 20$ m/min	for Paul: $n = 10.4$ r.p.m.
	for Peter: $n = 6.0$ r.p.m.

With such differences, it may well be asked why world-wide experience has not yet given the answer to the problem. There are many reasons for this and in particular the following:

- (a) Most tandems cannot be operated at these limiting speeds and so cannot reach any conclusion on the question.
- (b) The extraction depends on many other factors besides speed, which cannot be readily eliminated and which thus complicate the situation.
- (c) The design of mills does not allow the speed to be readily varied without at the same time varying the crushing rate, and this is an important factor which can completely falsify any conclusions.

Whatever the reason, we do not know of any precise experiment made on this subject on the industrial scale.

However, of the factors V and n , it is certainly V which plays the dominant role. It is the factor opposing the escape of the expressed juice, when the bagasse enters the grip of the rollers. The juice must in effect (Fig. 11.1) flow through the bagasse layer against its direction

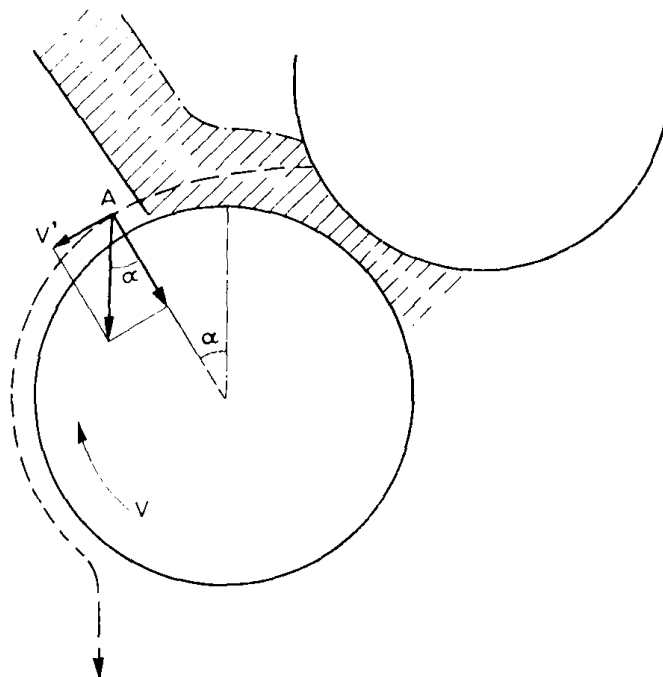


Fig. 11.1. Flow of extracted juice.

of movement. Now this factor **obviously** depends only on the linear speed V ; and this is certainly the greatest argument in favour of the predominance of V over n . The higher the speed V , the more difficult will it be for the juice to escape **against** the movement of the bagasse and the greater will be the amount of juice entrapped in the compressed bagasse at the moment where its permeability drops practically to zero (c/. p. 132).

Murry¹, in a study of the results obtained by *Bullock* and himself, also agrees on the predominance of V over n . However, considerations cannot be based on only one of these factors: for reasons of efficacy and economics of the milling plant, a compromise between the two factors must be adopted.

The Cail firm gave its mills a maximal speed expressed by the formula:

$$n_M = 6.37 - 1.835 D \quad (n = 6.37 - 0.56 D) \quad (11.3)$$

n_M = maximal rotation speed of rollers, in r.p.m.

D = roller diameter, in m (ft.).

This old formula has been surpassed by actual practice, and Fives Cail – Babcock, successors to Cail, have given a fixed scale of maximal speeds for their mills, expressed in the formula:

$$n_M = \sqrt{67 - 21.4 D^2} \quad (n_M \approx \sqrt{67 - 2 D^2}) \quad (11.4)$$

The few experiments made on this subject have shown the existence of a speed limit of some 23 or 24 m/min (75 or 80 ft./min), above which the tonnage drops. Hence, we propose the rule:

$$V_M = \frac{33 D}{D + 0.73} \quad \left(V_M = \frac{108 D}{D + 2.4} \right) \quad (11.5)$$

V_M = maximal speed in m/min (ft./min) for rollers of diameter D , in m (ft.)
which reduces to:

$$n_M = \frac{10.5}{D + 0.73} \quad \left(n_M = \frac{34.4}{D + 2.4} \right) \quad (11.6)$$

This rule is valid for diameters from 0.6 to 1.5 m (2–5 ft.) (Fig. 11.4).

MAXIMAL SPEEDS EMPLOYED

The technical literature gives little information concerning maximal speeds attained in practice. However, since most factories throughout the world periodically find themselves spurred on by an increase in tonnage which reaches or exceeds the capacity of their mills, the problem is inevitably posed to the great majority of them. In many cases the crushing rate has

been pushed to the maximum permitted by the equipment, a maximum which is very often well below the limits which we have given. Certain recent factories, on the other hand, have been so designed as to approach those limits.

At Central Plata, for example, in Puerto Rico², the tandem consisting of two sets of knives, a Searby shredder and 21 rollers of 889 X 1,676 mm (35 X 66 in.), was crushing at 200 t.c.h. at a speed n of 7 r.p.m., or $V = 19.5$ m/min (64 ft./min), with a normal extraction (figures not given, but one may assume 94–95%). Some factories now (1981) attain speeds of the order of 18–20 m/min (60–65 ft./min).

Tromp³ gave a speed limit of:

$$V_M = 18 D \quad (11.7)$$

which is equivalent to:

$$n_M = 5.73 \text{ r.p.m.} \quad (11.8)$$

In Louisiana⁴ a practical limit has been expressed as a peripheral speed: $V_M = 16$ m/min (53 ft./min), but this limit could probably have been equally expressed in r.p.m., since it is noted that the standard factory in Louisiana is equipped with mills of 863-mm (34-in.) rollers; this reduces therefore to $n_M = 6$ r.p.m.

SPEEDS IN GENERAL PRACTICE

As indicated above, most factories operate below these extreme speeds, to avoid risk of affecting their extraction. For example, 812-mm (32-in.) rollers are operated at approximately 5 r.p.m.; it is evident that the extraction cannot but improve when the mills are operated at a lower speed. However, a good economic use of the equipment demands that the roller speeds should approach the limits - rather cautious - which we have given (eqns. (11.5) and (11.6)). If it is desired to preserve a small margin of safety, the following may be considered as maximum economic industrial speeds:

$$\begin{aligned} V_E &= \frac{30 D}{D + 0.73} \quad \left(V_E = \frac{100 D}{D + 2.4} \right) \\ n_E &= \frac{9.55}{D + 0.73} \quad \left(n_E = \frac{32}{D + 2.4} \right) \end{aligned} \quad (11.9)$$

n_E = economic rotational speed, in r.p.m.

V_E = economic peripheral speed, in m/min (ft./min)

D = diameter of rollers, in m (ft.).

We show (Fig. 11.2) the speeds considered by Fives Cail–Babcock as maximal, normal and minimal, for their mills, as functions of roller diameter.

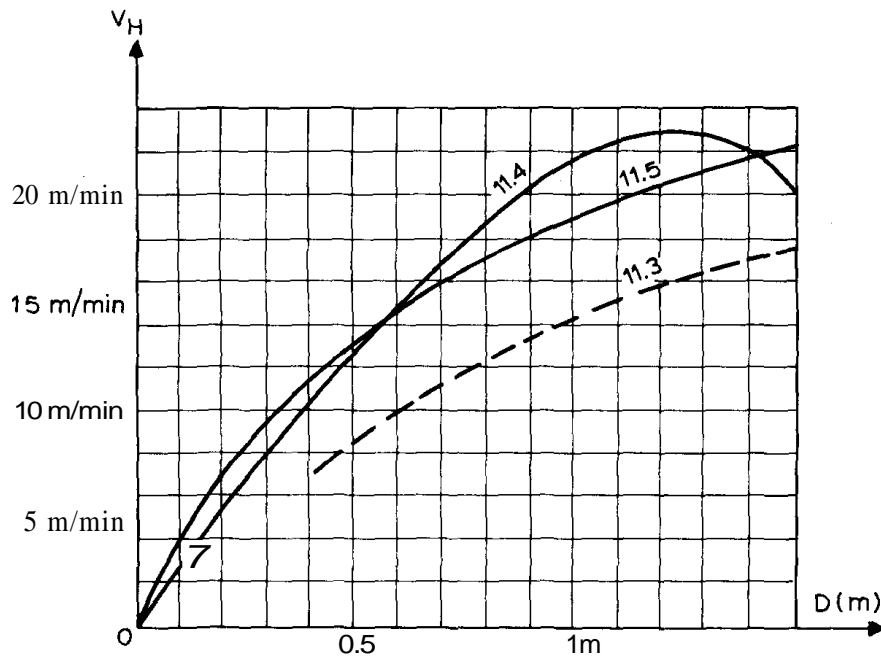


Fig. 11.2. Value of maximal linear speed according to eqns. (11.3), (11.4) or (11.5).

Sequence of speeds along the tandem

What should be the variation of speeds, in milling trains, from the crusher to the final mill?

Previously, two methods were distinguished:

- (1) The Javan school, where low speeds were favoured, decreasing from the first to the final mill
- (2) The Hawaiian school, for which speeds were higher, and increased from the first to the final mill.

The Javan school of thought has now vanished, and this distinction has no more than academic interest. Present practice is to adopt, very generally, either the same speed from one end to the other of a tandem, or increasing speeds.

Actually, it is predominantly a question of settings, and it is difficult to see how there would be any advantage in adopting increasing, equal or decreasing speeds: the gain realised by a thinner layer of bagasse is offset by the increase in peripheral speed, or *vice versa*. The simplest solution would therefore seem to be, *a priori*, to use a uniform speed for all mills of the tandem.

However, the American trend, a solution of convenience, has held sway, probably as a sort of safeguard against possible errors of setting and against accidental chokes towards the end of the tandem. It is considered that by running the later mills slightly faster, they will readily accept the bagasse furnished by the preceding mills. The angle of contact of the bagasse is improved as the speed increases, since the thickness of the bagasse layer decreases accordingly-

Table 11.1 gives further examples of mill speeds, from South Africa and Australia, in m/min (and ft./min), with figures grouped separately for tandems with pressure feeders on all mills. In all cases, the data are from tandems giving very good extraction figures.

TABLE 11.1

LINEAR SPEEDS FOR SEVERAL MILL TANDEMS, IN m/min (ft./min)

N	N_1	n	Mill number							
			1	2	3	4	5	6	7	
<i>Normal tandems</i>										
18	6	4	10.76 (35.3)	10.77 (35.3)	10.79 (35.4)	10.51 (34.5)	10.16 (34.6)	10.59 (34.7)		
21	7	2	11.16 (36.5)	11.01 (36.1)	10.87 (35.7)	11.11 (36.5)	11.53 (37.8)	11.33 (37.2)	9.82 (32.2)	
<i>Tandems with pressure feeders on all mills</i>										
20	4	1	13.10 (43.0)	11.82 (38.8)	10.86 (35.6)	9.84 (32.3)				
25	5	1	13.09 (42.9)	13.20 (38.7)	12.80 (42.0)	12.70 (41.7)	13.20 (43.3)			
30	6	1	13.20 (43.3)	16.08 (52.8)	15.66 (51.4)	15.96 (52.4)	13.86 (45.5)	12.06 (39.6)		

N = number of rollers in tandem; N_1 = number of mills; n = number of tandems on which average is based.

Return of fine bagasse

It is necessary, however, to take into account the return of fine bagasse, which often represents 20–25% on the weight of cane entering the milling tandem. The mill (generally the second) which receives such an extra load should be assisted, in one way or another. The hydraulic load applied to it should accordingly be lower, and at the same time it should be

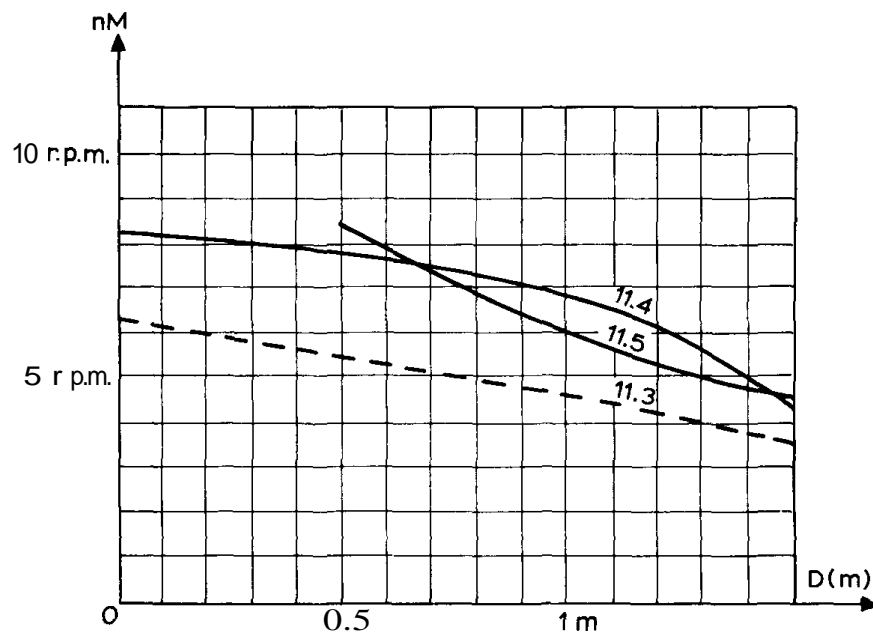


Fig. 11.3. Value of maximal rotational speed according to eqns. (11.3), (11.4) or (11.5).

operated at a sufficient speed; and, of course, the settings should be adjusted accordingly.

We may comment, however, that the mill is not handicapped by the full 20% of returned bagasse. If, for example, there are 5 mills, crushing cane of 13.7% fibre, and if the quantities of fibre in fine bagasse, per unit of cane, coming from each mill, are:

<i>1st mill:</i>	0.010
<i>2nd mill:</i>	0.009
<i>3rd mill:</i>	0.004
<i>4th mill:</i>	0.002
<i>5th mill:</i>	0.001
	0.026

the quantities of fibre passing through each mill in a given time will be, if the return is made at the second mill, in the following ratios:

<i>1st mill:</i>	137	or say	100
<i>2nd mill:</i>	153	or say	112

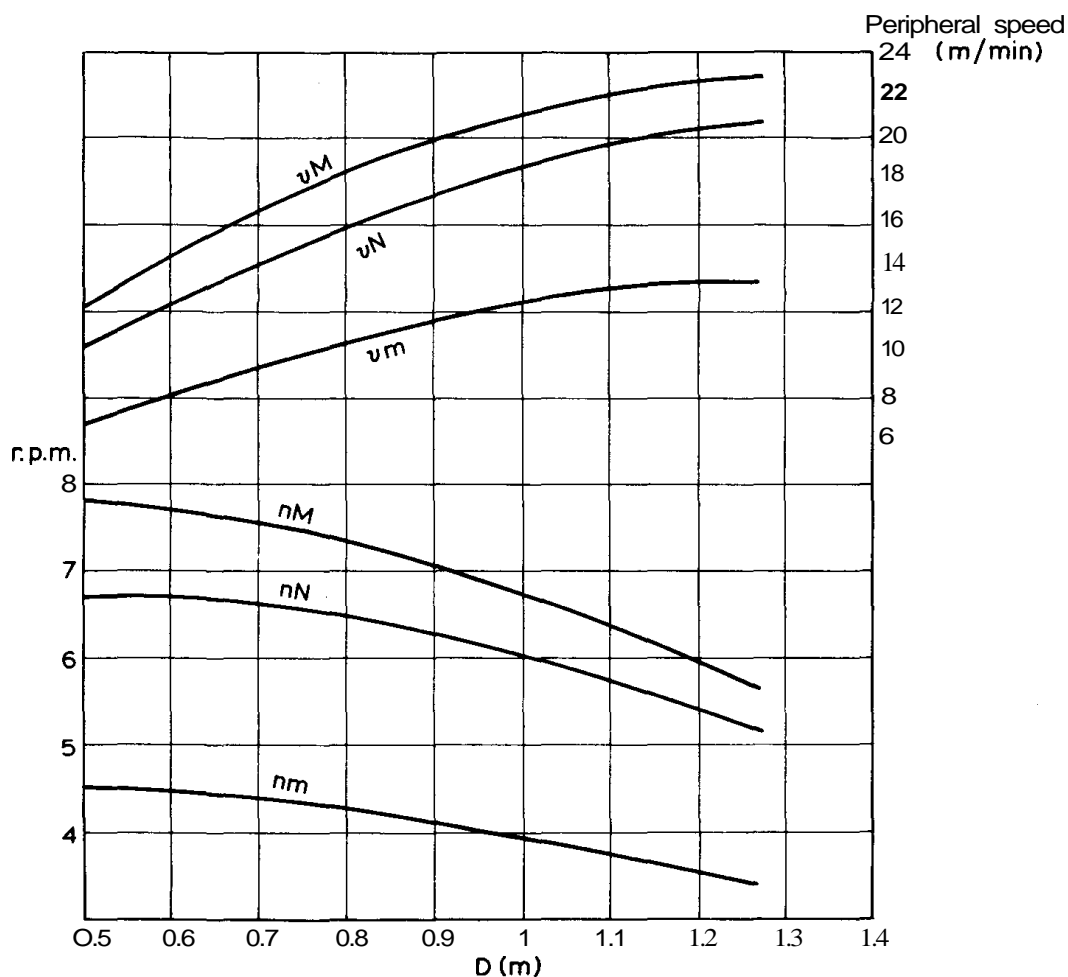


Fig. 11.4. Maximal (M), normal (N) and minimal (m) speeds in the Fives Cail-Babcock mills.

<i>3rd mill:</i>	144	or say	105
<i>4th mill:</i>	140	or say	102
<i>5th mill:</i>	138	or say	101

The extra quantity, however, is far from negligible. In spite of this consideration, the speed of the second mill often remains the lowest of the tandem.

Combined drives

It is in tandems with combined drive that the question of the sequence of speeds becomes most acute. When 11 or 14 rollers are driven by the one steam engine, the relative speed of the several mills can no longer be varied at will.

This case is becoming more and more rare, on account of the increased adoption of individual drives by electric motor or turbine. For combined drive, French designers generally maintain the same speed for all mills, to maintain uniformity, with standard gearing and frames. American firms prefer to vary the gear ratios so as to obtain speeds increasing from the first to the last mill. Tromp⁵ cites an 11-roller tandem in which the second motion pinions driving the spur wheels of the three mills had respectively 22, 23 and 24 teeth. This system, he added, would give good results.

If it is desired to maintain the same dimensions for the gear teeth, and the same spaces between mills, the ratios may be modified while retaining the same total number of teeth for spur wheel plus pinion. For example, 3 sets of: 83 and 22, 82 and 23, and 81 and 24 teeth.

Laredo, in Peru⁶, in altering its tandem (762 x 1,524 mm) (30 x 60 in.) driven by electric motors of 1,000 r.p.m., with a 42.5 : 1 reduction), arranged for the spur wheels of 104 teeth for each mill to be driven by pinions as shown below, with the resulting speeds:

<i>1st mill:</i>	21 teeth	$n = 4.725$ r.p.m.	$V = 11.25$ m/min (37 ft./min)
<i>2nd mill:</i>	20 teeth	$n = 4.50$ r.p.m.	$V = 10.5$ m/min (35 ft./min)
<i>3rd mill:</i>	22 teeth	$n = 4.95$ r.p.m.	$V = 12.0$ m/min (39 ft./min)
<i>4th mill:</i>	24 teeth	$n = 5.40$ r.p.m.	$V = 13.0$ m/min (42 ft./min)
<i>5th mill:</i>	24 teeth	$n = 5.40$ r.p.m.	$V = 13.0$ m/min (42 ft./min)
<i>6th mill:</i>	26 teeth	$n = 5.85$ r.p.m.	$V = 14.0$ m/min (46 ft./min)

Further good examples of sequences of speed are given in Tables 5.1 and 5.2.

Today, complete combined drive is no longer used. However, for economy, some factories drive two mills from one prime mover; but individual drive remains the rule.

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12. Mill Capacity

DEFINITION

The capacity of a train of mills is the quantity of cane which that train is capable of treating in unit time. It is generally expressed in tonnes of cane per hour (t.c.h.); in America it is often expressed in tonnes of cane per day.

The relation between these two figures is not as direct as one might be tempted to think. **Actually**, the hourly tonnage assumes that the mills have been operating without interruption for the hour under consideration. This obviously is generally the case, and the figure is easily obtained, at the end of the season, by dividing the tonnage crushed by the number of hours of operation of the mills; in effect, stops and breakdowns are always deducted.

The tonnage per day, on the other hand, is obtained by dividing the tonnage handled during the campaign by the number of working days, small stops not being deducted. Now, while an hour's crushing without interruption is the general thing, and while many daily reports show continuous crushing for 24 hours, many stops of 10 — 20 minutes are experienced in the course of a season. It follows that, even if a careful record is kept of the days of operation during the week (5.83, 6.25, etc.), the tonnes per day will not be 24 times the tonnes per hour.

A well-operated factory should not have lost time (mill stops) of more than about 1% of its operating time. From this mean figure, the t.c.h. may be converted to tonnes cane per day (t.c.d.) by multiplying by 23.75:

$$C = x \text{ t.c.h.} = 23.75 x \text{ t.c.d.} \quad (12.1)$$

FACTORS INFLUENCING CAPACITY

These are many. The most important are the following:

(a) Fibre content of cane. Though the volume presented to the crusher does not depend on the fibre content, its resistance to the action of the rollers is more or less proportional to the fibre. As for the later mills of the tandem, the quantity of material which they receive is closely proportional to the fibre, and the thickness of the layer of compressed bagasse at the delivery roller will, for a given r.h.p., be exactly proportional to the fibre.

(b) Dimensions and speed of the rollers. The quantity of bagasse is obviously proportional to the product of the thickness of the bagasse layer and the escribed area, *i.e.* the area described in unit time by a generatrix of the roller, *i.e.* it is proportional to $HLnD$. Since H

should be proportional to D , the tonnage will vary approximately as LnD^2 ; thus we see that it is proportional to the speed and length, and also to the square of the diameter, of the rollers.

(c) *Number of rollers.* A factor which is less obvious, but still important, is the number of mills. At first sight, it may seem surprising that it should have any effect: if a certain quantity of bagasse passes through a first mill, when it is followed by 5 others for example, why would it not pass through just as well if it were followed by only 4, 3 or 2 mills, instead of 5? It is the minimum permissible extraction that is important here: in a short tandem, the thickness of the bagasse layer must be reduced to obtain a satisfactory extraction; in a long tandem, on the other hand, the thickness of bagasse layer may be increased as the tandem becomes longer.

(d) *Cane preparators.* We have already seen how knives and shredders improve the capacity.

Besides these primary factors, a number of others also play an important part:

(e) *Imbibition.* The greater the imbibition used, the more difficult it is to feed the mills, especially when hot water is used as imbibition.

(f) *Grooving.* The grain of the roller metal, and the shape and depth of the grooving, have their effect on the grip of the rollers: the juice grooves play an important part at the feed roller.

(g) *Hydraulic pressure.* The mills accept the feed more readily if they are more lightly loaded.

(h) *Roller surface.* It is general practice to apply, by arc welding, granules of metal on the crest and flanks of the roller teeth. This practice (known as "arcing" or "spotting") considerably increases the grip of the rollers by reducing the slip of the cane or bagasse on them, and appreciably improves their capacity.

(i) *Use of feeding devices.* Feeding rollers, above-feed and particularly under-feed pushers and continuous pressure feeders allow the tonnage to be increased, sometimes to a substantial extent.

(j) *Design and condition of the plant.* Modern mills are provided with various improvements which increase their capacity. On the other hand, wear of grooving, of rollers and of surfaces subjected to friction, particularly in the hydraulic system, can impair the good operation of a mill.

(k) *Personnel.* Finally, the quality of the personnel responsible for the adjustment and maintenance of the mills is an intangible factor, but one which can be of immense importance.

CAPACITY FORMULAE PROPOSED

- A = capacity of the tandem, in t.c.h.
 f = fibre content per unit of cane
 c = coefficient for preparatory plant, given by Table 12.1
 L = length of rollers, in m (ft.)
 D = diameter of rollers, in m (ft.)
 n = speed of rotation of rollers, in r.p.m.
 N = number of rollers in the tandem
 N_1 = number of mills in the tandem
 N_2 = number of compressions applied by the tandem (1 mill = 2 compressions).
 Note that:
 Af = capacity of the tandem in tonnes fibre per hour (t.f.h.).

CAPACITY FORMULAE

Many formulae have been proposed for calculating the capacity of mills (Noel Deerr^{1,2}, Maxwell³, Nayar and Pillay⁴, Parr⁵, Tromp⁶, Gaspar⁷, and many others). The dimensions of the mill rollers form the main factor in all these formulae; they appear generally in the combination LD or LD^2 . Maxwell and Nayar are supporters of the formula with LD : but it is difficult to escape the logic which leads to the use of the second power of the diameter. One arrives at the exponent 1 by assuming that the thickness of the bagasse layer is constant and independent of the diameter. However, we start from the logical and necessary hypothesis that this thickness is proportional to roller diameter. Considering this hypothesis in terms of fibre, the weight of fibre passing in one hour is Af and this weight is distributed over a roller area of:

$$S = 60 \pi r D n L$$

It follows that:

$$\frac{Af}{S} = \frac{Af}{60 \pi D n L} = \lambda D$$

whence:

$$A = \lambda' \frac{nLD^2}{f} \quad (12.2)$$

However, the bagasse layer which is of interest here is the maximum layer compatible with good operation of the mills without choking. This maximum layer depends on the coefficient of friction between metal and bagasse. Now, the experiments of Bullock⁸ in Australia have shown that this coefficient itself depends on the peripheral speed, and the average of his experiments leads to an equation of the form:

$$\mu = 0.43 - 0.008 v \quad (\mu = 0.43 - 0.0024 v) \quad (12.3)$$

μ = coefficient of friction between bagasse and metal
 v = peripheral speed of the mill, = πnD , in m/min (ft./min).

Taking this factor into account, the maximum quantity of bagasse which will be gripped by the mill is no longer proportional to v , that is, to πnD , but to the product:

$$\pi nD (0.43 - 0.008 \pi nD) \quad (\pi nD (0.43 - 0.0024 \pi nD))$$

that is, to:

$$nD (1 - 0.06 nD) \quad (nD (1 - 0.018 nD)) \quad (12.4)$$

From an experiment made at Central Don Pedro and cited by Fives - Lille, the practical coefficient of friction is given as $(1 - 0.07 nD)$. However, we retain the value 0.06, which appears to be better established. We may note in passing that this same experiment showed a limiting speed $v = 23 - 24$ m/min (76 - 78 ft./min), above which the tonnage fell sharply.

Figure 12.1 shows the influence of slip, as given by this formula, the speed $v = \pi nD$ being shown as the abscissa. On the same graph, we show the coefficient adopted by Fives Cail - Babcock, which considers only speeds above 9 m/min (30 ft./min). The two graphs are displaced by about 30% in the useful zone, this difference of 30% being evidently taken into account by the respective formulae. If we reduce the upper graph in the proportion 130/100, they are very close together in the range 10-24 m/min (33 - 78 ft./min). We thus obtain the curve shown dotted.

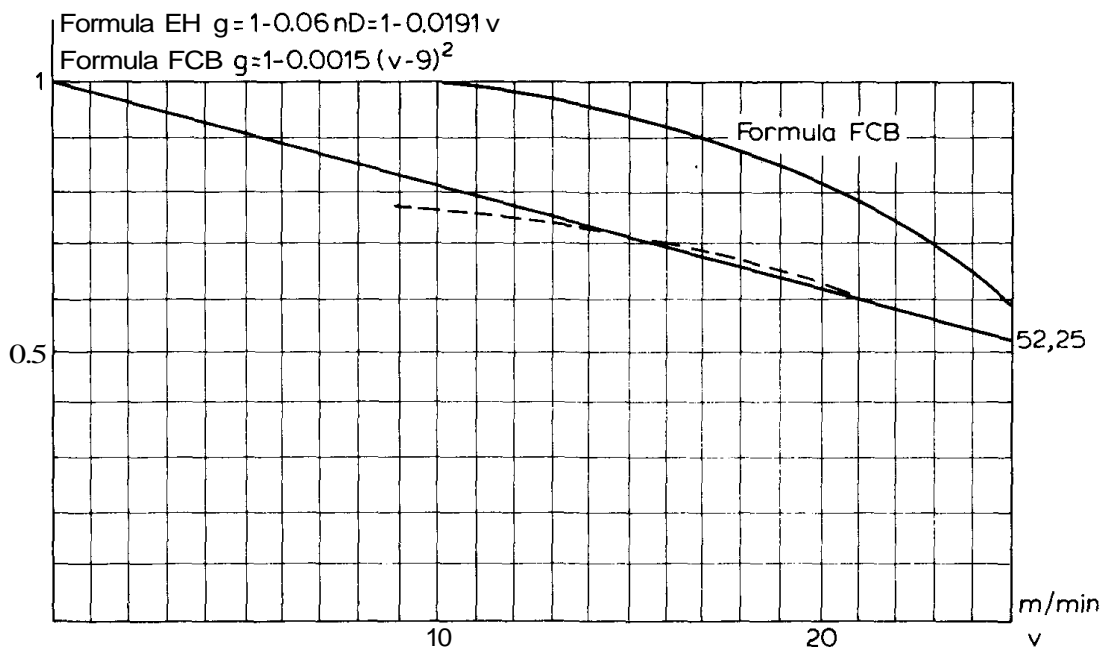


Fig. 12.1. Slipping factor in mill capacity formulae.

To allow for this corrected factor, we must now write:

$$A = X'' \frac{nID^2}{J} (1 - 0.06 nD) \quad \left(A = \lambda'' \frac{nID^2}{J} (1 - 0.018 nD) \right) \quad (12.5)$$

an equation in which we have already taken into account fibre, speed, and the dimensions of the rollers. There remain only, included in the coefficient λ'' , two primary factors to be considered explicitly:

- (1) The equipment for cane preparation
- (2) The influence of the length of the tandem.

Influence of preparatory plant

The degree of preparation and more or less complete subdivision of the cane has a strong influence on the capacity of the mills to grip the cane. Table 12.1 gives coefficients to be used to take this effect into account.

TABLE 12.1

COEFFICIENTS FOR PREPARATORY PLANT

1 Knife set of wide pitch	$c = 1.10$
2 Knife sets of wide pitch	$c = 1.15$
1 Knife set at close pitch	$c = 1.13$
2 Knife sets, one wide, one at close pitch	$c = 1.16$
2 Knife sets and one shredder	$c = 1.22$
Shredder only with chopper-harvested cane	$c = 1.18$

Influence of length of tandem

This is the factor which is most difficult to evaluate, because its effect is indirect. Let us consider an 11-roller train; this is chosen because it is the shortest of normal tandems, and because its maximum capacity is more definitely fixed. Its capacity cannot be increased much without the extraction suffering markedly. Let us add in turn 1, 2, 3, 4 mills. With each addition we have the choice between two solutions; either:

(a) To profit by the new mill to treat as much cane as possible, while maintaining the same extraction; or:

(b) To seek to improve the extraction as much as possible with the aid of the new mill, and not concern ourselves with capacity.

If solution (a) is followed, as in factories which have to cope with increasing production, the increase in capacity of the tandem will be almost proportional to the number of mills N_1 or, which is practically the same thing, almost proportional to the number of compressions N_2 or of rollers N .

If method (b) is followed, as in factories seeking to increase their efficiency, the capacity will increase very little with the number of rollers.

In practice, it is nearly always the pressure of production requirements that prompts the addition of a mill, but an effort is always made to obtain from the new mill both a gain in

capacity and a gain in extraction. If a reasonable attitude be adopted, which favours extraction very slightly more than capacity, the influence of length of the tandem will be proportional to VTV.

Formerly, Java issued complete synoptic tables for its 160 or 180 factories. If an analysis be made of this copious data, and the exponent of N be taken as the unknown, the actual values found are distinctly lower than 1, but slightly greater than 0.5. The power of N which would appear closest to the mean of all the figures (though somewhat high) would be 0.67, that is, $\sqrt[3]{N^2}$. This would be a complicated value to calculate, and would particularly involve seeking a precision which the problem does not warrant; so we consider that it is preferable to retain VTV.

For record purpose, and in spite of their early date, we consider that the results for Java for 1930, reduced to equivalent figures for (762 x 1,524 mm) (30 x 60 in.) mills, are worth retaining. The main differences are due to the influence of the crusher on capacity during that period (*cf.* Table 12.4).

Table 12.2 expresses the influence of length of the tandem, in terms of the foregoing discussion.

TABLE 12.2

COEFFICIENTS FOR LENGTH OF TANDEM		
<i>Number of rollers</i>	\sqrt{N}	<i>Relative values</i>
9	3	0.866
11	3.317	0.958
12	3.464	1.00
14	3.742	1.08
15	3.873	1.12
17	4.123	1.19
18	4.243	1.22
21	4.583	1.32
24	4.899	1.41

Complete formula

We are now in a position to give the complete formula for capacity:

$$A = \frac{cn(1 - 0.06nD)LD^2 \text{ VTV}}{f} \quad (12.6)$$

$$\left(A = 0.0255 \frac{cn(1 - 0.018nD)LD^2 \sqrt{N}}{f} \right)$$

(L and D in m (ft.).)

This formula assumes that the tandem is operated with a normal compromise between extraction and capacity, and that the rollers are subjected to regular and uniform "arcing". Where such "arcing" is not practised, it is advisable to apply a coefficient of 0.95, which indicates that the "arcing" allows for 5% in the capacity of the tandem.

TABLE 12.3
CAPACITY OF MILL TANDEMS, IN t.c.h. ($\sigma = 0.15; n = 6$ r.p.m.)

(in. x in.)	D x L (mm x mm)	LD ² (m ²)	Cr + 3M N = 11		4M N = 12		Cr + 4M N = 14		5M N = 15		Cr + 5M N = 17		6M N = 18		7M N = 21	
24 x 48	610 x 1,220	0.454	52	54	59	61	65	67	72	72	74	76	76	83	83	83
26 x 48	660 x 1,220	0.534	60	62	67	70	74	76	76	76	74	76	76	83	83	83
28 x 54	710 x 1,370	0.691	76	79	85	88	94	85	88	88	94	94	97	104	104	104
30 x 60	760 x 1,525	0.881	94	98	106	110	117	106	110	110	117	117	120	130	130	130
32 x 60	813 x 1,525	1.008	105	109	118	122	130	118	122	122	130	130	134	145	145	145
32 x 66	813 x 1,675	1.107	115	120	130	134	143	130	134	134	143	143	147	159	159	159
34 x 66	864 x 1,675	1.250	126	132	143	148	157	143	148	148	157	157	162	175	175	175
36 x 66	915 x 1,675	1.402	138	144	156	161	172	156	161	161	172	172	177	191	191	191
34 x 72	864 x 1,830	1.366	138	144	151	161	172	151	161	161	172	172	177	191	191	191
36 x 72	915 x 1,830	1.532	149	156	168	174	185	168	174	174	185	185	191	206	206	206
36 x 78	915 x 1,980	1.688	163	171	184	191	203	184	191	191	203	203	209	226	226	226
36 x 79	920 x 2,000	1.683	166	174	188	194	207	188	194	194	207	207	213	230	230	230
38 x 83	970 x 2,100	1.976	189	197	213	221	235	213	221	221	235	235	242	261	261	261
38 x 84	965 x 2,134	1.987	190	199	215	222	237	215	222	222	237	237	244	263	263	263
40 x 84	1,016 x 2,134	2.203	205	214	232	240	255	232	240	240	255	255	263	284	284	284
42 x 84	1,067 x 2,134	2.429	220	229	248	257	273	248	257	257	273	273	281	304	304	304
44 x 84	1,118 x 2,134	2.665	234	244	264	273	291	264	273	273	291	291	299	323	323	323
46 x 84	1,168 x 2,134	2.913	248	259	280	290	308	280	290	290	308	308	317	343	343	343
42 x 90.5	1,070 x 2,300	2.633	238	248	268	278	296	268	278	278	296	296	304	329	329	329
46 x 98	1,170 x 2,500	3.422	294	304	328	340	362	328	340	340	362	362	372	402	402	402
50 x 102	1,270 x 2,600	4.194	334	349	377	390	416	377	390	390	416	416	428	462	462	462

Table 12.3 gives capacities for mills of the sizes most often used, calculated for the following conditions:

(a) Preparatory plant: two sets of knives and a shredder

(b) Fibre: $f = 0.15$

(c) Speed: $n = 6$ r.p.m.

For any fibre content f' other than 0.15, it will suffice to multiply the indicated capacity by $0.15/f'$. However, if we make some concession to Gaspar's point of view, we would count as 0.12 all fibre values equal to or lower than that figure.

For any speed other than 6 r.p.m., we multiply by:

$$\frac{n' (1 - 0.06 n' D)}{6 (1 - 0.36 D)} \quad \left(\frac{n' (1 - 0.018 n' D)}{6 (1 - 0.108 D)} \right)$$

For any combination of preparatory plant corresponding to a coefficient c' differing from 1.22, multiply by $c'/1.22$.

Comment: The last line is only a theoretical indication for the sake of homogeneity of the table, since the speed of 6 r.p.m. adopted for the table as a whole would be conducive to a linear speed of 24 m/min (78 ft./min), which we know would be excessive.

Influence of crusher

We have already pointed out that many factories having only three-roller mills show capacities inferior to factories having one roller less, *i.e.* where the first mill is replaced by an ordinary crusher. Thus in Java, 15-roller trains achieve a mean tonnage inferior to that of 14-roller trains (*cf.* Table 12.4). It may be asked in these conditions, whether it would not be appropriate to separate the crusher from the rest of the tandem, to treat it as a preparatory unit, in the same class as a set of knives; to introduce a "crusher coefficient", and to count only the number of units (or of pressures, or of rollers) in the mills proper, in which case these figures would be fully comparable.

TABLE 12.4

AVERAGE FIGURES FOR JAVA								
Number of factories	Tandem		Number of rollers <i>N</i>	\sqrt{N}	Capacity (t.c.h.) <i>A</i>	A/\sqrt{N}	Extraction	Lost juice % fibre
3		3 mills	9	3	29.5	9.83	91.3	55
3	1 Crusher +	3 mills	11	3.32	31	9.34	93.5	48
26		4 mills	12	3.46	32.2	9.31	93.9	44
93	1 Crusher +	4 mills	14	3.74	41.7	11.15	94.6	39
15		5 mills	15	3.87	34	8.79	95.4	34
21	1 Crusher +	5 mills	17	4.12	45.5	11.04	95.7	33
1	1 Crusher +	6 mills	20	4.47	48.1	10.76	96.7	27

This would be true in tandems containing mills only. Some such tandems are still in existence, but fortunately they are becoming rare and, in all modern tandems with number of

rollers a multiple of 3, *e.g.* 12, 15 or 18, where the first mill is either preceded by a shredder, or provided with a suitable speed and suitable grooving, the capacity should attain and surpass that of a corresponding tandem of 11, 14 or 17 rollers. For our part, we are strongly in favour of a first unit of 3 rollers and we have always obtained from such units a tonnage superior to that obtained with a 2-roller crusher as first unit.

It is for this reason that we have reckoned the rollers of an ordinary crusher on the same basis as those of the mills.

The foregoing reasoning also shows why we have introduced in eqn. (12.6) the number of rollers, N , instead of the number of pressures, N_2 . It is because, from the point of view of capacity, a crusher has a value approaching that of a mill. However, in counting the number of *compressions*, one reckons a crusher as half a mill, whereas by counting *rollers*, one reckons it as two-thirds of a mill.

Reduction to standard dimensions

The influence of the roller dimensions is expressed, in logical capacity formulae, by the factor LD^2 .

In Java, the practice has been adopted of "reducing" the capacities of mills to that of a standard roller dimension of (762 X 1,524 mm) (30 x 60 in.), by multiplying the mean tonnage obtained by the ratio:

$$r = \frac{L_s D_s^2}{LD^2} \quad (12.7)$$

L and D being the length and diameter of the rollers of the tandem concerned, L_s and D_s those of the standard rollers.

It may be noted that the volume of a roller is given by:

$$V_c = \frac{\pi}{4} LD^2$$

and that in consequence the capacity is proportional to the roller volume. Hence it is estimated in Java that the capacity of a mill is 7.25 t.f.h./m³ (0.205 t.f.h./cu.ft.) of volume of one of its rollers. This scale has been derived from an average figure for a number of tandems comprising 1,254 rollers.

Noel Deerr⁹ has pointed out that an average calculated for 110 rollers, in India, gave a figure of 10.5 t.f.h./m³ (0.295 t.f.h./cu.ft.) of roller.

These two figures unfortunately have not taken into account the length of the various tandems considered. It would have been interesting to reduce the capacities, not only to a standard roller, but also to a standard tandem; at the time, this was one of 14 rollers.

At the present day this is a tandem of 15 rollers. If we apply formula (12.6) to a tandem of 15 rollers preceded by 2 sets of knives and a shredder, and operating at 6 r.p.m., with rollers of 813 mm (32 in.), this dimension serves only to calculate the term 0.06 nD (0.018 nD); we shall have:

$$\frac{Af}{c} = \frac{0.9 \times 1.22 \times 6 \times 0.7 \times 3.873}{0.7854} = 22\,740 \text{ t f h / m}^3$$

$$\left[\frac{Af}{V_c} = \frac{0.0255 \times 1.22 \times 6 \times 0.71 \times 3.873}{0.7854} = 0\,654 \text{ t f h / cu ft} \right] \quad (12.8)$$

or, for any speed n :

$$3.8 n \text{ t.f.h./m}^3 \quad (0.109 n \text{ t.f.h./cu.ft.}) \quad (12.9)$$

In South Africa, the length of the tandem is taken into account and work of the mills in t.c.h. is related to the total volume of all the rollers in the tandem. This amounts to accepting a formula giving capacity proportional to the number N of rollers, and we have commented (*cf.* p. 191) that this type of formula would exaggerate the influence of the number of rollers. The weight of fibre handled by the mills per hour is divided by the total volume of the rollers in the tandem, and the quotient is called "specific feed rate". In 1981 the South African factories had figures ranging from 727 to 1221 kg fibre per hour per m^3 of total volume of the main rollers (3 per mill) of the mills in the tandem (45-76 lb./h/cu.ft.). If the volume of the pressure feed rollers is added, this would give 692 – 953 kg/h/ m^3 (43 – 59 lb./h/cu.ft.).

Capacity of tandems with mills of different dimensions

The expressions given are applicable to tandems in which all the rollers have the same length L and the same diameter D ; but it often happens that tandems are not so homogeneous. Moreover, very often each mill has its own speed n . In this case in the equation (12.6) the expression:

$$G = n (1 - 0.06 nD) LD^2 \quad (G = n (1 - 0.018 nD) LD^2)$$

will be replaced by the expression:

$$G_{\bar{m}} = \frac{(N_1 - 1) G_1 + G_2 + G^* + \dots + G_n}{2 (N_1 - 1)} \quad (12.10)$$

G_1, G_2, \dots, G_n being the values of the expression G for the 1st, 2nd, . . . , and n th mills, and $G_{\bar{m}}$ being then the value to take for the calculation of capacity of the tandem.

Equation (12.10) was that employed in Java in 1939¹⁰.

If we express the work of a mill as a function of the fibre loading, we obviously have:

$$Af = 60 \pi DnLq = 188.5 qnLD = 188.5 \tau nLD^2 \quad (12.11)$$

A = crushing rate of the mills, in kg (lb.) cane per hour

f = fibre in cane in relation to unity

L = length of rollers, in m (ft.)

D = diameter of rollers, in m (ft.)

ω = speed of rollers, in r.p.m.

q = fibre loading, in kg/m² (lb./sq.ft.)

r = specific fibre loading, in kg/m²/m (lb./sq.ft./ft.).

Equating this expression to the normal capacity of the tandem (eqn. 12.6) we have:

$$188.5 r n L D^2 = 900 c n (1 - 0.06 n D) L D^2 \sqrt{N} \quad \text{(metric units)} \quad (12.12)$$

$$r = 4.774 c (1 - 0.06 n D) \sqrt{N}$$

$$188.5 \tau n L D^2 = 0.0255 \times 2.240 c n (1 - 0.018 n D) L D^2 \sqrt{N} \quad \text{(British units)} \quad (12.12)$$

$$\tau = 0.303 c (1 - 0.018 n D) \sqrt{N}$$

c = coefficient relating to preparatory plant

N = number of rollers in the tandem.

This expression indicates the normal values which may be given to T in order to operate the mill under normal conditions (see Table 12.5).

TABLE 12.5

NORMAL VALUES OF SPECIFIC FIBRE LOADING r AS A FUNCTION OF COMPOSITION OF TANDEM, IN kg/m²/m

c	nD	N					
		12	14	15	17	18	21
1	4	12.57	13.58	14.05	14.96	15.40	16.63
	5	11.45	12.51	12.94	13.78	14.18	15.32
1.1	4	13.83	14.34	15.46	16.46	16.94	18.29
	5	12.74	13.76	14.22	15.16	15.60	16.85
1.22	4	15.33	16.57	17.15	18.25	18.78	20.29
	5	14.12	15.26	15.79	16.81	17.40	18.69

(For values in lb./cu.ft., multiply by 0.0624.)

Capacity ratio of mills

The capacity ratio θ of a milling tandem is the ratio between the tonnage by the tandem and the tonnage of which it is theoretically capable. We have just calculated these two values: Tonnage actually crushed:

$$A_f = 188.5 \tau n L D^2 \text{ kg fibre/h (lb.fibre/h)}$$

Theoretical standard tonnage which the tandem should normally crush:

$$A_0 f = 900 c n L D^2 \sqrt{N} (1 - 0.06 n D) \quad (A_0 f = 57 c n L D^2 \sqrt{N} (1 - 0.018 n D)) \quad (12.13)$$

The capacity ratio then is:

$$\theta = \frac{A}{A_0} = \frac{0.21 T}{c \sqrt{N} (1 - 0.06 nD)} \quad \left| \quad \theta = \frac{A}{A_0} = \frac{3.3 \tau}{c \sqrt{N} (1 - 0.018 nD)} \right| \quad (12.14)$$

This measures the relative effort demanded of the tandem:

If $\theta < 1$, the tandem is under capacity

If $\theta = 1$, the tandem is working at normal capacity

If $\theta > 1$, the tandem is overloaded, or working above normal capacity.

For a given tandem, c , N and D are fixed, n generally varies only between very narrow limits, and the denominator of eqn. (12.14) may be calculated. For example: $c = 1.22$, $N = 15$, $nD = 5$ (metric units); hence:

$$c \sqrt{N} (1 - 0.06 nD) = 3.3$$

$$T = 4.77 \times 3.3 = 15.79$$

Hence:

$$\theta = \frac{0.21 \times 15.79}{3.3} = 1$$

and it is seen that, for a given tandem, the specific fibre loading measures, to a close approximation, the capacity ratio.

TONNAGE RECORDS

The highest capacities recorded, to our knowledge are the following:

(a) Tonnage of cane crushed in 24 hours by a factory: 24,000 tonnes, average for 1979 at Sao **Martinho**¹.

(b) Capacity of one tandem: 12,613 tonnes (t.c.) in 24 hours, by Bryant in Florida², in 1978, with 6 mills 990 x 2,134 mm (39 x 84 in.), or approx. 525 t.c.h.

(c) Tonnage of cane crushed in one season: 2,781,021 tonnes (t.c.), by Central Romana³, San Domingo.

(d) Year's production of sugar by a factory: 295,939 tonnes, by Central Romana³, in 1960.

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- 13 T. O. ELLIS, *Sugar J.*, 25 (5) (1962) 11.

13. Mill Settings

OBJECT

"Setting" a mill consists of selecting the most favourable relative positions to be given to the 3 rollers and the trashplate, in order to obtain the best conditions for feeding and the best extraction results.

Every mill has provision for adjustments (Fig. 13.1) permitting alterations to be made to the position at rest of each of the two lower rollers relative to the top roller. The relative positions with the mill empty determine those when operating, when the top roller lifts against the hydraulic pressure, under the thrust of the bagasse.

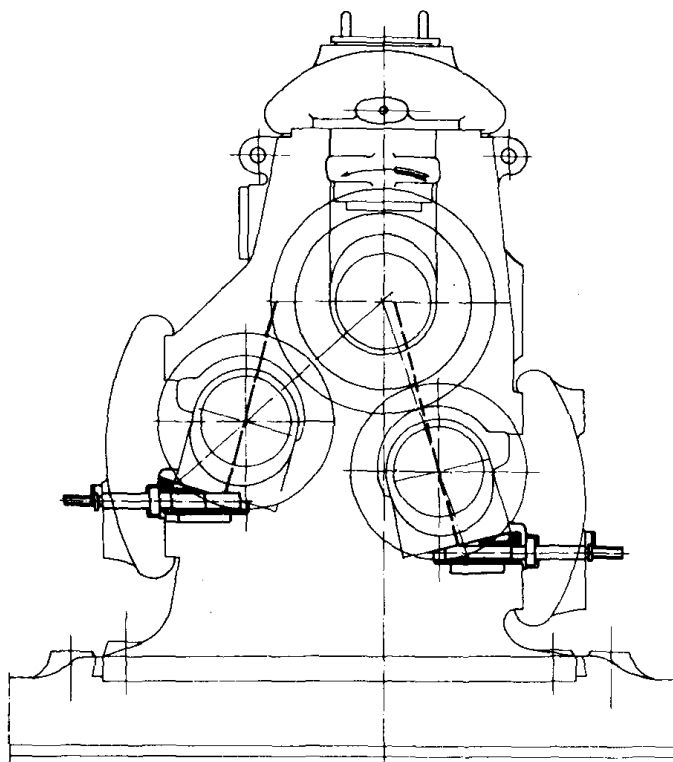


Fig. 13.1. Mill housing showing wedges and screws for adjustment of rollers (Fives Cail—Babcock).

Adjustment of settings is a delicate and important operation. A good setting is indispensable for obtaining good operation of the mill, that is, regular operation without choking and with a suitable extraction.

The settings include two main adjustments:

- (A) The setting of the feed and delivery openings
- (B) The setting of the trashplate.

A. Feed and Delivery Openings

The front roller is a feeding apparatus; the delivery roller a pressure device. The feed opening should obviously be greater than the delivery opening: if they were made equal, either the output of the mill would be ridiculously low (small openings), or the pressure would be very poor (large openings).

The three-roller mill would in fact not be justified: its success and its universal adoption are due to the fact that, due to the feed-roller pressure and the go-between of the *trashplate*, it permits of delivery-roller pressures which would be unobtainable in a unit of two rollers.

On the other hand, if the feed opening were much too wide, the bagasse arriving at the delivery roller would be much too wet, and the mill would choke or would give very poor results.

There is then an optimum ratio between the two openings, for a given set of conditions. It is a question of determining this ratio.

This problem does not allow of a mathematical solution: the adjustment is in reality an *art*, based on empirical considerations. But it involves some calculations, if logical and optimal solutions are to be attained.

It is always the delivery opening that is calculated first; the feed opening is next calculated according to the value found.

MEASURE OF THE OPENINGS

The openings are expressed as those between the mean diameters of the rollers considered.

When the grooving of the two rollers is similar, the opening is expressed very simply: the opening is represented by the distance between any two points on the surface of the two rollers, situated in their common axial plane and in the same plane perpendicular to the two axes (Fig. 13.2).

When the two rollers have different grooving, care must be taken to measure the mean diameters which are represented for each roller by the mean of the diameter at the tips of the teeth and that at the bottom of the grooves (Fig. 13.3).

With complex grooving, the mean diameter is calculated as that defined by the straight line which, on a longitudinal section of the roller, intercepts equal full and empty areas. In other

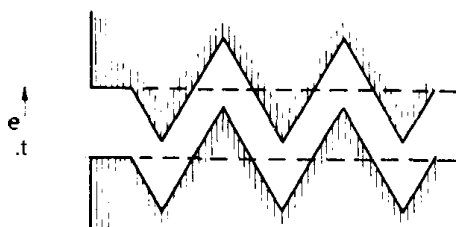


Fig. 13.2. Measure of openings with equal grooving.

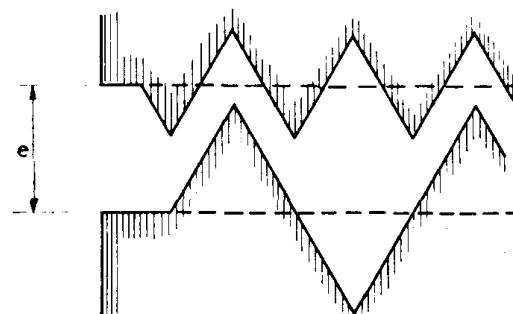


Fig. 13.3. Measure of openings with different grooving.

words, the mean diameter is the diameter of the geometric cylinder having the same length and the same volume as the mill roller under consideration.

However, the Messchaerts should not be taken into account in the determination of mean diameter by calculation or drawing.

JAVA METHOD

The established method of determining settings is that of the Experiment Station of Java. The method adopted is the following:

- (1) Choice of mill speeds
- (2) Calculation of the delivery opening
- (3) Calculation of the feed opening.

1. Choice of mill speeds

It was considered in Java that mill speeds were not unimportant. We have already seen (p. 182) that the Javan engineers had definite ideas on the value as well as on the sequence of speeds in a milling plant.

The Experiment Station had accordingly established a table of average values for Java of the thickness of bagasse blanket in each type of tandem, from the first to the last mill. This was expressed in grams of fibre per square decimetre of area described by a generatrix of the roller (or "scribed area").

We may comment in passing that we find again here, in a different unit, our concept of the fibre loading, the use of which in Java, long before our mention of it, confirms its importance. We have:

$$1 \text{ g/dm}^2 = 100 \text{ g/m}^2 = 0.1 \text{ kg/m}^2 (= 0.0205 \text{ lb./sq.ft.})$$

The Java table gave this value for the standard mill of 762 x 1,524 mm (30 x 60 in.). For a mill of different diameter, we should have, following the principle that thickness of bagasse is proportional to diameter:

$$\frac{q'}{Q} \approx \frac{D'}{D}$$

The Experiment Station recommended that a mill speed should be taken which would give a bagasse layer corresponding to that shown in the table. The calculation is very easy. We illustrate it by an example:

Example. Tandem of 15 rollers 813 x 1,675 mm (32 x 66 in.), which is required to crush 120 t.c.h. at 14% fibre. The speed of the 3rd mill is required.

The table (Table 13.1) gives $q = 157 \text{ g/dm}^2$. For a roller of 813 mm, this corresponds to:

$$q' = 157 \times \frac{813}{762} = 168 \text{ g/dm}^2$$

The weight of fibre to be treated is:

$$120,000 \times 0,14 = 16,800 \text{ kg/h}$$

Hence the area S to be escribed by the roller:

$$\frac{16,800,000}{S} = 168 \text{ g/dm}^2 \quad S = 100,000 \text{ dm}^2/\text{h}$$

Now:

$$S = 60 n \pi DL = 60 \times n \pi \times 8.13 \times 16.75 = 100,000$$

whence:

$$n = 3.9 \text{ r.p.m.}$$

This speed is low for the tonnage planned. We have already commented that the speeds practised in Java were low, particularly in the later mills of the tandem.

It must be added, however, that the Java method would be equally applicable to mills operating at speeds higher than those given by Table 13.1, the important point being the fibre index rather than the fibre loading.

2. Calculation of delivery opening

It is now a matter of fixing the delivery opening with the mill at rest. For this we begin by determining the opening during operation.

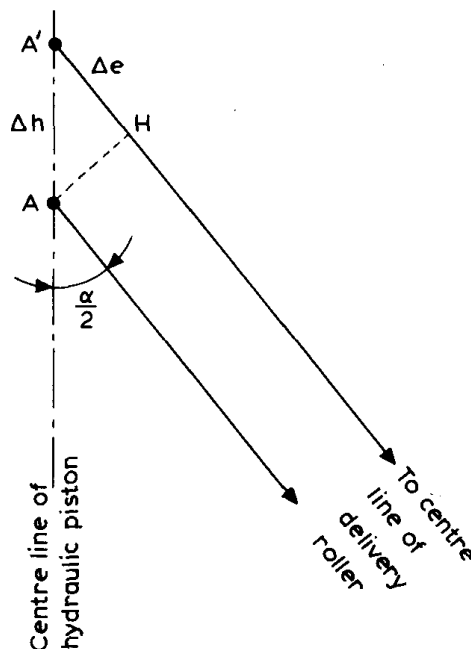


Fig. 13.4. Lift of accumulators.

The Experiment Station similarly furnishes a table of mean values recommended for the weight of fibre passed through unit volume of delivery opening while operating, in other words, the fibre index for each mill.

The complete table is given in Table 13.1. The Station recommended adoption of a delivery opening while operating which would furnish a weight of fibre per unit volume equal to that given by the table.

TABLE 13.1

MILL SETTINGS IN JAVA

<i>Tandem</i>	<i>4 mills</i>	<i>Crusher + 4 mills</i>	<i>Crusher + shredder + 4 mills</i>	<i>5 mills</i>	<i>Crusher + 5 mills</i>
<i>q = g fibre per dm² of escribed roller surface (D = 762 mm)</i>					
Crusher	—	113	115	—	107
1st mill	96	130	137	101	132
2nd mill	135	150	143	143	147
3rd mill	144	167	156	157	166
4th mill	166	179	159	168	174
5th mill	—	—	—	183	189
<i>φ = g fibre per dm³ escribed volume delivery (fibre index)</i>					
Crusher	—	390	330	—	320
1st mill	530	610	570	370	620
2nd mill	610	750	670	640	720
3rd mill	570	840	720	800	810
4th mill	620	940	770	910	870
5th mill	—	—	—	960	960

Example. Continuing the calculation of the previous example, the operating delivery work opening will have a value of:

$$e_A = \frac{q'}{\phi} = \frac{170}{800} = 0.21 \text{ dm} = 21 \text{ mm}$$

From the operating delivery opening, the opening "at rest" is derived by assuming that the mean lift of the top roller has an optimum value of 20% of the mean value of the operating opening (method of Helmer); 30% is also sometimes recommended¹, a value which would appear preferable. In the foregoing example, we would have then:

$$\text{Desired lift of the top roller} = Ah = 21 \times 0.30 = 6.3 \text{ mm}$$

whence (cf. Fig. 13.4):
corresponding change in the delivery opening:

$$Ae = Ah \cos \frac{\alpha}{2} = 6.3 \times 0.8 = 5 \text{ mm}$$

Hence:

$$\text{Delivery opening, after deducting lift} = S' = 21 - 5 = 16 \text{ mm}$$

The method employed in Java took full account of the corrections necessary to convert from theoretical openings (measured from top of tooth to bottom of groove) to true openings. In particular, the empty volume due to chevrons and to wear and breakages of teeth was determined by a test with a piece of clay, leading to the same results as the procedure with lead which we have recommended (*cf.* p. 124). The wear of the rollers at the centre was likewise taken into account. All these corrections were deducted from the value chosen for the operating opening when deciding the setting "at rest" to be employed; the corrections for wear at the centre and for the free volume were then deducted from the value obtained. If these totalled 4 mm, we would have:

$$\text{Delivery setting for mill at rest: } s_0 = 16 - 4 = 12 \text{ mm}$$

3. Calculation of feed opening

The feed opening during operation is deduced immediately from the delivery opening, the ratio between the two openings being fixed at the values given in Table 13.2:

TABLE 13.2

RATIO BETWEEN FEED AND DELIVERY OPENINGS (WHEN OPERATING) IN JAVA

For 1st mill:	2.2 if preceded by a crusher, 2.6 otherwise
For 2nd mill:	1.9
For 3rd mill:	1.9
For 4th mill:	1.8

Example. Continuing the preceding example, the operating opening at the feed roller will have a value of:

$$E = 1.9 e_A = 40 \text{ mm}$$

Hence the opening at rest (in the case of standard symmetrical housings):

$$\text{Feed opening, deducting lift} = a' = 40 - 5 = 35 \text{ mm}$$

and, with the corrections for wear at the centre and for free volume, taken as 6 mm for the feed roller:

$$\text{Feed setting required for mill at rest} = a_0 = 35 - 6 = 29 \text{ mm}$$

Application of the Java method in other countries

The disadvantage of this method is that it is tied to the Java school of speeds. Its application, based on the Java table (Table 13.1), leads to low and decreasing speeds. However, one may not be a partisan of this school of thought, and in consequence the mills also may not be designed for it. The French manufacturers, in particular, do not provide, for their steam-engine drives, governor pulleys permitting of such low speeds, and their steam engines, electric motors and turbines would be too powerful and would function below economical conditions. Finally, tandems with combined drive do not permit of giving each mill a particular speed, which the Javan table demands.

TABLE 13.3

SETTINGS FOR MILLS IN QUEENSLAND (JAVA METHOD)

Tandem	3 mills		4 mills		5 mills	
	<i>I</i>	<i>q'</i>	<i>I</i>	<i>q'</i>	<i>Q</i>	<i>q'</i>
1st mill	112	2.3	117	2.4	102	2.1
2nd mill	107	2.2	122	2.5	137	2.8
3rd mill	102	2.1	132	2.6	112	2.3
4th mill			122	2.5	127	2.6
5th mill					97	2.0
	ϕ	ϕ'	ϕ	ϕ'	ϕ	ϕ'
1st mill	450	28	384	24	336	21
2nd mill	512	32	464	29	450	28
3rd mill	800	50	545	34	480	30
4th mill			672	42	496	31
5th mill					672	42

q = g fibre per **dm**² roller surface (*D* = 762 mm) in.
q' = **lb.** fibre per sq.ft. escribed surface (*D* = 30)
Q = g fibre per **dm**³ delivery opening
 ϕ = **lb.** fibre per cu.ft. escribed volume (delivery).

However, certain countries have adopted the principle of the Java method, while modifying the table. Table 13.3 gives the table constructed by Behne, following the settings practised in Queensland².

This table was established for mills with fixed top roller, which were frequent at the time in Queensland. Settings for these would not be the same as for mills with hydraulics.

Russell³ comments that, in 1968, most Australian factories, independent of the number of mills in the tandem, used figures of $\phi = 486 \text{ g/dm}^3$ (30.3 lb./cu.ft.) for the first mill, and 821 (51.3) for the last, intermediate mills having figures between these values.

Optimal values for the Java table

Douwes Dekker and Van Hengel⁴ have commented that the best figures for fibre in bagasse, and hence the best individual mill extractions, were obtained in Java by mills in which the delivery opening when operating was designed for a ratio $\phi/f' = 1.75 \text{ kg/dm}^3$ (109 lb./cu.ft.).

Now we have (cf. p. 142):

$$\phi = \frac{Af}{60 \pi nDL e_A} \tag{13.1}$$

Hence, replacing ϕ by its optimal value of $1.75f$ (109 f):

$$e_A = \sim 330 \frac{Af}{nDLf} \text{ dm} \quad \left(e_A = \frac{Af}{20,600 nDLf'} \right) \quad (13.2)$$

f' being fibre per unit of bagasse for the mill considered. Values for f' will preferably be chosen as suggested by Table 13.7.

From the opening while operating, e_A , thus derived, we calculate the opening for the empty mill, so, as indicated above (p. 203), taking for average lift 30% of the maximal lift possible.

We may comment that the value thus recommended by Douwes Dekker and Van Hengel gives, for a 15-roller tandem, and for the fibre figures of Table 13.6, fibre indices of 530, 705, 785, 865, and 915 kg/m^3 (33, 44, 49, 54, and 57 lb./cu.ft.), respectively, for mills 1 to 5.

We may comment also that these values adopted in the Java method for the fibre index fix the re-absorption factor r . Taking for example the bagasse from the first mill, with $\phi = 530 \text{ kg/m}^3$ (33 lb./cu.ft.), the weight of bagasse (of 0.30 fibre content) corresponding to this weight of fibre will be $530/0.30 = 1,765 \text{ kg}$ ($33/0.30 = 110 \text{ lb.}$). This 1,765 kg (110 lb.) of bagasse will occupy a no-void volume (*cf.* p. 134) of $(1,765 \times 0.873)/1,000 = 1.54 \text{ m}^3$ ($110 \times 0.873/62.4 = 1.54 \text{ cu.ft.}$). If this is to pass through 1 m^3 (1 cu.ft.) of delivery opening, the re-absorption factor must be $r = 1.54$; a similar calculation for the 5 mills would give a re-absorption factor ranging from 1.54 to 1.43 from the first to the last mill. Actually, the opening derived by this calculation is not necessarily that assumed by the mill, the top roller of which will lift more or less than estimated. The re-absorption factor accordingly may be different.

Nevertheless, we still regard the values recommended by Douwes Dekker and Van Hengel as very valuable.

METHOD OF CALCULATING OPERATING OPENINGS

We shall give the calculations in the same order as previously.

1. Choice of mill speeds

We attach to the speed of the mills only a secondary importance, on the condition, of course, that it remains between normal limits (*cf.* p. 181). The speed is nearly always fixed by the necessity to attain a given tonnage.

The optimal value is given by:

$$1,000 Af = 60 n_0 \pi DLq_0 = 60 \pi n_0 D^2 L \tau_0 \quad (2,240 Af = 60 n_0 \pi DLq_0 = 60 \pi n_0 D^2 L \tau_0)$$

whence:

$$n_0 = 5.3 \frac{Af}{LD^2 \tau_0} \quad \left(n_0 = 11.88 \frac{Af}{LD^2 \tau_0} \right) \quad (13.3)$$

n_0 = optimal speed, in r.p.m.

A = crushing rate required of the tandem, in t.c.h.

f = fibre per unit of cane

L = length of rollers, in m (ft.)

D = diameter of rollers, in m (ft.)

q_0 = optimal fibre loading, in kg/m² (lb./sq.ft.)

TO = optimal specific fibre loading, given by eqn. (12.12), in kg/m²/m (lb./sq.ft./ft.).

If, for any reason, this optimal speed cannot be realised, we shall simply endeavour to approach it as closely as possible, and adopt the speed which we are able to attain, even if it be appreciably different.

Further, this speed may be made constant throughout the tandem, or decreasing from the first to last mill, or better, slightly increasing.

2. Calculation of operating delivery opening

We shall illustrate the calculation by a concrete example.

Example. It is required to calculate the operating delivery openings for a 15-roller tandem of 990 X 2,134 mm (39 X 84 in.), following three sets of knives and a shredder:

Crushing rate required: $A = 300$ t.c.h. with fibre $f = 0.15$

Speed: $n = 6$ r.p.m.

Imbibition: $X = 2$ distributed between the last two mills (*cf.* eqn. (19.6)).

Sequence required for fibre contents of successive bagasses: 33 – 42 – 46 – 48 – 50.

We assume that the brix of the residual juices in the successive bagasses are those found in the calculation in the second edition (pp. 282 – 285):

$$B_1 = 18.74 \quad \text{density} = 1.075 \quad \text{density under pressure} = 1.01 \times 1.075 = 1.086$$

$$B_2 = 16.28 \quad \text{density} = 1.065 \quad \text{density under pressure} = 1.01 \times 1.065 = 1.076$$

$$B_3 = 12.63 \quad \text{density} = 1.049 \quad \text{density under pressure} = 1.01 \times 1.049 = 1.059$$

$$B_4 = 8.71 \quad \text{density} = 1.033 \quad \text{density under pressure} = 1.01 \times 1.033 = 1.043$$

$$B_5 = 6.32 \quad \text{density} = 1.023 \quad \text{density under pressure} = 1.01 \times 1.023 = 1.033$$

We now calculate the volumes and the densities for the various bagasses under pressure (*cf.* eqns. (10.15) and (10.17)):

$$\begin{aligned} \text{1st mill: } V_0 &= \frac{1}{1.000} - \left(\frac{1.2}{1.086} - 0.86 \right) 0.33 = 0.8400 \\ d_1 &= \frac{1}{0.8400} = 1.190 \text{ kg/m}^3 \text{ (74.3 Ib./cu.ft.)} \end{aligned}$$

$$\begin{aligned} \text{2nd mill: } V_0 &= \frac{1}{1.076} - \left(\frac{1.2}{1.076} - 0.86 \right) 0.42 = 0.8222 \\ d_2 &= \frac{1}{0.8222} = 1.216 \text{ kg/m}^3 \text{ (75.9 Ib./cu.ft.)} \end{aligned}$$

$$\begin{aligned} \text{3rd mill: } V_0 &= \frac{1}{1.059} - \left(\frac{1.2}{1.059} - 0.86 \right) 0.46 = 0.8187 \\ d_3 &= \frac{1}{0.8187} = 1.221 \text{ kg/m}^3 \text{ (76.2 Ib./cu.ft.)} \end{aligned}$$

$$4th \text{ mill: } V_0 = \frac{1}{1.043} - \left(\frac{1.2}{1.043} - 0.86 \right) 0.48 = 0.8193$$

$$d_4 = \frac{1}{0.8193} = 1.221 \text{ kg/m}^3 \text{ (76.2 lb./cu.ft.)}$$

$$5th \text{ mill: } V_0 = \frac{1}{1.033} - \left(\frac{1.2}{1.033} - 0.86 \right) 0.50 = 0.8172$$

$$d_5 = \frac{1}{0.8172} = 1.224 \text{ kg/m}^3 \text{ (76.4 lb./cu.ft.)}$$

It now remains to fix a value for the re-absorption factor; but we can only form an approximate idea of its value. We shall not make use, for this, of the formulae (10.74) and (10.75), on account of the uncertainty of the constant term and of the coefficients v and ϕ in the complete formula. We obtain an approximation to the factor by assuming that these two factors play an equal role, in other words, by taking the mean between the results of eqns. (10.73) and (10.75).

The former gives: $r' = 1.06 + (0.017 \times 18.66) = 1.377$ for all mills.

For the latter, we assume as a probable and desirable fibre-index value that suggested by Douwes Dekker and Van Hengel: $0 = 1.75f'$, from which: $0.65 \phi = 1.1375 f'$.

Hence:

$$1st \text{ mill: } r_1'' = 0.87 + (1.1375 \times 0.33) = 1.245 \quad r = \frac{1}{2} (1.377 + 1.245) = 1.311$$

$$2nd \text{ mill: } r_2'' = 0.87 + (1.1375 \times 0.42) = 1.348 \quad r = \frac{1}{2} (1.377 + 1.348) = 1.362$$

$$3rd \text{ mill: } r_3'' = 0.87 + (1.1375 \times 0.46) = 1.393 \quad r = \frac{1}{2} (1.377 + 1.393) = 1.385$$

$$4th \text{ mill: } r_4'' = 0.87 + (1.1375 \times 0.48) = 1.416 \quad r = \frac{1}{2} (1.377 + 1.416) = 1.396$$

$$5th \text{ mill: } r_5'' = 0.87 + (1.1375 \times 0.50) = 1.439 \quad r = \frac{1}{2} (1.377 + 1.439) = 1.408$$

We must have no illusions as to the precision of these values for the re-absorption factor. However, we have no way of estimating them more accurately, and a more accurate figure would not appreciably alter the value of the setting which we require. We thus have to be content with this approximation.

We now write that the volume of bagasse passing through the delivery opening, multiplied by its density and the re-absorption factor, gives the weight of bagasse:

$$60 n \pi DL e_A dr = B = A \frac{4}{J} \quad (13.4)$$

hence:

$$e_A = \frac{Af}{60 \pi n DL} \cdot \frac{1}{f dr} = \frac{Af}{60 \pi n DL} \cdot \frac{1}{\phi} = \frac{\sigma}{\phi} \quad (13.5)$$

with:

$$q = \frac{Af}{60 \pi n DL} = \frac{300.000 \times 0.15}{60 \times 3.1416 \times 6 \times 9.9 \times 21.34} = \dots \quad 5 \text{ lb./sq.ft.)}$$

Hence (eqns. (10.40), (13.5)):

$$\begin{aligned}
 \text{1st mill: } fa &= 0.33 \times 1.190 \times 1.311 = 0.515 & e_A &= \frac{0.188}{0.515} = 0.36 \text{ dm} = 36 \text{ mm} \\
 \text{2nd mill: } fa &= 0.42 \times 1.216 \times 1.362 = 0.696 & e_A &= \frac{0.188}{0.696} = 0.27 \text{ dm} = 27 \text{ mm} \\
 \text{3rd mill: } \phi_3 &= 0.46 \times 1.221 \times 1.385 = 0.778 & e_A &= \frac{0.188}{0.778} = 0.24 \text{ dm} = 24 \text{ mm} \\
 \text{4th mill: } \phi_4 &= 0.48 \times 1.221 \times 1.396 = 0.818 & e_A &= \frac{0.188}{0.818} = 0.23 \text{ dm} = 23 \text{ mm} \\
 \text{5th mill: } fa &= 0.50 \times 1.224 \times 1.408 = 0.862 & e_A &= \frac{0.188}{0.862} = 0.22 \text{ dm} = 22 \text{ mm}
 \end{aligned}$$

If we had used the values for ϕ from Java (Table 13.1) or from Russell (Table 13.3), we should have had:

	ϕ Java	ϕ Russell	e_A Java	e_A Russell	e_A calculated
1st mill:	$\phi_1 = 370$	486 g/dm ³	51	39	36 mm
2nd mill:	$fa = 640$	570 g/dm ³	29	33	27 mm
3rd mill:	$fa = 800$	654 g/dm ³	24	29	24 mm
4th mill:	$fa = 910$	737 g/dm ³	21	26	23 mm
5th mill:	$fa = 960$	821 g/dm ³	20	23	22 mm

Though differing somewhat due to the uncertainty of the value of the re-absorption factor, these figures are quite useful in practice.

3. Delivery opening with mill empty

Knowing the openings when operating, we now proceed to fix their values with the mill at rest.

It is customary to adopt a value as large as possible, but nevertheless sufficiently small to avoid the risk of the top roller working on its lower bearings when fluctuations in feed reduce the blanket of bagasse. A safety coefficient is therefore adopted, and the practice is to allow a greater margin of safety when there are fewer mills following the one under consideration. We designate this safety coefficient by ψ :

$$\psi = \frac{\text{delivery opening when empty}}{\text{average delivery opening when working}} \quad (13.6)$$

This coefficient ψ is not of great importance and consequently varies widely. Its only object in effect is to assure that the bagasse will be constantly under full pressure. The margin of safety which it allows should be greater when the throughput of the mill is liable to be irregular.

Table 13.4 gives values of this coefficient as advised by Farrel (1971), the values ranging from 0.773 for the first mill to 0.435 for the last, and varying according to the number of mills in the tandem.

TABLE 13.4

RATIO ψ OF EMPTY TO OPERATING VALUES OF DELIVERY OPENING
(FARREL)

<i>Tandem</i>	<i>4 mills</i>	<i>5 mills</i>	<i>6 mills</i>	<i>7 mills</i>
1st mill	0.773	0.773	0.773	0.773
2nd mill	0.657	0.657	0.690	0.707
3rd mill	0.555	0.555	0.594	0.619
4th mill	0.473	0.473	0.524	0.555
5th mill		0.435	0.473	0.505
6th mill			0.435	0.455
7th mill				0.435

Table 13.5 gives values recommended by Fives Cail – Babcock and by the author.

TABLE 13.5

RATIO ψ OF EMPTY TO OPERATING VALUES OF DELIVERY
OPENING (FIVES CAIL - BABCOCK, FCB; AUTHOR, EH)

	<i>FCB</i>	<i>EH</i>
Crusher		0.6
1st mill	0.65	0.5
2nd mill	0.4	0.4
Intermediate mills	0.5	0.35
Penultimate mill	0.5	0.3
Final mill	0.45	0.25

Continuing *the* previous example, we take as delivery settings (empty):

$$1st\ mill: s_0 = 36 \times 0.50 = 18\ mm\ (0.70\ in.)$$

$$2nd\ mill: s_0 = 27 \times 0.40 = 11\ mm\ (0.43\ in.)$$

$$3rd\ mill: s_0 = 24 \times 0.35 = 8\ mm\ (0.33\ in.)$$

$$4th\ mill: s_0 = 23 \times 0.30 = 7\ mm\ (0.27\ in.)$$

$$5th\ mill: s_0 = 22 \times 0.25 = 5\ mm\ (0.22\ in.)$$

4. Calculation of feed opening

We return here to the Java solution, which consists of fixing a certain ratio m of the operating openings at feed and delivery:

$$m = \frac{\text{mean feed opening in operation}}{\text{mean delivery opening in operation}} \quad (13.7)$$

Table 13.6 indicates values recommended for the ratio m by Farrel, by Fives Cail – Babcock (FCB) and by the author (EH).

TABLE 13.6

RATIO m OF FEED TO DELIVERY OPENINGS IN OPERATION

	<i>Farrel</i>	<i>FCB</i>	<i>EH</i>
1st mill	2	2.3–2.5	2.2
2nd mill	2.2	2.2–2.3	1.9
3rd mill	2.4	2.1	1.8
4th mill	2.6	2	1.7
5th mill	2.8	1.9	1.7
6th and later mills	3	1.8	1.7

It will be seen that there is a very wide divergence between the different values recommended. This arises to a great degree from the predominance given in the American hemisphere to capacity rather than extraction.

From the operating feed openings thus determined, we return to consideration of the empty openings with the comment that, in mills of typical design with vertical caps, the feed and delivery openings increase by the same amount as the roller lifts.

Example. Continuing our example, we have the feed openings while operating:

$$1st\ mill: E_A = 36 \times 2.2 = 79\ mm\ (3.12\ in.)$$

$$2nd\ mill: E_A = 27 \times 1.9 = 51\ mm\ (2.00\ in.)$$

$$3rd\ mill: E_A = 24 \times 1.8 = 43\ mm\ (1.70\ in.)$$

$$4th\ mill: E_A = 23 \times 1.7 = 39\ mm\ (1.54\ in.)$$

$$5th\ mill: E_A = 22 \times 1.7 = 37\ mm\ (1.47\ in.)$$

The opening when empty is equal to the operating opening reduced by the difference between the operating and empty settings: $E_0 = s + E_A - e_A$.

Example.

$$1st\ mill: E_0 = 18 + 79 - 36 = 61\ mm\ (2.40\ in.)$$

$$2nd\ mill: E_0 = 11 + 51 - 27 = 35\ mm\ (1.38\ in.)$$

$$3rd\ mill: E_0 = 8 + 43 - 24 = 27\ mm\ (1.06\ in.)$$

$$4th\ mill: E_0 = 7 + 39 - 23 = 23\ mm\ (0.90\ in.)$$

$$5th\ mill: E_0 = 5 + 37 - 22 = 20\ mm\ (0.79\ in.)$$

Corrections for wear are to be added, as in the Java method. We may allow 1–2 mm for a new mill, 3–4 in an old mill.

Value of the ratio m in various countries. We have given the ratios adopted in Java (Table 13.2).

In Australia, Behne⁵ recommends ratios m varying from 2.3 to 2.6 for the first mill to 2.5–3.0 at the last mill.

In Louisiana, Lowe⁶ advises increasing gradually from 1.75 at the first mill to 2.25 at the last.

In Cuba, Maiz⁷ indicates ratios ranging from 1.80 at the first mill, 2.5 at the second, to 2.75 at the last, for a 6-mill tandem.

For the reasons given above, the author considers that the high values of m quoted in these countries are due to neglecting to calculate the openings when the roller begins to exert pressure (*cf.* p. 122). If these are taken into account and determined with care, it will be found that the values proposed in Table 13.6 are generally sufficient and that even lower values can be used. We have operated and obtained excellent results with heavily loaded final mills set with a ratio m of 1.6. If the mean lift is merely added to the opening at rest, erroneous and very high values of m will obviously be found.

Variation of fibre along the tandem

Calculation of the operating delivery opening (p. 207) is based on the fibre content of the bagasses leaving the various mills of the tandem. How does this factor vary along the tandem?

For correctly set mills, the fibre percentages of Table 13.7 may be taken:

TABLE 13.7

VARIATION OF FIBRE f' IN BAGASSE ALONG THE TANDEM

	<i>Crusher</i>	<i>1st mill</i>	<i>2nd mill</i>	<i>3rd mill</i>	<i>4th mill</i>	<i>5th mill</i>	<i>6th mill</i>
<i>Tandem of:</i>							
11 rollers	28	40	46	50			
14 rollers	28	40	44	48	51		
17 rollers	28	40	44	47	50	52	
<i>Tandem of:</i>							
12 rollers		33	42	47	51		
15 rollers		33	42	46	50	52	
18 rollers		33	42	45	48	50	52
<i>Douwes Dekker and Van Hengel⁸ suggest:</i>							
Tandem of 15 rollers		32	40	45	48	50	
Tandem of 18 rollers		30	39	43	46	48	50

Actually, these values vary slightly with the fibre and variety of cane, but there is little disadvantage in adopting the values indicated in Table 13.7, in the absence of more precise data.

DELIVERY OPENINGS AND FIBRE LOADING

The delivery openings during operation of the mill depend on the state of preparation and the degree of previous breaking up of the cane, on the hydraulic pressure (to some extent), on the re-absorption coefficient, but particularly on the fibre loading. If, for normal condi-

TABLE 13.8

VALUE OF RATIO e_A/q OF DELIVERY OPENING IN OPERATION TO FIBRE INDEX (q IN kg/m^2)

Tandem	N = 12			N = 14		N = 15			N = 17	N = 18		N = 21	
	FAR	FCB	EH	FCB	EH	FAR	FCB	EH	EH	FAR	EH	FAR	EH
Crusher	—	—	—	2.76	2.34	—	—	—	2.34	—	—	—	—
1st mill	2.62	2.07	1.90	1.80	1.53	2.62	2.07	1.90	1.53	2.62	1.90	2.62	1.90
2nd mill	1.73	1.41	1.45	1.34	1.36	1.73	1.46	1.45	1.36	1.97	1.45	2.03	1.45
3rd mill	1.31	0.91	1.28	0.90	1.22	1.31	0.97	1.30	1.23	1.48	1.32	1.56	1.32
4th mill	1.13	0.85	1.15	0.80	1.14	1.15	0.86	1.20	1.16	1.25	1.22	1.34	1.23
5th mill	—	—	—	—	—	1.05	0.79	1.14	1.13	1.13	1.16	1.20	1.17
6th mill	—	—	—	—	—	—	—	—	—	1.05	1.12	1.09	1.14
7th mill	—	—	—	—	—	—	—	—	—	—	—	1.05	1.12

(For opening in inches per lb./sq.ft. of fibre index, multiply the figures in the table by 0.1922.)

tions and the hydraulic pressures generally employed, we express the delivery openings while operating, e_A , as a ratio of the fibre loading, q , we shall obtain figures applicable to mills of all dimensions, and thus of general value, and sufficiently accurate for practical needs. These values have particularly the great advantage of avoiding the preceding calculations and of giving a practically immediate solution to the problem of determining settings.

Table 13.8 gives the values of the ratio e_A/q , as ascertained from the instructions of Farrel (FAR) and of Fives Cail – Babcock (FCB). We add those recommended by the author (EH).

The operating opening for any mill, therefore, may be immediately obtained by multiplying these values by the fibre loading q . For example, if $q = 10 \text{ kg/m}^2$ (2 lb./sq.ft.), the following will be the delivery openings for a 12-roller tandem: 19, 14.5, 12.8 and 11.5 mm (0.74, 0.58, 0.50 and 0.46 in.).

In South Africa, certain firms are content with calculating working openings by means of the simplified formula:

$$e_A = \frac{2.5 Cf}{nDLf'}$$

which reduces to:

$$e_A = 0.47 \frac{q}{f'} \quad \left(e_A = 0.09 \frac{q'}{f'} \right) \quad (13.8)$$

C = tonnage crushed, in t.c.h.

q = fibre index, in kg/m^2 (lb./sq.ft.).

To reconcile approximately with the author's figures, it is necessary at least in metric units to replace the coefficient 0.47 by 0.64 or the coefficient 2.5 by 3.4 (or better, 3.8, 3.4, 3.3, 3.2 and 3.1 for first to fifth mills, respectively). This formula gives very low values.

EFFECT OF INCLINED HOUSINGS

We have assumed (p. 211) that the feed and delivery openings increased by the same quantity when the top roller lifted. This is obvious in the case of the classical type of mill with vertical housings.

However, today several manufacturers build mill housings with the axis of the caps inclined towards the feed side (Fig. 13.5).

We shall proceed to calculate for this case the ratio of the increase dE of the feed opening to the increase de of the delivery opening:

$$\frac{dE}{de} = \mu \quad (13.9)$$

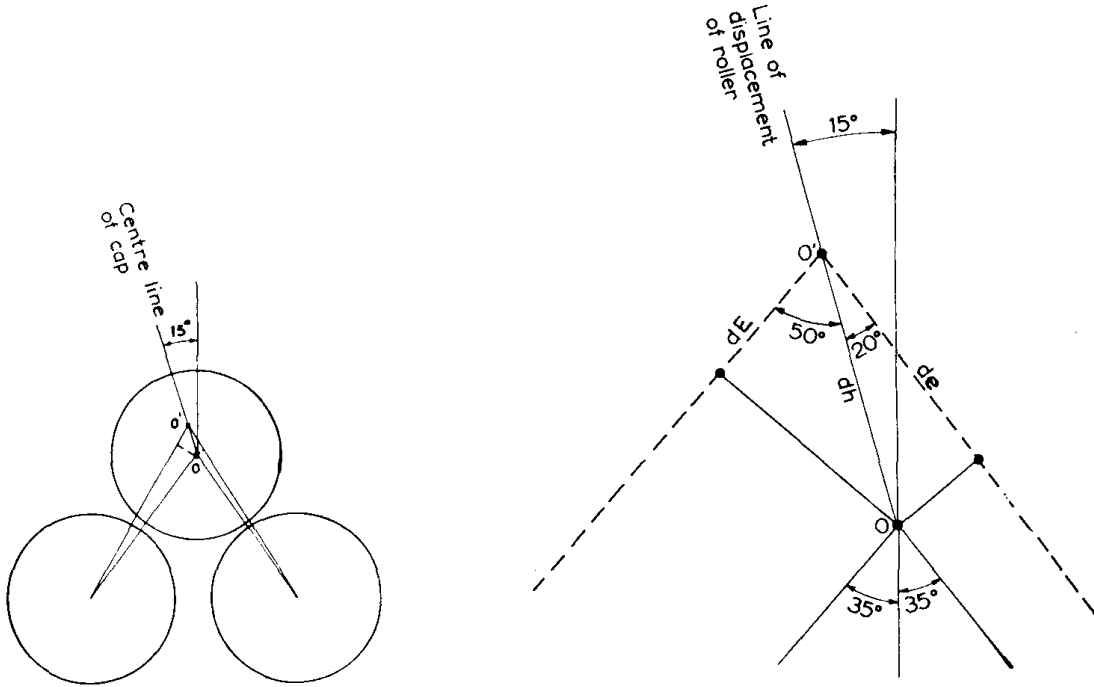


Fig. 13.5. Displacement of top roller in mill with inclined housing.

Fig. 13.6. Ratio of increase dE of feed opening to increase de of delivery opening (inclined housing).

In Fig. 13.6, let:

$a/2$ be the half-top-angle of the mill (about 35° in general, $33\frac{1}{2}$ for Fives Lille housings)

β be the inclination of the mill cap (15° in nearly all cases).

We have:

$$\frac{dE}{dh} = \cos\left(\frac{\alpha}{2} + \beta\right) \quad \frac{de}{dh} = \cos\left(\frac{\alpha}{2} - \beta\right)$$

whence:

$$\mu = \frac{dE}{de} = \frac{\cos\left(\frac{\alpha}{2} + \beta\right)}{\cos\left(\frac{\alpha}{2} - \beta\right)} \quad (13.10)$$

If $a/2 = 33.5^\circ$ and $\beta = 15^\circ$:

$$\mu = \frac{\cos 48.5^\circ}{\cos 18.5^\circ} = \frac{0.663}{0.948} = \mathbf{0.7}$$

It follows that, when the delivery opening increases or decreases by 25 mm (1 in.), the feed opening will increase or decrease by 18 mm (0.7 in.).

With the aid of this ratio, it is easy, in a mill with caps mounted in inclined housings, to determine the empty feed and delivery settings by means of the corresponding operating openings (*cf.* p. 211).

The coefficients m adopted for the standard case (Table 13.6) will, of course, be retained; and the greater values given for the feed settings "empty" by the new ratio $\mu < 1$ will be accepted.

Extent of hydraulic lift (Fives Cail-Babcock, 1973). Safety considerations have led certain manufacturers to provide for a maximum lift of the top roller. This lift depends not only on the dimensions of the mill. Provision for it should not be neglected, otherwise the passage of a stone or metallic object would risk serious breakage. Fives Cail-Babcock suggest calculating the normal lift of the roller as a function of the maximal lift, and suggest the following normal amounts of lift:

<i>Maximal lift:</i>	15	20	25	30	35	40	45 mm
<i>Normal lift: 1st mill:</i>	8	10	12	13	14	16	18 mm
<i>2nd mill:</i>	5	6	8	10	11	12	14 mm
<i>Following mills:</i>	4	5	1	9	10	10	10 mm

If the fibre index is given, the normal lift will fix the roller speed to adopt.

The lift adopted, multiplied by the coefficient of increase of opening when the roller lifts (0.8, 0.7, 0.96, 0.5, etc.), will give the difference between the working opening and the opening with the mill empty.

SETTINGS EMPTY AND OPENINGS IN OPERATION

We have just shown how to calculate the openings to be given to the rollers at rest; openings which are commonly designated by the name of "settings".

These settings for the empty mill, however, have **only** a secondary importance. What is especially important is the values assumed by the openings when the mill is operating. The values for the empty mill are of interest only in the case where the irregularity of feeding causes a decrease in thickness of bagasse blanket of the order of the coefficient ψ (Tables 13.4, 13.5). If the value $\psi = 0.3$, which we have proposed, is adopted for the intermediate mills, these mills will cease to exert their full pressure as soon as the bagasse blanket has decreased to 30% of its mean value.

In other words, the settings for the empty mill correspond to a margin of safety against irregularities in the feed.

Now, a great many persons responsible for the setting of mills do not realise this fact, and attach considerable importance to the feed and delivery settings of the mill *when empty*. It seems childish to repeat it, but a long experience has taught us that it is desirable: only the *difference* between the feed and delivery openings is of importance (we are discussing the case of standard housings).

Provided the margin of safety is exceeded, it is of little importance whether the mill has

been adjusted to a setting of 17/4 (feed opening, empty = 17 mm; delivery opening, empty = 4 mm) or 16/3, or 15/2, or 14/1, or 13/0 (*i.e.* 13 mm at feed, and metal to metal at delivery). In all these cases, the mill will function, in normal regular operation, with openings of 26/13 and, as long as the roller has not descended to 17/4, nothing will reveal whether its stop is located at 1, 2, or 9 mm lower.

Conversely, when the top roller rests on its stops at 17/4, it should rise as soon as the bagasse blanket has exceeded the thickness corresponding to 4 mm under the r.h.p. acting, and nothing will prevent it attaining 26/13 if a normal quantity of cane is supplied to it. It will not rise less readily nor float less readily if it starts from 13/0 than if it starts from 17/4.

Although the margins of safety provided by our coefficients ψ are sufficient, there would be no disadvantage in setting the mill at 13/0 instead of 17/4. This would not be a fault.

It is by no means the same if a setting of 17/0 is adopted instead of 17/4.

It is a rather frequent tendency with many engineers to close in the delivery roller. If this is done while maintaining the difference given by the calculation, it is not serious, but very generally the opposite is found, and the operation of the mill and its extraction suffer in consequence.

As it is peculiarly difficult to make this understood, it is preferable simply to insist upon the normal setting, which has been fixed by calculation and by practice.

We say "and by practice". Actually, the calculation furnishes interesting information, but only experience and practice will permit of a final decision on the exact value of the ratio m to be adopted for each mill. It may sometimes be advisable to modify slightly the values indicated in Table 13.6.

B. Trashplate

The position of the trashplate has a great influence on the smooth operation of the mill. Now, the correct setting of the trashplate is a delicate operation.

To define its position completely, it is necessary to know (Fig. 13.7):

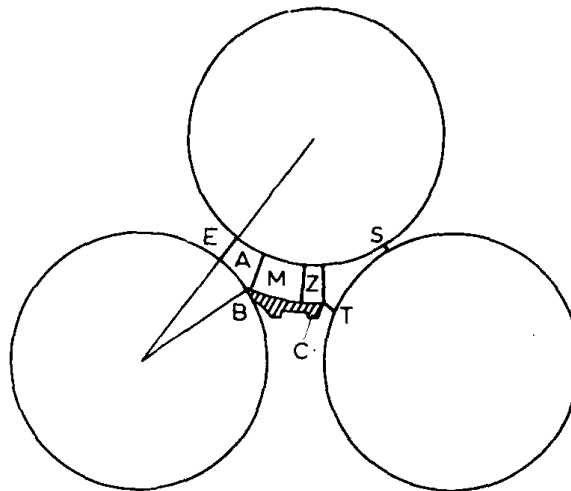


Fig. 13.7. Position of trashplate.

- (a) The position of the point B where the toe of the trashplate meets the feed roller
- (b) The dimension A , the distance from the toe of the plate to the top roller.
- (c) The dimension M , the distance from the plate to the top roller, measured in the vertical axial plane of the top roller
- (d) The dimension Z , the distance from the heel of the plate to the top roller
- (e) The dimension T , the distance from the heel to the delivery roller.

Form of trashplate

The bagasse which has just passed between the two feed rollers tends to recover its volume while passing from E to A (Fig. 13.7). If it be conceded that the trashplate should render possible the continuation of this expansion, the distance from the surface BC to the top roller should increase from B to C . If it is postulated that the radial velocity and the tangential velocity of a particle of bagasse should be constant in the course of its path across the plate, it may be readily shown that the curve BC should take the form of a logarithmic spiral with centre O (Fig. 13.8). This proof has been given by Bergmann in Java as early as 1889, and has been repeated by Deerr⁹. We do not propose to revive it.

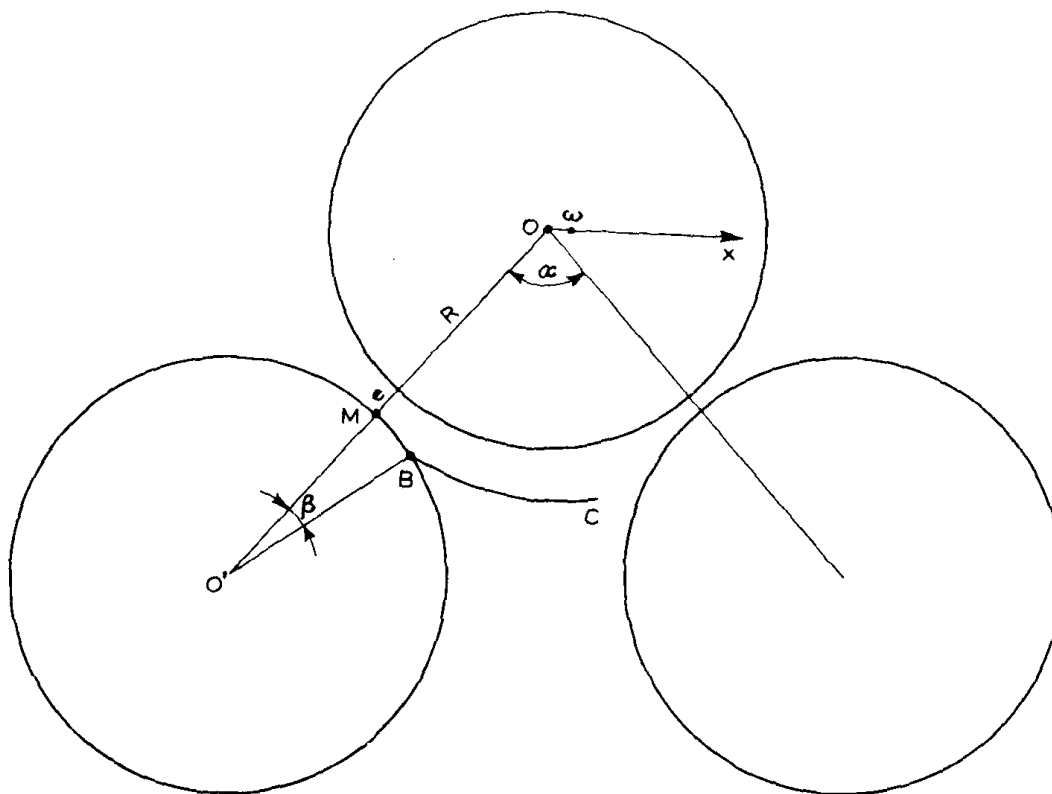


Fig. 13.8. Laying out position of the trashplate.

Setting out the trashplate

In practice, an arc of a circle is used as an approximation for the arc of the logarithmic spiral. Muller von Czernicky and Gogelein, in Java, have proposed constructions the best of which is the following.

In the horizontal Ox passing through O (Fig. 13.8), take in the direction shown:

$$O\omega = \frac{OM}{25} = \frac{R + E}{25}$$

South African companies take:

$$O\omega = \frac{R + E_0}{20}$$

but the centre of the circle is placed not at ω , but on the vertical through that point, ω' , at $0.0055 D$ mm above the point O , the angle β remaining at 13° .

The trashplate will be the circle drawn with ω as centre and with ωB as radius. The point B , where the toe of the trash should meet the feed roller, is obtained by taking:

$$MO'B = \theta = \frac{\alpha}{6} \quad (13.11)$$

If $\alpha = 72^\circ$, $\theta = 12^\circ$, the arc MB has a value of:

$$MB = \pi D \frac{\beta}{360} = \frac{\pi D \alpha}{6 \times 360} = 0.00145 Da \quad (13.12)$$

D = mean diameter of the roller (MB will be measured on a ridge, but will be transferred to the middle of the flank of a tooth, and the middle of the teeth of the trashplate will be made to pass through this point)

α = top angle of the mill, in degrees.

This diagram assumes that the top roller is lifting at its working position. Then E is the operating feed opening.

In Java, the settings were designed so that the working position of the top roller corresponded to a lift equal to 30% of the maximum possible.

To avoid having too fine a point at B , which would introduce a marked risk of bending the points, the toe of the plate is rounded (Fig. 13.9), but only slightly, in order to avoid formation of a pocket of bagasse at the toe of the plate.

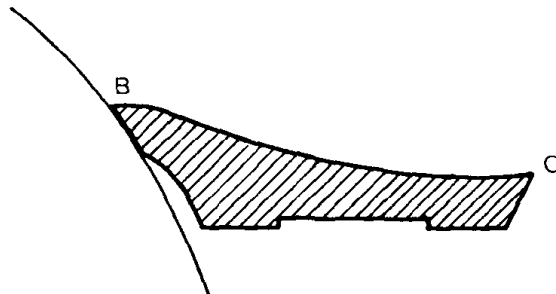


Fig. 13.9. Form of the trashplate.

Height of the trashplate

There remain to be fixed the dimensions A , M , Z , and T (Fig. 13.7).

These dimensions should be reckoned in relation to the *mean* diameters of the top and delivery rollers. It should be noted that foremen often make the measurement from the tips of the teeth, and that the values to give them in this case should be those calculated in relation to the mean diameter of the roller, reduced by half the height of the tooth.

Different authors give a confusion of diverse settings. We shall give first the standard method. Mean operating values, according to this method, relative to an *operating* feed opening of 100, are given in Table 13.9:

TABLE 13.9

TRASHPLATE SETTINGS

Feed roller opening	$E = 100$
Height from top roller to toe of trashplate	$A = 150$
Height in vertical axial plane of top roller	$M = 175$
Height of top roller at heel of trashplate	$Z = 190$
Distance from heel of trashplate to delivery roller	$T = 80$

In order to deduce, from the working values obtained from this table, the "empty" settings to be established, it is necessary to take the difference between the working opening E and its value empty, e_0 , to increase it by 20 — 25% (to take account of the fact that the dimensions A , M and Z increase practically to the same extent as Ah when the top roller rises by dh , whereas E increases only by $dh \cos a/2$), and deduct from each of the dimensions A , M and Z the measurement so obtained. The "empty" values A_0 , M_0 and Z_0 are thus obtained.

The dimension T does not change.

Example. Let us calculate the height of the trashplate for the first mill of our example of pp. 207 and 210. We have (*cf.* p. 210):

$$E_A = 79 \text{ mm (3.11 in.)}$$

Hence:

$$\begin{aligned} A &= 79 \times 1.50 = 118 \text{ mm (4.64 in.)} \\ M &= 79 \times 1.75 = 138 \text{ mm (5.43 in.)} \\ Z &= 79 \times 1.90 = 150 \text{ mm (5.91 in.)} \\ T &= 79 \times 0.80 = 63 \text{ mm (2.48 in.)} \end{aligned}$$

The difference between the feed opening under load and the empty setting chosen is (*cf.* p.211):

$$79 - 61 = 18 \text{ mm (0.71 in.)}$$

When this opening decreases by 18 mm, A and Z decrease by:

$$18 \times 1.2 \approx 22 \text{ mm (0.87 in.)}$$

and M by:

$$18 \times 1.25 \approx 23 \text{ mm (0.91 in.)}$$

Hence the settings empty are:

$$\begin{aligned} E_0 &= 61 \text{ mm (2.40 in.)} \\ A_0 &= 118 - 22 = 96 \text{ mm (3.78 in.)} \\ M_0 &= 138 - 23 = 115 \text{ mm (4.52 in.)} \\ Z_0 &= 150 - 22 = 128 \text{ mm (5.04 in.)} \\ T_0 &= 63 \text{ mm (2.48 in.)} \end{aligned}$$

Simplified settings

In Java, it was often considered satisfactory to locate the toe of the trashplate by the procedure indicated by Fig. 13.7. This fixed a certain value of dimension A . The heel of the plate was then simply located to make dimension Z equal to A increased by 4% of the width of the trashplate:

$$Z = A + 0.04 L \quad (13.13)$$

Z = distance from the top roller to the heel of the trashplate

A = distance from the top roller to the toe of the trashplate

L = width of the plate, all being expressed in the same units.

If $L = 30$ cm, we shall take:

$$Z = A + 1.2 \text{ cm}$$

Recommended setting

Generally, modern mills do not permit of ready adjustment of dimensions A , M and Z , as was possible previously. Once the relative positions of the 3 mill rollers are fixed, in the positions considered best, it is, in general, possible to alter the setting of the trashplate in 2 directions only: vertically and horizontally; that is, it can only be moved bodily, parallel to itself, up and down or forwards and backwards, the toe of the plate in contact with the feed roller (Fig. 13.13). In certain modern mills, this freedom of action is further reduced: the trashplate is set on an axis which itself is mounted in an eccentric (Fig. 18.4); it can then be raised or lowered by rotating the eccentric, and made to pivot about this axis, in order to bring it into contact with the feed roller. We must agree that these mills function as well as others, and that we have never had cause to regret the restrictions thus imposed on the settings, which moreover become easier, more rapid and more practical.

In these conditions, only one of the dimensions A , M or Z can be realised; in general, M

is selected, and it is the responsibility of the designer to design mills and trashplates in such a way that, for any reasonable value of M , the corresponding dimensions A and Z will have correct values, whatever the state of wear of the rollers. The engineer responsible for the settings should moreover verify that the dimension T is kept at a satisfactory value, neither too wide nor too narrow.

The height M of the trashplate, like the feed and delivery openings E and e (obviously we are considering the working openings), depends on the fibre loading, with only this difference: that, in a mill with hydraulics, the opening e depends directly on the fibre loading and adjusts its value immediately, whereas the opening E and the space M assume the values which the instantaneous position of the top roller gives them, but which also depend on the setting adopted. By thus relating the working heights of trashplate to the fibre index q , we obtain values applicable to tandems of any dimension and of all capacities.

Table 13.10 gives specific values expressed in mm per kg/m^2 of fibre index.

TABLE 13.10

RATIO M/q OF HEIGHT OF TRASHPLATE (mm) TO FIBRE INDEX q (IN kg/m^2)						
<i>Tandem</i>	$N = 12$	$N = 14$	$N = 15$	$N = 17$	$N = 18$	$N = 21$
1st mill	8	6.5	8	6.5	8	8
2nd mill	6	5.6	6	5.6	6	6
3rd mill	4.8	4.7	4.8	4.8	4.8	4.8
4th mill	4.5	4.4	4.5	4.5	4.6	4.6
5th mill	—	—	4.4	4.4	4.4	4.5
6th mill	—	—	—	—	4.3	4.4
7th mill	—	—	—	—	—	4.3

The table thus gives the correct height for the trashplate in operation, taken in the vertical axial plane of the top roller, by multiplying the above values by the fibre index q for any mill.

For example, if $q = 10 \text{ kg}/\text{m}^2$ (2 lb./sq.ft.), we should have, in a 15-roller tandem, the following heights in operation for the five trashplates: 80, 60, 48, 45, 44 mm (3.15, 2.36, 1.89, 1.77, 1.73 in.).

This gives only the mean values; some technicians adopt settings varying by 10 or 20% around the values given. Values for the mills when empty are easily established. It suffices to subtract the mean lift (*cf.* p. 216). It should be remembered that mechanics often measure the setting in relation to the point of the teeth of the mill rollers; thus it is necessary to subtract half of the height of the teeth.

To obtain the height of the trashplates when operating in inches per lb./sq.ft. of fibre index, multiply the numbers in Table 13.10 by 0.1922. For example, for a fibre index of 3.85 lb./sq.ft., the height of the trashplate in operation for the first mill of a 15-roller tandem would be: $8 \times 0.1922 \times 3.85 = 5.92$ in.

Considerations on height of trashplate

Some engineers have a tendency to set the trashplate in a rather high position, others prefer it low.

A trashplate set too high absorbs too great a proportion of the hydraulic pressure, at the expense of the extraction, increases the power consumption of the mill, and wears more rapidly. Further, the mill will choke at a lower fibre loading, and its capacity will therefore be diminished.

Too low a trashplate gives too high an angle of contact of the bagasse with the delivery roller (Fig. 13.10), rendering feeding at this point more difficult and in consequence increasing the risk of choking. Further, the bagasse tends to form a roll, since the top layer is gripped by the top roller while the lower layer drags on the trashplate. Hence chokes, chattering and poor pressure at the delivery. To prevent small pieces of bagasse from dropping through at P' , a closer setting must be adopted for the distance T between the heel of the plate and the delivery roller.

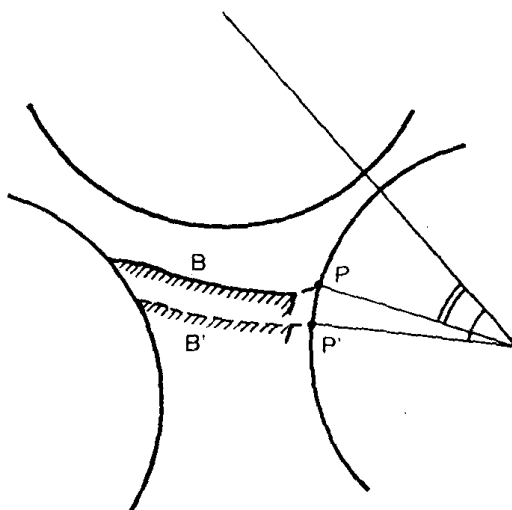


Fig. 13.10. Angle of contact with delivery roller.

In making a choice between these two groups of disadvantages, a too high trashplate would still be preferable to one too low.

Abnormal noises

When a mill "snores", it is generally due to the fact that the toe of the trashplate is set too low at the feed roller (angle β of Fig. 13.8 too great). It then forms at B (Fig. 13.9) a pocket from which the compressed bagasse escapes with difficulty, and so produces this characteristic vibration and snoring sound.

This is no cause for anxiety; but, contrary to a widespread idea, it does not in the least prove a good setting or a proper functioning of the mill.

When a mill "growls", and especially when it "chatters", it is generally due to a trashplate set too low, provoking the formation of a roll of bagasse on the trashplate,

Squirting

A mill is said to "s squirt" or "spit" when particles of wet bagasse are projected at intervals through its delivery opening. In most cases, this phenomenon is due to too close a setting between trashplate and delivery roller (dimension T too small). The juice, having no further

outlet, remains entrapped in the pockets formed by the bagasse, accentuating the phenomenon of re-absorption (*cf.* Fig. 10.13) to the point where the juice is expelled violently from the free side at the moment it is released from the minimal space formed by the delivery opening. The extraction therefore suffers badly.

A mill can squirt, even when the dimension T is sufficient. This then is caused by the combination of all factors causing re-absorption: speed, degree of compression, too great a feed opening, absence of juice grooves in the feed roller, etc.

Chokes

The term "choke" describes the condition when the mill will not grip all the feed which arrives, which therefore accumulates at the feed roller. Certain chokes may be due to the following phenomenon: the feed roller has worn its bearings on the feed side. When the pressure builds up, the bagasse forces the roller into the pocket so formed, the depth of which may reach 1 or 2 mm; the roller then separates from the trashplate; the bagasse remains caught on the toe of the plate, and constricts the passage.

Conversely, if the trashplate is adjusted close against the roller, the roller bears heavily against it, when the mill runs empty, and wears it rapidly.

The symptoms of this trouble then are:

- (a) Rapid wear of the trashplate
- (b) Slight displacement of the feed roller towards the feed side, at the moment the bagasse arrives at the mill.

Chokes are most often due to:

- (a) Either a bad setting of the mill, the feed opening too small or too large, the trashplate badly placed or poorly adjusted at the feed roller, the space at the heel of the plate too small
- (b) Teeth of the plate being lifted or bent towards the back
- (c) A roller too smooth, or too new, or polished
- (d) A roller of too small a diameter, too worn, or with teeth rounded
- (e) Excessive imbibition for the crushing rate required, or imbibition water too hot.

Displacement of flanges

When the trashplate is too low, it often happens that the pressure reacts on the side flanges of the top roller, and causes them to lift off, by pulling out the fixing screws.

Lifting of the flanges may therefore be a sign of too low a trashplate setting.

However, it can also be produced due to too great a difference in thickness of the bagasse blanket between the right- and left-hand sides, or due to passage of a foreign body on one side; or insufficient play between flanges and bottom rollers.

Wear and channelling of trashplate

The surface of the trashplate does not wear uniformly: it generally forms grooves corresponding, at the toe of the plate, to those of the feed roller, but diverging from the centre towards the extremities of the plate, in the direction of the bagasse movement (Fig. 13.11).

This divergence is due to the fact that the bagasse load at the mills is always inevitably heavier at the centre than at the ends of the rollers. When the compressed bagasse, held in straight channels by the grooves of the feed roller, is discharged onto the trashplate, it tends

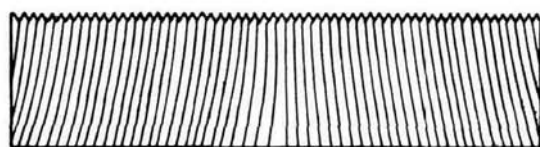


Fig. 13.11. Trashplate worn in grooves (plan view).

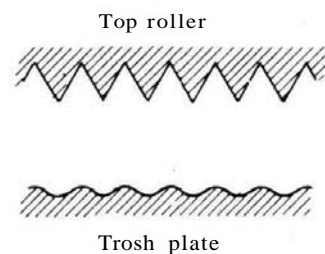


Fig. 13.12. Grooved trashplate.

to distribute itself uniformly, and to spread from the centre towards the ends of the roller.

The same cause provokes a more severe wear of the rollers at the centre than at the extremities: this is readily appreciated by placing a straightedge on the rollers at the end of the season. When the feed and top rollers are worn in this way, the bagasse blanket becomes all the thicker at the centre, and the effect on the trashplate is accentuated.

The Archimedean screw which returns the fine bagasse from the cush-cush screen to the intermediate carrier also returns more of this material to the middle of the conveyor than to the sides, and also contributes to loading the centre of the rollers.

Finally, there is friction on the sides of the carriers, retardation of the bagasse at the sides

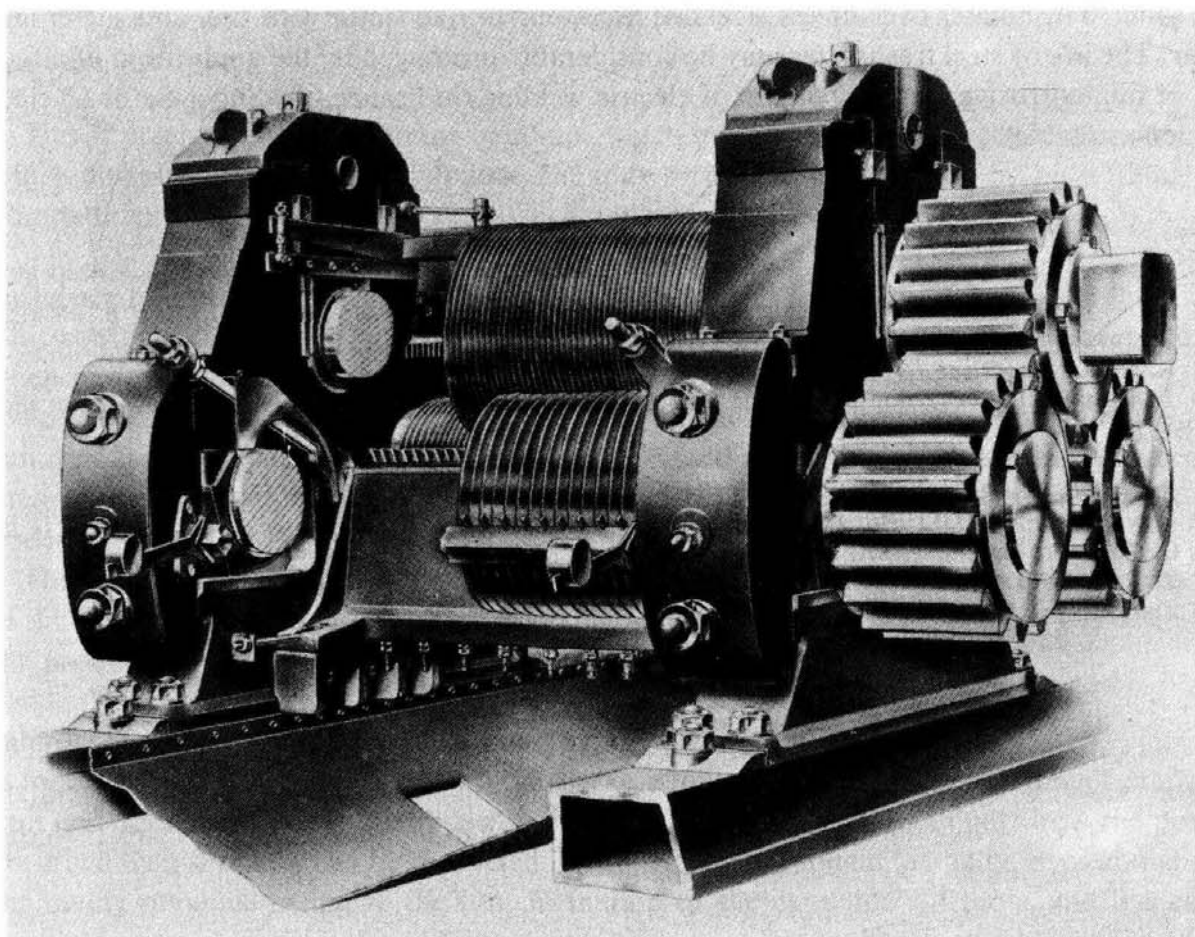


Fig. 13.13. View of mill showing method of fixing and adjusting trashplate. Note double pitch of feed roller.

in favour of that in the middle, and obstruction of the outer portions of the bagasse by the idler pulleys and their supporting runners, which again tends to concentrate the bagasse at the centre.

All these causes tend to diminish the layer of bagasse at the edges, producing what is called the "edge effect": the cane which arrives at the mills on the sides of the carrier is less effectively compressed; it sometimes leaves the first mill scarcely flattened, and still recognisable as stalks.

With the object of avoiding the bursting stress on the flanges which is caused by the diverging movement of the bagasse on the trash plate, certain manufacturers have supplied plates machined with shallow grooves corresponding to those of the feed roller, *i.e.* matching at a distance those of the top roller, and parallel with them (Fig. 13.12). These have the disadvantage of tending to maintain, at the delivery roller, the uneven distribution of the bagasse across the width of the feed roller, which is in part corrected by smooth trashplates. They may, however, be of interest in cases where the extent of the divergent grooves on the trashplate would be liable to lead to *difficulties* at the delivery roller.

Metal for trashplates

Trashplates are made in cast iron, in steel or in stainless steel. Their cost increases in the order indicated, but their life similarly, so that the cost per tonne of cane remains approximately the same. Trashplates of stainless steel acquire a mirror-like finish with use, and suffer less wear. The life of steel trashplates may be considerably prolonged by the application of a facing of non-corrosive metal by means of electric welding (in France, the Super-Safor 60 electrode is suitable).

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14. Power Requirements of Mills

FACTORS INFLUENCING POWER REQUIREMENTS

Work of crushing the cane

In crushing cane, the resistance presented by the pith is negligible compared with that offered by the woody parts, that is, the rind and the nodes; so that the resistance to crushing is analogous to that which would be offered by a hollow tube partitioned and reinforced by full discs distributed along its length (Fig. 14.1).



Fig. 14.1. Diagrammatic representation of resistance of cane to crushing.

This is precisely the structure of bamboo, which moreover is closely related to cane, and which possesses the same woody parts. The only difference is that it has no pulp filling the compartments thus formed (*cf.* p. 32).

Influence of state of cane

This conformation of the woody tissues of the cane determines its reactions under the influence of pressure. If we compare the manner in which cane and bagasse behave when subjected to a certain pressure P , we shall see that, for the same increment of pressure dP , the cane will be deformed by an amount dh greater than that by which the bagasse will be compressed. This is due to the elasticity of the cylinder of rind and the small partitions of the nodes.

The definition of work, and consequently of power, shows then that the power expended in the case of cane is superior to that expended in the case of bagasse. The work is of course the product of a force (here P) and a displacement (dh).

Interdependence of preparatory and milling equipment

It follows that the power exerted by a crusher or a mill will depend on the degree of preparation undergone by the material which it receives. This also explains in part how the power absorbed by a shredder can be recovered in the rest of the tandem (actually, over and above the saving obtained owing to the transformation of the cane into a fibrous and less compressible material, there is the additional possibility of reducing slightly the load on the mills, thanks to the ease of extraction of juice from this disintegrated mass).

Magnitude of power requirement

The figures for power which are given in technical books are generally stated as a function of the tonnage of fibre treated.

In steam-driven plants, these power figures are therefore expressed in indicated horse-power per tonne fibre per hour (i.h.p. per t.f.h.) or per tonne of cane per hour (i.h.p. per t.c.h.).

Deerr¹ gives the figures shown in Table 14.1. These values are for a mill of 864 x 1,981 mm (34 x 78 in.) crushing 45 t.c.h. with cane of 12.7% fibre, under hydraulic pressure of 356 tonnes at the mills.

Maxwell² gives for Java and Hawaii the figures shown in Table 14.2. This table demonstrates the desire in Hawaii to obtain the maximum from each mill, whichever it may be.

Tromp³ gives figures slightly different for Java (Table 14.3).

In Mexico⁴ one can reckon on 14- 18 i.h.p./t.f.h. for absorbed power and 22 i.h.p. for power to be provided.

TABLE 14.1

POWER REQUIREMENTS OF MILLS (DEERR)			
		<i>i.h.p./t.c.h.</i>	<i>i.h.p. /t.f.h.</i>
Carrier	} total	3.3-4.4	26-34
Crusher		av. 3.7	av. 29
1st mill			
2nd mill	} total	2.2-3	17-23
3rd mill		av. 2.5	av. 20
4th mill			

TABLE 14.2

POWER REQUIREMENTS OF MILLS (MAXWELL)		
		<i>i.h.p. /t.f.h.</i>
		<i>Java</i> <i>Hawaii</i>
Crusher		16- 18 15 approx.
Double crusher		30-38
1st mill if no crusher		30-38 30 -
1st mill preceded by crusher		26-30 30 -
Intermediate mill		15-26 30 -
Final mill		15-23 30 -

TABLE 14.3

POWER REQUIREMENTS OF JAVA MILLS (TROMP)		
		<i>i.h.p. /t.f.h.</i>
		<i>Extreme values</i> <i>Average</i>
Crusher		8-35 17
1st mill		15-51 28
2nd mill		14-38 24
3rd mill		11-31 19
4th mill		10-28 18
5th mill		9-23 16

Relation between fibre and power. The power figures which we have just quoted are interesting and useful, but the variations shown are considerable.

This emphasises that the tonnage of cane and even the tonnage of fibre are related only remotely to the power developed. We shall proceed to find the reason.

GENERAL FORMULA FOR POWER CONSUMPTION OF MILLS

The determination of the power consumed by a mill is rather complex because a number of factors enter into it.

To begin with, this power may be split up into 6 different principal terms:

- (1) Power consumed by compression of the bagasse
- (2) Power consumed in friction between the shafts and bearings of the rollers
- (3) Power consumed by friction between bagasse and trashplate
- (4) Power consumed by friction of scrapers and toe of the trashplate against the rollers, to which should be added the work of dislodging the bagasse at these points
- (5) Power consumed in driving the intermediate carriers
- (6) Power absorbed in the gearing.

Furthermore, these components of the power themselves depend on certain factors rather difficult to measure or estimate, such as: variety of cane (at equal weights of fibre, the crushing of two different varieties can exact appreciably different power requirements), state of the friction surfaces, quality and regularity of lubrication, adjustment of settings and of the trashplate, etc.

Owing to the impossibility of taking into account all these factors, there is no point in seeking great precision in the calculation of power for milling. One must be content with an order of magnitude; such a figure is all the more useful since many people have not even a rough idea of it. The values found in practice may differ appreciably from the mean power figures which we shall derive. This difference may be as much as 20 or even 25% of the normal value furnished by the formula.

Detailed calculations were given in the second edition (pp. 224 - 232) for the power absorbed by a mill. The following results were obtained.

1. Power absorbed by compression of bagasse

$$P_1 = 0.4 \cdot \frac{6r - 5}{\sqrt{r}(1 + \sqrt{r} - 1)} \cdot FnD \sqrt{\epsilon_A} \quad (14.1)$$

$$\left(P_1 = 0.12 \cdot \frac{6r - 5}{\sqrt{r}(1 + \sqrt{r} - 1)} \cdot FnD \sqrt{\epsilon_A} \right)$$

F = hydraulic load of the mill, in tonnes (long tons)

D = mean diameter of the rollers, in m (ft.).

For power in kW, replace the coefficient 0.4 by 0.3 (or 0.12 by 0.09).

Table 14.4 gives values of the term $\frac{6r - 5}{\sqrt{r}(1 + \sqrt{r} - 1)}$ for various values of r :

TABLE 14.4

VALUES OF r TERM IN EQN. (14.1)

r	1	1.1	1.2	1.3	1.4	1.5	1.6	1.8	2
$\frac{6r - 5}{\sqrt{r}(1 + \sqrt{r - 1})}$	1	1.159	1.388	1.587	1.760	1.913	2.049	2.282	2.475

Example. For $F = 340$ tonnes, $r = 1.3$, $n = 5$ r.p.m., $D = 813$ mm (32 in.), $\epsilon_A = 0.0164$, we have

$$P_1 = 0.4 \times 1.587 \times 340 \times 5 \times 0.813 \times 0.1281 = 111 \text{ h.p.}$$

2. Power absorbed in friction between journals and bearings

$$P_2 = 0.04 FnD \quad (P_2 = 0.012 FnD) \quad (14.2)$$

This is a mean value which may vary between 0.033 and 0.053 FnD (0.010 and 0.015 FnD).

3. Power absorbed in friction between bagasse and trash plate

$$P_3 = 0.035 FnD \quad (P_3 = 0.011 FnD) \quad (14.3)$$

A mean value which may vary between 0.016 and 0.054 FnD (0.005 and 0.016 FnD). These values are slightly lower than those given in the second edition. In fact, from the work of Murray⁵ and Cullen, Russell⁶ deduces that the coefficient of friction which we have reckoned as 0.035 is more often between 0.2 and 0.3. The calculation has thus been revised with the mean value 0.3 and extremes of 0.2 and 0.35.

4. Power absorbed by friction between scrapers and rollers

$$P_4 = 2 LnD \quad (P_4 = 0.18 LnD) \quad (14.4)$$

5. Power absorbed for intermediate carrier drive

$$P_5 = 2 LnD \quad (P_5 = 0.18 LnD) \quad (14.5)$$

The values for these five main factors are tabulated in Table 14.5.

We have taken care to give in this table the values which P_2 and P_3 assume for the extreme values of settings and of the coefficients of friction. This shows the importance of the uncertainty involved in certain terms, and demonstrates the differences which may exist between well-lubricated and well-set mills and those poorly maintained.

Moreover, the terms P_1 , P_4 and P_5 in turn are subject to variations of nearly 25% upwards or downwards, either as a matter of cane variety, or for reasons similar to those just given.

TABLE 14.5
POWER CONSUMPTION OF MILLS

	P_1	Minimum value	Mean value	Maximum value
Cor. presion of bagasse			$0.4 (N.12) \cdot \frac{6r - 5}{\sqrt{r(1 + \sqrt{1 - 1})}} \cdot Fr_{15} \sqrt{\epsilon_A}$	
Friction in bearings	P_2	$0.03 FrD$ (0.010 FrD)	$0.04 FrO$ (0.02 FrD)	$0.053 FrO$ (0.015 FrO)
Third plate	P_3	$0.016 FrO$ (0.005 FrD)	$0.04 FrO$ (0.01 FrD)	$0.054 FrD$ (0.016 FrO)
Scraper and roller of ashplate	P_4		$2 FrO$ (0.18 FrD)	
Intermediate carrier	P_5		$2 FrO$ (0.18 FrD)	
Total		$0.046 FrD$ (0.015 FrD)		$0.07 FrD$ (0.03 FrD)

L and O in metres (ft.)

Summing these terms and taking their mean value, we obtain the total power consumed by the mill proper (metric units):

$$P = nD \left\{ F \frac{f}{L} \left[0.4 \cdot \frac{6r - 5}{\sqrt{r} (1 + \sqrt{r - 1})} \cdot \sqrt{\epsilon_A} + 0.075 \right] + 4 L \right\} \quad (14.6)$$

6. Power absorbed in the gearing

We shall take this into account by dividing eqn. (14.6) by Q , the mechanical efficiency of the gearing, which gives the final equation:

$$P = \frac{nD}{e} \left\{ F \left[0.4 \cdot \frac{6r - 5}{\sqrt{r} (1 + \sqrt{r - 1})} \cdot \sqrt{\epsilon_A} + 0.075 \right] + 4 L \right\} \quad (14.7)$$

$$\left(P = \frac{nD}{e} \left\{ F \left[0.12 \cdot \frac{6r - 5}{\sqrt{r} (1 + \sqrt{r - 1})} \cdot \sqrt{\epsilon_A} + 0.023 \right] + 0.36 L \right\} \right)$$

Value of Q . Lehky⁷ gives the following values:

Efficiency of mill pinions = 0.95 X 0.95 = 0.90

Efficiency of each pair of gears = 0.98

Friction in various bearings = 0.95

This gives for the overall efficiency:

$$0.90 \times 0.98 \times 0.98 \times 0.95 = 0.82$$

However, Varona⁸ reports very substantial differences in the efficiencies of different types of gearing:

Double-helical gearing, fine pitch, double reduction, well lubricated: 0.96

Ordinary gearing, worn: 0.80

MacNee⁹ gives for a train of gearing:

(a) For 2 pairs of ordinary gears $Q = 0.90-0.95$

(b) For 2 pairs of helical gearing $e = 0.98$.

The following figures are commonly used for well-maintained gearing, allowing for friction of bearings:

TABLE 14.6

EFFICIENCY OF GEARING

Efficiency of ordinary pinions (effect on the total power taken by the prime mover)	0.85
Efficiency of "pinion housing" (cf. p. 167) (effect on the total power)	0.97
Efficiency of a pair of ordinary gears	0.95
Efficiency of a pair of helical gears	0.96
Efficiency of a double-reduction helical gearing	0.985
Efficiency of a triple-reduction helical gearing	0.98
Efficiency of turbine reduction	0.98

It follows that the ratio of power exerted on the bagasse to that developed by the turbine or motor has a value:

$$0.98 \times 0.985 \times 0.96 \times 0.85 = 0.78-0.79$$

Values furnished by eqn. (14.7). We have adopted in this equation values which we have described as averages. The application of this formula in practice shows that the values so adopted furnish good results for mills of standard type.

With old mills, with wide trashplate, sluggish hydraulic systems, and mediocre lubrication, the term 0.08 (0.024) in the brackets should be replaced by 0.10 (0.030). On the other hand, for modern mills, well maintained and with a narrow trashplate and automatic lubrication, this coefficient could be replaced by 0.06 (0.018).

1st Mill. The power required for breaking up the structure of the cane, even when prepared by knives, is substantially greater than that absorbed by compression of bagasse (*cf.* p. 209). For this reason we would recommend replacing for the 1st mill, the coefficient 0.4 (0.12) in the brackets in the expression for P_1 by:

$$\begin{aligned} \text{1st mill after knives:} & \quad 0.5 \quad (0.15) \\ \text{1st mill after knives and shredder:} & \quad 0.45 \quad (0.135) \end{aligned}$$

Power for crushers. Repeating the same calculations, we obtain for crushers the figures given in Table 14.7.

Taking into account the efficiency of the gearing, we have then:

$$P = \frac{nD}{e} \left\{ \left[0.50 \cdot \frac{6r - 5}{\sqrt{r}(1 + \sqrt{r} - 1)} \cdot \sqrt{\epsilon_A} + 0.04 \right] F + 1.00 L \right\} \quad (\text{metric units}) \quad (14.8)$$

For the equation in British units the numerical coefficients become 0.15, 0.012 and 0.09, respectively.

TABLE 14.7

POWER CONSUMPTION FOR CRUSHERS (METRIC UNITS)

Compression of cane	$P_1 = 0.50 \cdot \frac{6r - 5}{\sqrt{r}(1 + \sqrt{r} - 1)} \cdot FnD \sqrt{\epsilon_A}$
Friction in bearings	$P_2 = \frac{2}{1.9} \times 0.04 FnD = 0.04 FnD$
Trashplate	$P_3 = 0$
Scrapers	$P_4 = 2p \times 0.0126 LnD \approx LnD$
Intermediate carrier	$P_5 = 0$

General simplified formula

Much of the value of the complex equation (14.7) is lost due to the problematical nature of the coefficient 0.075 (0.023) corresponding to the components P_2 and P_3 of the power consumption. When the value of this coefficient cannot be chosen with sufficient precision and when the exact value of the efficiency of the gearing is not known, it is preferable to be content with the approximate formulae given by eqn. (14.9):

$$\begin{aligned} \text{Normal power:} & \quad P_N = k F n D \\ \text{Maximum (peak) power:} & \quad P_P = K F n D \end{aligned} \quad (14.9)$$

The formulae are applicable to all mills, and the coefficients k and K have the values given in Table 14.8 (F in tonnes, D in metres).

TABLE 14.8

NORMAL POWER AS A FUNCTION OF CANE PREPARATION: VALUE OF COEFFICIENT k						
Preparation plant	2 knives 50-mm pitch		2 knives 20-mm pitch		2 or 3 knives + shredder	
	(h.p.)	(kW)	(h.p.)	(kW)	(h.p.)	(kW)
1st mill	0.271	0.199	0.246	0.181	0.239	0.176
2nd mill	0.230	0.169	0.222	0.163	0.214	0.157
3rd mill	0.205	0.151	0.201	0.148	0.201	0.148
Later mills	0.197	0.145	0.197	0.145	0.197	0.145
MAXIMUM POWER AS A FUNCTION OF METHOD OF FEEDING: VALUE OF COEFFICIENT K						
	Manual control					
1st mill	0.330	0.243	0.310	0.228	0.284	0.209
2nd mill	0.260	0.191	0.250	0.184	0.245	0.180
3rd mill	0.219	0.163	0.220	0.162	0.220	0.162
Later mills	0.209	0.154	0.209	0.154	0.209	0.154
	Feeler plate					
1st mill	0.277	0.244	0.298	0.219	0.273	0.201
2nd mill	0.256	0.188	0.245	0.180	0.238	0.175
3rd mill	0.219	0.161	0.219	0.161	0.218	0.160
Later mills	0.209	0.154	0.209	0.154	0.209	0.154
	Donnelly chutes					
1st mill			0.280	0.206	0.257	0.189
2nd mill			0.238	0.175	0.228	0.168
3rd mill			0.216	0.159	0.215	0.158
Later mills			0.209	0.154	0.209	0.154

These equations assume gearing of high efficiency. They are useful for planning projects, to determine the power to be provided.

To get the power in British units (F in long tons, D in ft.), multiply the coefficients of Table 14.8 by 0.31.

Crusher. For a crusher we may take:

$$\begin{aligned} P_N &= 0.75 FnD & (P_N &= 0.0040 FnD) \\ P_P &= 0.20 FnD & (P_P &= 0.0050 FnD) \end{aligned} \quad (14.10)$$

GENERAL RELATIONSHIPS

Influence of crushing rate

Even from consideration of the more accurate formula (14.7) one fact is immediately obvious, that is, the small effect of the crushing rate on the power consumption.

If we take for example the case of a final mill of 813 X 1,676 mm (32 x 66 in.) crushing, other things being equal: (1) 100 t.c.h. and (2) 80 t.c.h., with the same fibre $f = 0.145$ in both cases, the same speed of 5 r.p.m. and the same t.h.p. of 340 tonnes, the re-absorption factor being 1.39 in the first case and 1.28 in the second, we obtain the following figures for power consumption:

$$\begin{aligned} \text{1st case (100 t.c.h.): } P &\approx 200 \text{ kW} \\ \text{2nd case (80 t.c.h.): } P &= 192 \text{ kW} \end{aligned}$$

For an increase of 25% in crushing rate, the power consumption has therefore increased by only 4 – 5%.

From the point of view of the steam consumption at the mills, it is therefore helpful to work with the thickest possible bagasse blanket.

On the contrary, if the increase in tonnage is obtained by an increase in speed of the mill, without alteration to the setting or the fibre loading, the power consumed will increase practically in proportion to the speed and tonnage.

Distribution of power among the three rollers

Measurements have been made on a first mill of 813 x 1,676 mm (32 x 66 in.), operating at 5.2 r.p.m., with a t.h.p. of 475 tonnes, and crushing 80 – 100 t.c.h., with an empty setting of 22/3 mm (0.87/0.12 in.) rising under full load to 34/20 mm (1.34/0.79 in.). For this mill (a Fives Lille – Cail C46 unit) the torque, or the power, was distributed as follows:

Top roller:	56%
Feed roller:	22%
Delivery roller:	22%

The proportions between the two bottom rollers might appear unexpected. It is due to the following facts:

(a) For a first mill, the work of crushing unbroken pieces of cane fibre is effected predominantly at the feed roller, and is much less at the delivery

(b) The work due to friction on the trashplate, which must be set closely against the feed roller, is greater than that due to friction of the scraper plate on the delivery roller.

Results are not available for later mills, but it is considered that the distribution should be of the order of:

Top roller:	55%
Feed roller:	10-20%
Delivery roller:	25-35%

Torque - load coefficient

Murry¹⁰ has commented that the expression:

$$N = \frac{G}{FD} \quad (14.11)$$

N = torque - load number

G = total torque for driving the two rollers considered

F = load applied to the top roller

D - mean diameter of the rollers

is a **dimensionless** coefficient, and characterises the power consumed in compressing cane or bagasse between two rollers.

In effect:

$$P = G\omega$$

P — power absorbed

ω = rotational speed of the rollers.

We have:

$$P = G\omega = \frac{2\pi n}{60} \cdot G$$

Hence:

$$N = \frac{60 P}{2\pi n FD} = 9.55 \frac{P}{FnD} \quad (14.12)$$

This agrees well with our formulae of Table 14.8. Murry moreover was interested only in the power absorbed by compression (our power P_1), and he and Munro suggested:

$$N = \lambda \sqrt{\epsilon_A} C_0^{0.75} \quad (\text{cane; Murry}) \quad (14.13)$$

$$N = X \sqrt{\epsilon_A} C_0^{0.21} \quad (\text{bagasse; Munro}) \quad (14.14)$$

which are similar to our equation:

$$N = \lambda \sqrt{\epsilon_A} f(r) \quad (14.15)$$

which is another way of writing our equation (14.1).

Power of a mill tandem

We have just been studying the power consumption of an individual mill. How does the total power consumption of the tandem vary?

Obviously, it is equal to the sum of the consumptions of the units of which it is composed. But it does not follow that the power consumed by one of these units will be the same, whether it belongs to a tandem of 11 or one of 17 rollers.

Actually, the power consumed by a mill depends on the hydraulic pressure chosen, on the speed at which it is operated and the corresponding fibre loading. Now, as we have seen, particularly for Java (*cf.* pp. 173 and 182), certain technicians modify these elements according to the position of the mill in the tandem.

Two principal schools of thought were recognised:

(1) The Javan school, where a lower hydraulic pressure was used with mills forming part of a longer tandem. The object of this school was to avoid consuming too much power in long tandems, which, even for the same total power, still give a valuable gain in extraction. This is the solution for countries where the price of sugar is low.

(2) The Hawaiian school, in which the maximum work is obtained from each mill, whatever its place in the tandem. This is the solution for countries where the price of sugar is high.

TABLE 14.9

TOTAL POWER CONSUMPTION OF A TANDEM (IN i.h.p.
PER t.f.h.)

<i>Tandem</i>	<i>Java</i>	<i>Hawaii</i>
11 rollers	105	112
12 rollers	107	132
14 rollers	101	148
15 rollers	110	165
17 rollers	110	181

Maxwell¹¹ gives the figures reproduced in Table 14.9, for the total power consumed in each of these two cases. The figures for Java are taken from figures published by that country; those of Hawaii correspond to the values generally reported in the Islands.

It is of interest to note that, in Java, it is the 14-roller tandems which have the lowest power consumption, when it is related to the tonnage of fibre treated.

Maxwell then compares¹² the tandems of 14 rollers 837 x 1,981 mm (34 x 78 in.) for the two countries, and emphasises, in this connection, the differences in the manner in which the cane is prepared and in which the mills are operated in the two countries:

(a) Composition:

Java: 19 tandems of 14 rollers, 837 x 1,981 mm (34 X 78 in.)

Hawaii: 6 tandems of 14 rollers, 837 x 1,981 mm (34 x 78 in.), preceded by a set of knives and a shredder.

(b) Conditions of operation:

TABLE 14.10

COMPARISON BETWEEN 14-ROLLER TANDEM IN JAVA AND HAWAII

	Capacity (t.f.h.)	Power (i.h.p./t.f.h.)	Total (t.h.p. tonnes)	Av. speed		Imb. % fib.	Extn.
				Crushers	Milts		
Java	8.3	101	1,630	3.15	1.85	150	94.7
Hawaii	7.7	173	1,820	4.13	2.84	292	97.9

The 173 i.h.p./t.f.h. for Hawaii includes prime movers for the knives and shredder.

While these are very old figures, they throw into relief the reasons for the high extractions obtained in Hawaii at that period:

- (1) Low crushing rates
- (2) Application of high pressures
- (3) Use of high speeds, though much below the limiting speeds possible
- (4) High expenditure of power (due to the 3 preceding factors)
- (5) High imbibition.

Parr's formula. Parr¹³ proposes the following formula for power:

$$P = 0.3 fN^{0.45}$$

P = power absorbed by a tandem, in i.h.p./t.c.h.

f = fibre in cane, %

N = number of rollers in the tandem.

This formula has the disadvantage that it is based on the tonnage treated, whereas we have seen that tonnage actually has only a small influence on the power consumed. However, it demonstrates well the economy in power per mill with a long tandem. Taking 100 as the power necessary for a tandem of 5 mills according to Parr's formula, the total power and power per mill of different tandems would be:

Tandem	12 rollers	75 rollers	18 rollers	21 rollers
Total power	90.5	100	108.6	116.6
Power per mill	113	100	90.5	83.3

Power as a function of fibre rate

Although the figures for power related to weight of fibre are not of much significance, we shall indicate the most frequent values, since they are of interest from the point of view of the practical efficiency of usage of power by the mills.

This power is expressed in i.h.p./t.f.h. and per mill (i.h.p./t.f.h./m). It is obtained by a simple rule of three, that is, by multiplying by $3/N$ the total power expended in the tandem per t.f.h. (N = number of rollers (cf. p. 56)).

In a short tandem, the figures approach the Hawaiian values: 25 – 30 i.h.p./t.f.h./m. Sometimes they reach 32 and even 35.

In tandems of average length, of 14 or 15 rollers, the values generally lie in the range 20-25 i.h.p./t.f.h./m.

In long and very long tandems, of 17 – 25 rollers, the values fall to 18 and 15. Central Tarlac, in the Philippines, with 25 rollers (2 crushers and 7 mills of 914 x 2,133 mm (36 x 84 in.)), reports 12.4 i.h.p./t.f.h./m.

This is the lowest value that has come to our notice. In South Africa¹⁴, we have approximately:

24 i.h.p./t.f.h. per mill for 15- and 18-roller tandems
20 i.h.p./t.f.h. per mill for 21-roller tandems.

Influence of variety

We have commented above (p. 229) that, for the same weight of fibre treated, the power may vary with the degree of resistance of the particular variety being handled.

One factory, in Australia, handling two different varieties of cane, EK 28 and Badila, has stated that its tandem consumed:

With EK 28: 69 i.h.p./t.f.h.
With Badila: 52.3 i.h.p./t.f.h.

or one-third more for the former variety than for the latter¹⁵.

Power to be installed

Knowing the mean power which should be required by a mill for the maximum values of the hydraulic pressure and the fibre loading, it will be sufficient to increase it by 15% to obtain the power of the prime mover to be installed.

In the case of combined drive, that is, if the prime mover under consideration drives two or more mills, it would be sufficient to fix the nominal power at 10% above the mean maximum power calculated.

Total power for the milling station

For total power consumed by the milling station (unloading equipment excluded), a South African report¹⁶ gives an average figure of 181.3 kW (243 h.p.) per t.f.h., including numerous diffusion factories, with one recording a figure of 255.5 (343), another 63.2 kW (84.7 h.p.) per t.f.h. with diffusion of bagasse. A good normal power for extraction by diffusion would be 125 (165); for factories with a tandem of mills, 188 kW (252 h.p.) per t.f.h., of which 82.3 (110) would be for cane preparation and 105.7 (142) for the milling tandem.

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15. *Electric Drive of Mills*

Advantages

Electrically driven mills are the exception. However, they offer several advantages:

(a) **Neatness and cleanliness.** An electric mill house is much neater than one where the mills are driven by steam engines, and even somewhat neater than one with turbine drive: there are no steam joints leaking or dripping, no oil splashing from the crank oiler or dripping from the lubricators; and no cumbersome steam pipes (Figs. 15.1 and 15.2).

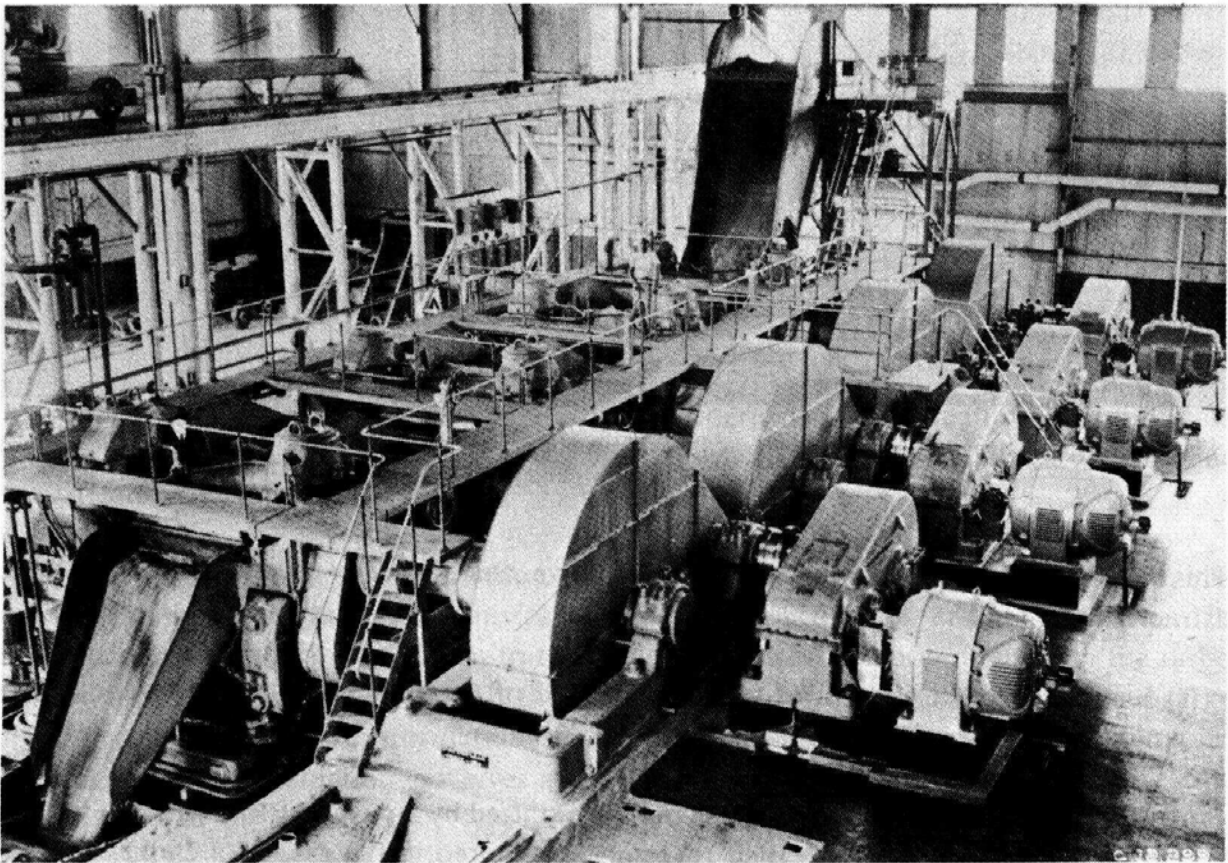


Fig. 15.1. Mill tandem with electric drive.

(b) **More complete and definite control.** Each mill is driven by a separate motor, since electric drive lends itself much better to individual drive than the steam engine and at least as well as the steam turbine, without increasing the requirements of floor space or manpower. The power consumed by each mill is ascertained at any moment by the simple reading of an ammeter; thus information is immediately available on the result of an alteration of setting.

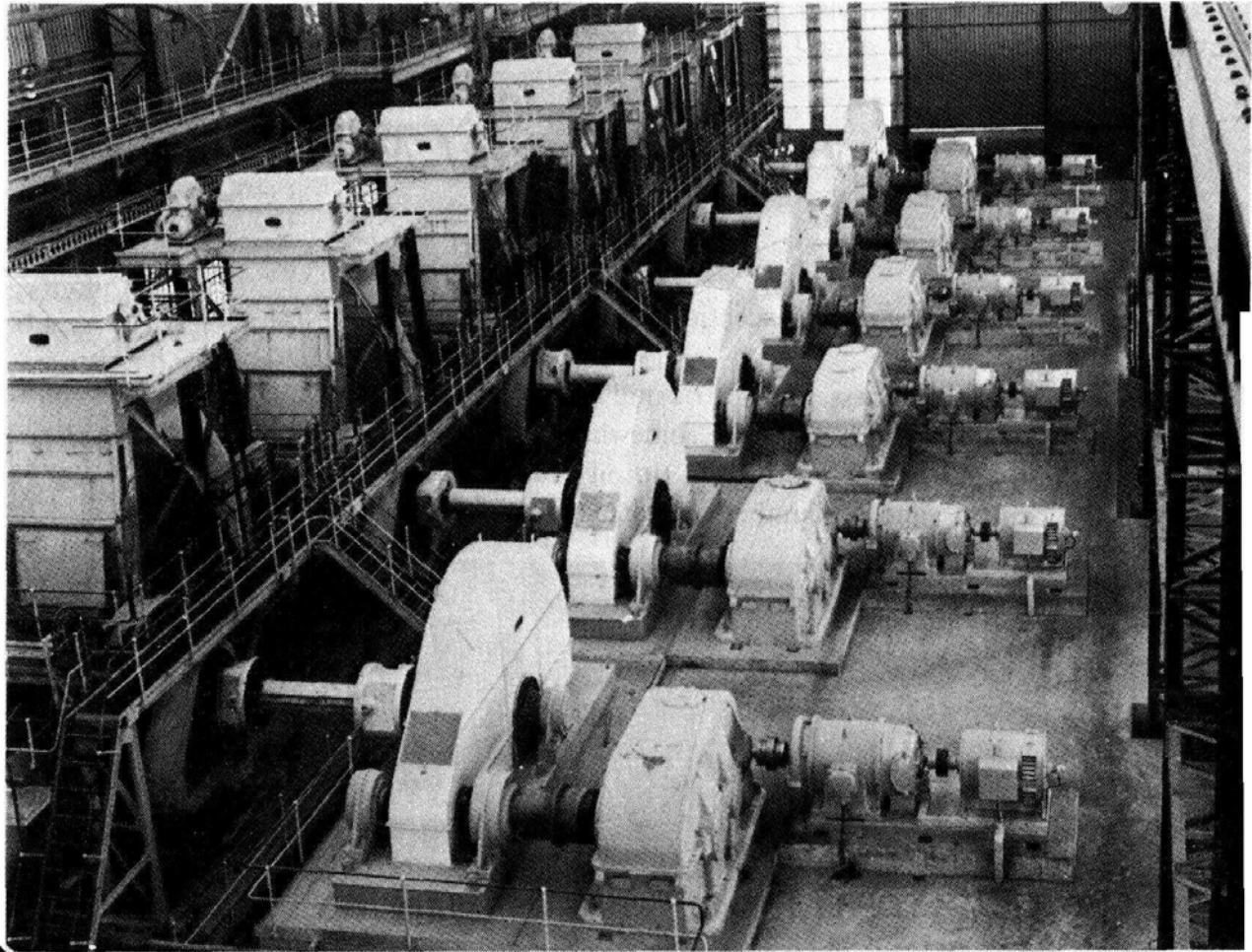


Fig. 15.2. Mill tandem with drive by Siemens "cascade" motors: constant-ratio mills (Fives Cail - Babcock) with Donnelly chutes.

This is an important point in favour of electric drive, and is quickly translated into improved extraction.

(c) **Ready general regulation of speed.** The speed of the whole mill tandem is controlled, very conveniently, from the power house.

(d) **Ease of starting and stopping.** The mills are started by a push-button control. Stopping the mills is equally simple, also their reversal, as required in the case of a choke. There is no attendant exerting all his strength to turn the control wheel of the valve gear, or "swinging" the engine to pass the dead point.

(e) **Lower operating and maintenance costs.** Costs of lubrication of electric motors are much lower than those involved for steam engines. In the same way, costs of maintenance are much lower: no piston rings to be changed, ovalisation to be corrected, slack to be taken up, valves to be ground, stop-valves to be replaced, or delicate adjustments to be made. A replacement armature costs less than a spare cylinder.

(f) **Accidents fewer.** No fear of water-hammer, or of fracture of a crank-pin. The spare armature ordered by the author in 1949 with the tandem for La Mare was still there in 1970 and had never been used.

(g) **Safeguard against passage of large pieces of tramp iron.** When a large piece of iron reaches the mill, the electric motor stops immediately; with a steam engine the inertia of the flywheel compels the foreign body to pass, at the expense of the roller grooving.

(h) **Exhaust steam free of oil.** The exhaust steam from the turbo set is not contaminated: the tubes of the first effect, the juice heaters working on exhaust, and pans remain clean on the outside. In particular, exhaust steam may be used in the distillery, which is much more economical than reduced live steam: the least contamination with oil is prohibitive for manufacture of rum by distillation with direct steam (if the difficulty is overcome by heating the still by means of a pre-evaporator, it becomes very expensive).

Disadvantages

On the other hand, electric drive presents certain disadvantages:

(a) **Higher first cost of installation.** The combination: high-pressure boiler, turbo-alternator, switchboards, motors and cables, and supplementary stage of gearing, costs more than the combination: low-pressure boiler, steam pipes and steam engines; and even more than the total of high-pressure boiler, high- and low-pressure piping, steam turbines, and supplementary reduction gearing.

If a factory has low-pressure steam system, the change from steam engines to electric drive involves either a huge immediate expenditure, or a progressive plan spread over several years and resolutely followed, so as to allow for full amortisation of the equipment by the time electrification is completed.

(b) **Additional double transformation of energy.** With a steam-engine installation, the steam produced at the boilers is transformed into mechanical work in the mill engines. In an electrical installation, in addition to the analogous transformation effected in the turbo set, the energy must also undergo: (1) the transformation of movement into electric energy in the alternator; (2) the transformation of electric energy into movement in the mill motors. Each of these transformations involves a loss of efficiency. We shall see later (p. 244) the final balance of the two systems.

(c) **Extra stage of reduction gearing.** The speed of electric motors necessitates the interposition of an extra stage of speed reduction between motor and mill, taking up additional space and involving further loss in power.

(d) **Less complete speed control.** While the regulation of mill speeds is more immediate, it is conveniently effected only on the general speeds of the tandem; individual speed regulation is generally obtained only with a sacrifice in efficiency, unless one of the more complex systems indicated later (p. 246) is adopted; these are more expensive.

With this same reservation, turbines and particularly steam engines maintain their power better at low speeds, and are more flexible. The speed range obtainable with a steam engine is much greater and more complete.

Herein lies the greatest disadvantage of electric drive by an ordinary induction motor.

(e) **Accidents more serious.** While they are less frequent, the accidents liable to occur with electric drive are more serious, and necessitate a judicious provision of spare parts.

(f) **More specialised personnel.** The electric installation requires more specialised personnel.

SYSTEMS OF ELECTRIC DRIVE FOR MILLS

There are four main systems for electric drive of mills:

- (1) A.c. supply with variable frequency
- (2) Ward – Leonard d.c. system
- (3) A.c. – d.c. cascade connection
- (4) Direct current obtained by rectification of a.c. by means of thyristors.

Before describing these systems, we recall that the mill is a machine functioning at constant torque. The characteristics which we describe will be interpreted as applying to conditions of constant torque.

On the other hand, the principal consideration in our comparisons will be the manner in which we can obtain the necessary variations in speed. A mill should be able to function between 100% and 50% of its maximum speed. It is not practical, and sometimes not possible, to operate below 50% of maximum speed. It will thus be necessary that the constant torque can be obtained at 50% of the maximum speed.

Finally, the power demand of a mill is relatively stable, but it must nevertheless provide for variations of demand arising from momentary overloads. It is therefore necessary to provide the motors with a certain reserve of power, which we shall fix at 10% as a minimum.

We shall now consider how these requirements are realised.

1. Variable-frequency a.c. system

This is the simplest method.

General regulation. General regulation of the tandem is obtained by altering the frequency of the supply. When this system is adopted, a separate turbo-alternator must be provided for the mills, since the changes in frequency provided for the mills cannot be inflicted on the other motors in the factory.

The solution generally adopted consists then of providing 3 turbo-alternators, which are often of similar power:

One, of variable frequency, for the mills

Another, of fixed frequency, for the remainder of the factory

A third unit, of variable frequency, which can serve as a spare for either of the others.

This **regulation** of frequency is obtained by means of a special governor controlling the speed of the turbine.

The frequency generally adopted by European firms is 50 cycles. The range of speed variation often is $\pm 20\%$, *i.e.* from 40 to 60 cycles. The standard American frequency is 60 cycles, and American firms generally *provide a regulation of +10 to -25%*, or from 45 to 66 cycles.

If one turbine is provided for use for either variable-frequency or fixed-frequency supply, it should be provided with the margin of safety necessary for either case.

As the speed is varied, the excitation and the load remaining constant, the frequency and the voltage vary proportionally with the speed.

The exciter therefore should furnish a constant current over the range of speed variation adopted. It should be capable of furnishing this current at the voltage corresponding to the lowest speed, and the form of the saturation curve should allow **approximately** the same current to be maintained at other speeds.

The voltage regulator similarly must be of special design, with a circuit using reactance rather than resistance, in such a way that the current will similarly remain constant when the voltage remains proportional to the frequency.

Individual regulation. The general speed regulation serves to adjust the speed and consequently the capacity of the tandem to the tonnage required. The speeds of all the mills thus increase or decrease in the same proportion; but we are often concerned with one mill which *takes the* feed (cane or bagasse) less effectively than the others, and chokes more *readily*. For this case an **individual** regulation of each motor is required. It is generally considered sufficient to provide a regulation of 10% below normal speed. This is obtained by means of a variable resistance, which thus involves a drop in efficiency due to the energy dissipated as heat in the rheostat. The speed can thus be varied for each motor between 90 and 100% of its normal speed, relative to that of the others. For the crusher, 15% is sometimes provided, *i.e.* a range from 85 to 100% of maximum speed.

The use of such individual speed control involves inconvenience and loss, due to the drop in efficiency, resulting from the production of heat, and above all the necessity to reduce the speed, not of the faulty mill, which must on the contrary be left at full speed, but of all the others, or at least those preceding it; these are reduced to 90% of normal speed. Hence the use of such individual speed control is avoided as far as possible. For this reason, a tandem of mills driven by this system nearly always functions as a combined-drive tandem, that is, with the relative speeds of the several mills rigidly fixed. This is a great disadvantage.

Voltages. The commonest voltages are 220, 380, 440 and 500 or 550. That of 380 permits **the** use of 220/380-V motors. This, and that of 440 V, are the most useful; they offer the advantage that at the usual power factors, the current read on the ammeter corresponds roughly to the horsepower. Since 1970, the trend is towards voltages of 5,500 and 11,000.

Speeds. The motors may be of 750 r.p.m., giving 730 r.p.m. under load, or of 500 r.p.m., giving about 485 r.p.m.

To reduce from these speeds to those of the mills, 3 stages of gearing are necessary; the

first reduction is generally made by double-helical gearing of high efficiency, running in an oil bath. The present tendency is to replace one at least, or even 2 stages of gearing, by an enclosed speed-reduction unit. Triple-reduction units are also made which replace any large gearing.

Types of motor. The motors employed for mill drives are generally wound-rotor induction motors with slip-rings; these have a good starting torque (Fig. 15.1). Their direction of rotation may be reversed simply by interchanging two phases.

2. Ward – Leonard d.c. system

This system was particularly widespread in Java. It consists of a squirrel-cage motor (*cf.* p. 1090) driving at constant speed a d.c. generator which supplies current to a motor with constant excitation. The voltage produced by the generator may be varied from a certain value in one direction to the same value in the opposite direction, passing through zero. As the excitation of the final motor is fixed, its speed varies with this voltage; thus all speeds from a maximum in one direction to a maximum in the opposite direction may be obtained, passing through zero. The torque varies little with speed. The efficiency of the assembly is of the order of 0.79.

This system comprised 3 machines for one drive, and was therefore less dependable and more expensive than the preceding systems of one motor only. Moreover, its efficiency was only moderate. It has been abandoned in favour of the d.c. system with rectification by thyristors (see below).

3. A.c. - d.c. cascade connection

This system, equally designated "Kraemer cascade", consists of a wound-rotor induction motor direct-coupled to a d.c. motor (Fig. 15.2). Its rotor, through the brushes and a silicon rectifier, supplies the d.c. motor. In this way if, for example, the slip amounts to 20%, in other words, if its speed is 80% of synchronous speed, the motor supplies to the shaft 80% of its nominal power, and consequently the rotor furnishes the difference to the d.c. motor, or 20% of the nominal power, which the d.c. motor in its turn supplies to the shaft. The total power supplied to the shaft is then constant and equal to 100% of the nominal power.

Reversal of rotation is obtained by changing the connections of the motor and the d.c. motor, or by a reversing gear.

For a speed range of 50 – 100%, the nominal power of the d.c. motor should be the same as that of the a.c. motor. For a reduced speed range, the power of the d.c. motor could be reduced proportionally.

This is the system applied by Siemens at the factory of Jaagbaan, in South Africa, and devised by Hughes¹. He points out that an electric motor can carry an overload of 75% for 15 seconds, and 25% for 2 hours. For this reason, it will be sufficient to use a cascade assembly of 110% of the power desired.

The system operates as follows. The wound-rotor induction motor is for example of 650 kW at 6600 V, running at 1475 r.p.m. The d.c. motor on the same shaft supplies 160 kW at 425 V when it runs at 1030 r.p.m., the speed to which it is expected the system will drop: the mill rotates at 4.2 r.p.m. when the drive shaft runs at 1475 r.p.m., and at 3.1 when the

shaft runs at 1030 r.p.m. At full speed (1475 r.p.m.), the induction motor develops all its power of 650 kW. The d.c. motor is then practically running at no load, the slip of the former being low. If we assume that the speed descends progressively below the maximum speed, the induction motor furnishes a lower and lower power output, falling to 490 kW when the speed has dropped to 75%, although the d.c. motor at that moment furnishes the 25% of power lost by the induction motor (160 kW) in such a way that the total output of the system thus remains constant at 650 kW.

The torque C of the motor increases as its speed drops, since $C = P/n$, P being the power and n the speed in r.p.m. We recover thus the advantage of the steam turbine, which allows coping with momentary loads, as the Kraemer cascade system does here, since the torque of the motor increases at that moment. In effect, when the speed of an induction motor decreases, its torque first increases (*cf.* Fig. 43.16), passes through a maximum, then drops. It is the remainder of the current, going to the d.c. motor, which compensates this drop, prevents the system from stalling, and enables it to cope with these overloads.

The efficiency varies from 0.86 to 0.90 when the speed varies between 50 and 100%; the system is thus very efficient.

The cost is of the same order as for a Ward – Leonard unit up to about 500 kW, and becomes more and more attractive per kW installed above that figure. It represents between 2 and $2\frac{1}{2}$ times the cost of an induction motor for our system (1).

4. D.c. system with rectification by thyristors

This is the latest development which seems likely to replace all others, even including the steam turbine. The a.c. supply, for example at 5500 V, is stepped down to 500 V, rectified by thyristors and sent to a d.c. motor which drives the mill through a reduction gearing or a train of gearing. This motor is of shunt or compound characteristics. For a given excitation, its speed is practically constant and independent of load.

The speed may be varied practically from zero to full speed, and at least from 40 or 50% to 100% of maximum speed. The efficiency is that of a d.c. motor, approximately 95%; the thyristors, heating up and requiring cooling, operate at an efficiency of 92 — 95%. The overall efficiency can thus be reckoned as 90%, and this efficiency is the same whether running at 50% or 100% of maximum speed. The power consumption is approximately 2.7 kWh per tonne cane and per mill, say 18 — 20 kW (24 — 27 h.p.) per t.f.h. and per mill; the power factor is about 0.65 and can be raised to 0.80 by the use of condensers. The motor can carry an overload of 20% for 20% of the time.

The first factory to instal this system was that of Borotou-Koro, on the Ivory Coast, in 1979. The cost of the system is not high, since there is only one motor in place of three for the Ward – Leonard system, and two for the Kraemer cascade system. Power consumption of the mill registers immediately on an ammeter or wattmeter, which is not available with a steam turbine; the speed may be regulated at will without affecting the power or the efficiency. Steam consumption depends only on the efficiency of the turbo-alternator of the factory, and that of the motor and the thyristors, and compares favourably with that of a two-stage turbo reduction gear. Cost of the installation and upkeep is not higher than for a steam turbine. To sum up, this is the best system available at present for mill drive.

Steam consumption

At the efficiency corresponding to the system adopted, steam consumption for these methods of electric drive is proportional to the power developed. This is an advantage over turbine drive, for which the steam consumption per kWh increases as the speed is reduced.

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16. Mill Drive by Steam Turbine

This method of drive was tried for the first time in 1947 in Louisiana. It proved successful and expanded very rapidly; it is the system most in use at the present day. It offers some important advantages:

(a) It avoids the double transformation and the transmission of energy required by electric drive. Even allowing for the slightly lower efficiency of the individual mill turbine relative to that of the central turbine provided for the whole tandem, a saving of 15–20% of the energy required for the mills is realised.

(b) It permits of a large range of speed variations for the individual mills, without serious

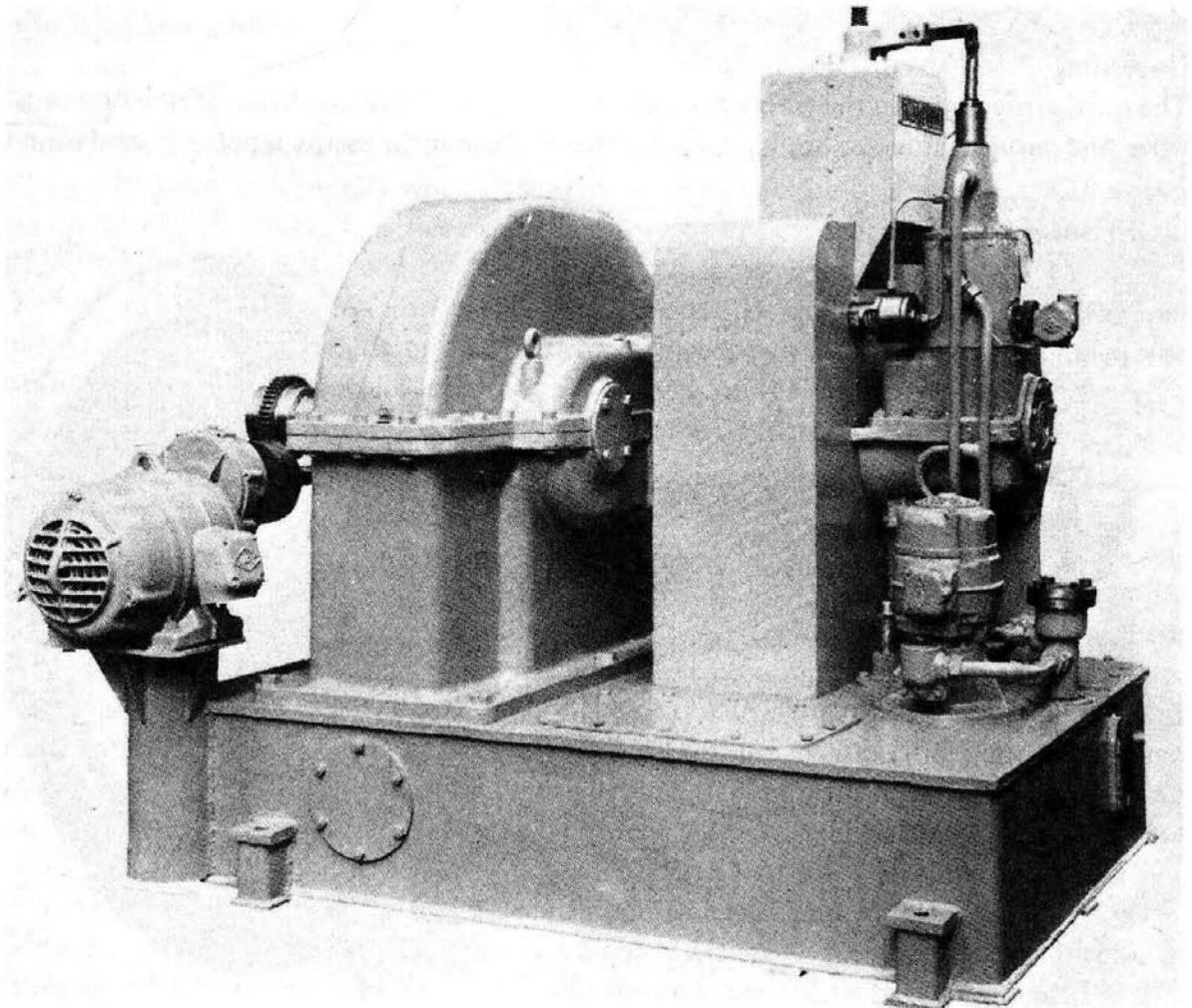


Fig. 16.1 Turbine and reduction gear for mill drive (Fives Cail–Babcock).

detriment to efficiency. This is a very big advantage compared with ordinary induction motors.

(c) Comparing it with the electrified tandem, although the individual turbines are more expensive than the electric motors, it dispenses with the turbo-alternator, the switchboard in the power house and the electric cables, so that the complete installation is less expensive.

(d) It retains the same advantages over reciprocating engines as electric drive: good starting torque, exhaust steam not contaminated with oil, less space taken up, reduction of manpower for supervision and maintenance, etc.

TURBINES FOR MILL DRIVE

On account of the high price of steam turbines, and the steam economy inherent in the system, simple turbines are generally adopted, for example, those with double wheel (Fig. 16.2). Their efficiency is not as good as multi-stage types, that is, their steam consumption is higher; but the total steam consumption of the whole installation remains very favourable. A typical installation is shown in Fig. 16.3.

Governing

The most critical unit in the turbine-driven mill is certainly the governor; it must be very sensitive and very rapid in action; it must not "hunt", and must assure a perfect regulation over

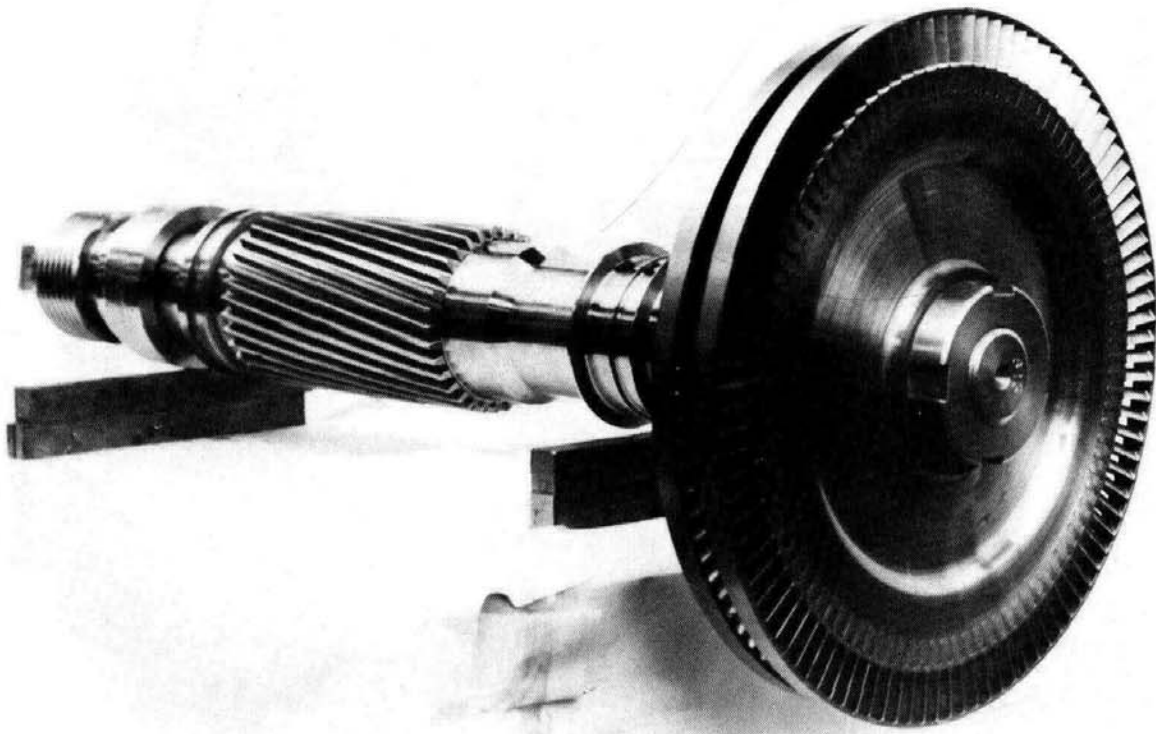


Fig. 16.2. Double wheel of steam turbine for mill drive.

the whole range of speed variation obtainable. This range of speed should go from 33 or 40 to 100, which variations are readily obtainable. A range of 50—100 is insufficient having regard to the possibilities of the steam turbine.

It is always at the first mill that the difficulties of regulation are by far the most serious. They are already reduced at the second mill, and are negligible in the following mills. Excellent governors are available, even for first mills; but they must be of very first-class type.

Power

The steam turbine is a very flexible machine from the point of view of power. Its power output may be varied over a very wide range. It suffices to adjust the steam supply by varying the number of nozzles between the nozzle chest and the turbine wheel. For this purpose, the manufacturers provide mill turbines with supplementary nozzles, which can be opened when the power output of the turbine is on the point of becoming insufficient. In this way it may always be arranged that the turbine functions near its full load, so that the steam consumption per kWh remains close to the minimum.

Since the turbine should furnish a peak power corresponding to the maximum demand of the mill which it drives, but has to function at a mean power output appreciably lower, it should be designed with a flat steam-consumption curve, and with maximum efficiency at two-thirds of full load.

Speed

The steam turbine is, however, sensitive to variations in speed in that the consumption of steam per kWh increases rapidly when it departs, in one direction or the other, from its optimum speed (*cf.* p. 1054). It is necessary therefore to ensure that it works in the neighbourhood of this optimum speed.

We may comment also that, for a given fixed speed, for example, 5 r.p.m. at the mill rollers, the specific steam consumption of an engine increases with the power developed (for example, as the hydraulic pressure is increased), whereas, in the same conditions, the specific consumption of a steam turbine decreases as the power is increased.

Types of turbine

The actual installations made have oscillated between two alternatives: either a more expensive turbine, of high efficiency (for example, one with 4 pressure stages), or a simpler turbine, of somewhat lower efficiency (for example, a single-wheel turbine). The choice between these two alternatives will obviously depend on circumstances, and must be studied for each case. However, in most of the cases which have come to our knowledge, it has been advisable to avoid the extreme solutions, *i.e.* both the single-wheel turbine, as too crude, and the 4-stage turbine, as too expensive for individual mill drive; the best compromise lay in a double-wheel turbine, with two velocity stages. Working between 22.5 kg/cm² (320 p.s.i.) and 350°C (660°F) admission, and 1.5 kg/cm² (20 p.s.i.) exhaust, a two-stage turbine of good design should have a steam consumption of between 10 and 15 kg (22 and 33 lb.) steam per h.p./h, according to its speed and load. The thermodynamic efficiency of a double-wheel turbine is of the order of 0.65; the overall efficiency of turbine and speed reducer, approximately 0.56.

The following figures show the difference in steam consumption between a double-wheel

turbine (FLC-38Y) and a four-wheel machine (FLC 52 CP) for admission steam at 22 kg/cm^2 (315 p.s.i.)- 345°C (650°F) and exhaust at 1.5 kg/cm^2 (20 p.s.i.):

Speed, r.p.m.		1,200		600
Power, h.p.		1,020		510
38Y	{	Steam consumption, kg/h (lb./h)	10,150 (22,380)	7,850 (17,300)
		Specific consumption, kg/h.p./h (lb./h.p./h)	10.0 (22.4)	15.4 (34.6)
52CP	{	Steam consumption, kg/h (lb./h)	9,150 (20,175)	7,300 (16,100)
		Specific consumption, kg/h.p./h (lb./h.p./h)	9.0 (20.2)	14.3 (32.2)

Figure 16.3 shows a turbine-driven tandem.



Fig. 16.3. Mill tandem driven by steam turbines.

17. Mill Gearing

SPEED REDUCTION

Gearing is required to reduce the speed of the prime movers:

Ordinary steam engines:	40- 75 r.p.m.
High-speed engines:	100 – 150 r.p.m.
Electric motors:	485 – 730 r.p.m.
Turbine reducers:	250 – 750 r.p.m.

to that of the rollers, 3 – 6 r.p.m.

This reduction is obtained by trains of reduction gearing:

2 stages for ordinary engines
3 stages for the other cases

One stage of ordinary spur gearing can effect a speed reduction of 7 or 8 to 1, but ratios greater than about 5 to 1 are not generally employed.

Two stages of gearing can thus give a reduction of $5 \times 5 = 25$, and 3 stages a ratio of $5 \times 5 \times 5 = 125$.

Types of gearing

Such trains of gearing were generally composed of ordinary spur gears, with machine-cut teeth (Fig. 17.1). Double-helical gearing (Fig. 17.2), while much more expensive, has a higher efficiency. It is employed mainly in high-speed gearing, and for high ratios of reduction. It readily permits of a reduction ratio of 10 to 1.

Ordinary spur gearing (Fig. 17.1) is encountered less and less. When it is employed, it is now kept for the last stage before the mills, the slowest. At present (1982), helicoidal gears (Fig. 17.3) are generally used.

Construction of gearing. Formerly, gears were of cast iron. Today, they are always made of cast steel, with machine-cut teeth.

Wear. When the gear wheels are worn, particularly the roller pinions, which have a limited life, their working life may be doubled by reversing them end for end on their shafts, so that the unused face of the teeth then does the work.

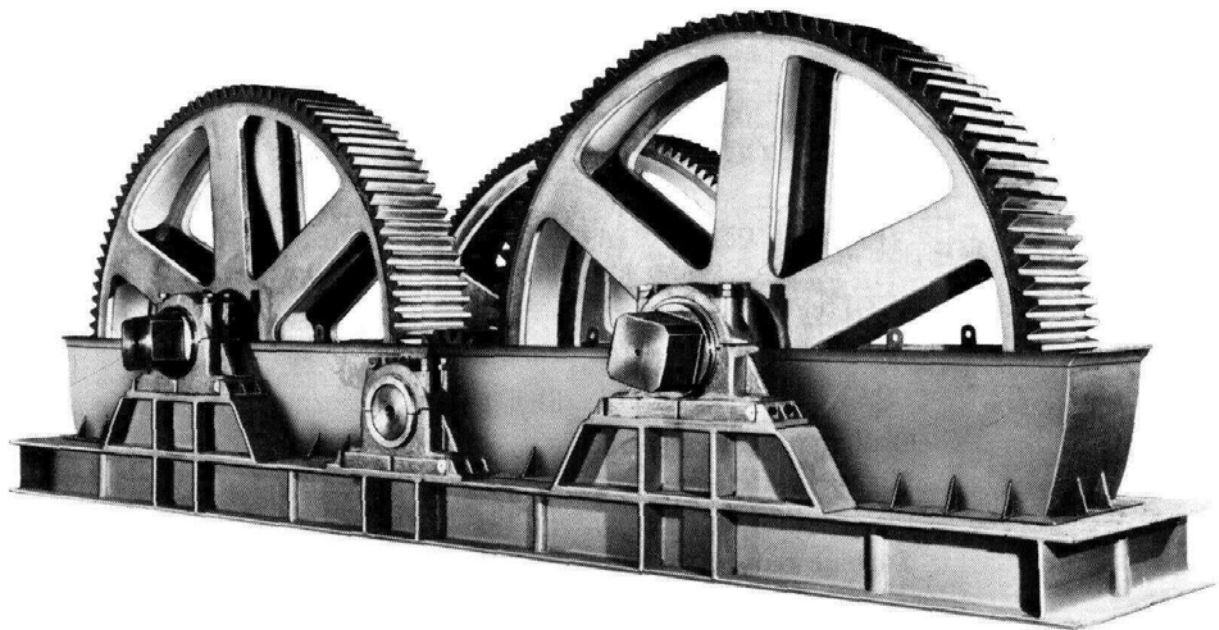


Fig. 17.1 Ordinary spur gearing (BMA)

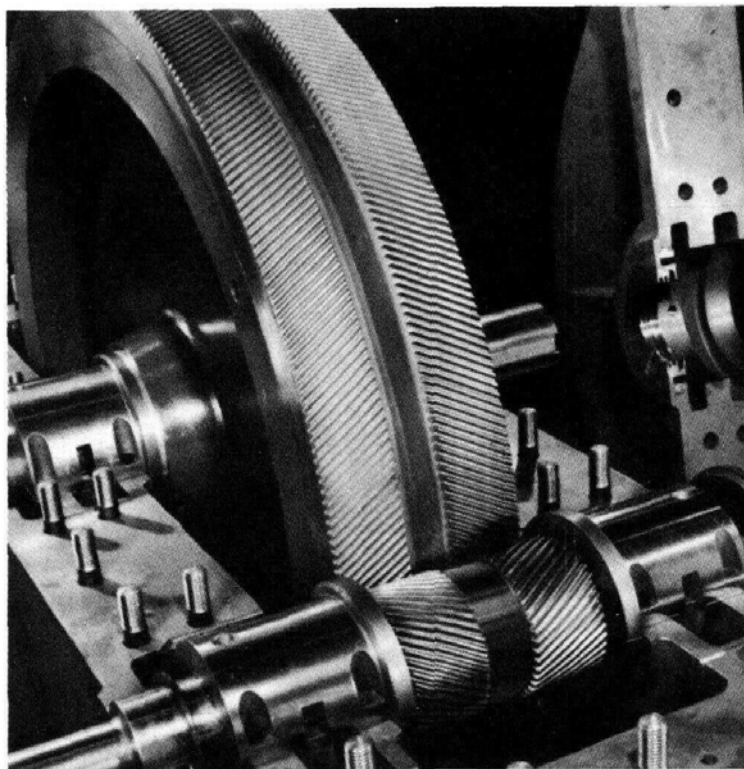


Fig. 17.2. Double helical gearing (Fives Cail - Babcock)

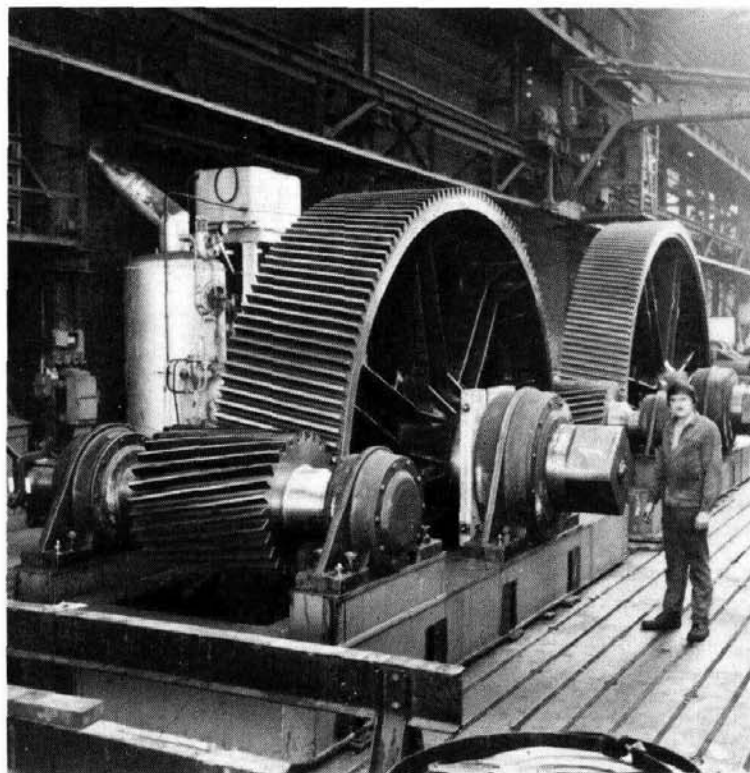


Fig. 17.3. Helicoidal gearing (Fives Cail - Babcock)

Peripheral speed. It is generally accepted that spur gearing can be operated up to a peripheral speed of about 700–825 m/min (2,300–2,700 ft./min). Double-helical gearing may attain considerably higher speeds.

Oil consumption. This may be reckoned as approximately 0.035 l/h (0.0077 gallons/h) for each bearing of the gearing or of the mill (*cf.* p. 267).

Enclosed gearing

The trend is more and more to replace the first, or the first and second stages, of the gearing by a precision reduction gearing with oil bath, and enclosed in a gear case. These speed reducers are made in specialised machine-shops. They are expensive, but much less bulky than ordinary gearing, consuming less lubricant and giving a much higher efficiency, of the order of 98–99% for each stage, compared with 90–95% for ordinary gearing and 95–98% for double-helical gearing.

The only factor limiting their use and preventing their substitution for the last-motion gearing of mills of ordinary dimensions, is the maximum torque of which they are capable. As long as this limit (which is increasing progressively with technical progress) is not reached it is of advantage to make use of such speed reducers.

In a turbine-driven mill, for example, the range of speeds will be as follows (Fives Cail–Babcock tandem of 920 x 2,000 mm (36 x 79 in.), at Surac, in Morocco:

The turbine is designed for:	4,500-9,000 r.p.m.	}	$K = 7.5$
The speed reducer attached to the turbine reduces this speed to:	600 - 1,200 r.p.m.		
The independent speed reducer to:	22- 44 r.p.m.	}	$K = 27.3$
The helical gearing (pinion and wheel) to:	3 - 6 r.p.m.		

DRIVE TO THE ROLLERS

Since the top rollers have to lift by some millimetres, their drive must be designed for and must allow of such movement. For this purpose, the end of the shaft of the final stage of gearing is provided with a square end; the end of the roller shaft is given the same form, and between the two is interposed a shaft of the same cross-section, termed a "tail-bar". The tail-bar is attached to the ends of the shafts, which it serves to connect, by means of two couplings (Figs. 17.4, 17.5).

For purposes of mounting, the tail-bar must have a length 75 - 100 mm (3 - 4 in.) greater than the combined length of the two couplings, and 13 - 25 mm ($\frac{1}{2}$ - 1 in.) shorter than the interval between the two shaft ends which it is to connect. The ends are often cut straight, but certain foreign manufacturers make them spherical, their surface forming a portion of a sphere with the centre at the centre of gravity of the tail-bar.

Roller pinions

These are the pinions (sometimes called "crown wheel") fixed on the end of the roller shafts, which permit the two lower rollers to be driven by the top roller.

They may have from 14 to 21 teeth. The smaller their number, the larger and longer are the teeth, and the greater the range of settings for which a practicable engagement of the teeth can be obtained when the rollers become worn (Figs. 10.25 and 10.26).

However, when the pinions are worn, a 14-tooth pinion gives a slightly jerky motion, and at each tooth the lower rollers make a very slight but perceptible pause. Tromp¹, for this reason, recommends pinions of 17 teeth. The French manufacturers sometimes keep to 14 teeth, but now tend to adopt 16 or 18.

Normal pinions should permit of the same relative setting being obtained for roller wear of 6 - 7% of the diameter; it should thus be possible to adjust the setting of 762-mm (30-in.) rollers until they wear down to 710 mm (28 in.) as a final limit. Fulton² has provided the 1,168-mm (46-in.) mills of Belle Glade with pinions permitting a variation of 75 mm (3 in.) between the axes of the rollers, say 6.5% of the diameter.

Keys. The pinions are fixed to the roller shafts by means of keys. These keys should have a taper of 1 in 100.

Previously 2 keys were used, but one is sufficient. The pinions moreover are fixed to the shaft while hot, and often are very difficult to remove.

Cardan shafts

Tail-bar and couplings form the classic and near universal method of connection between

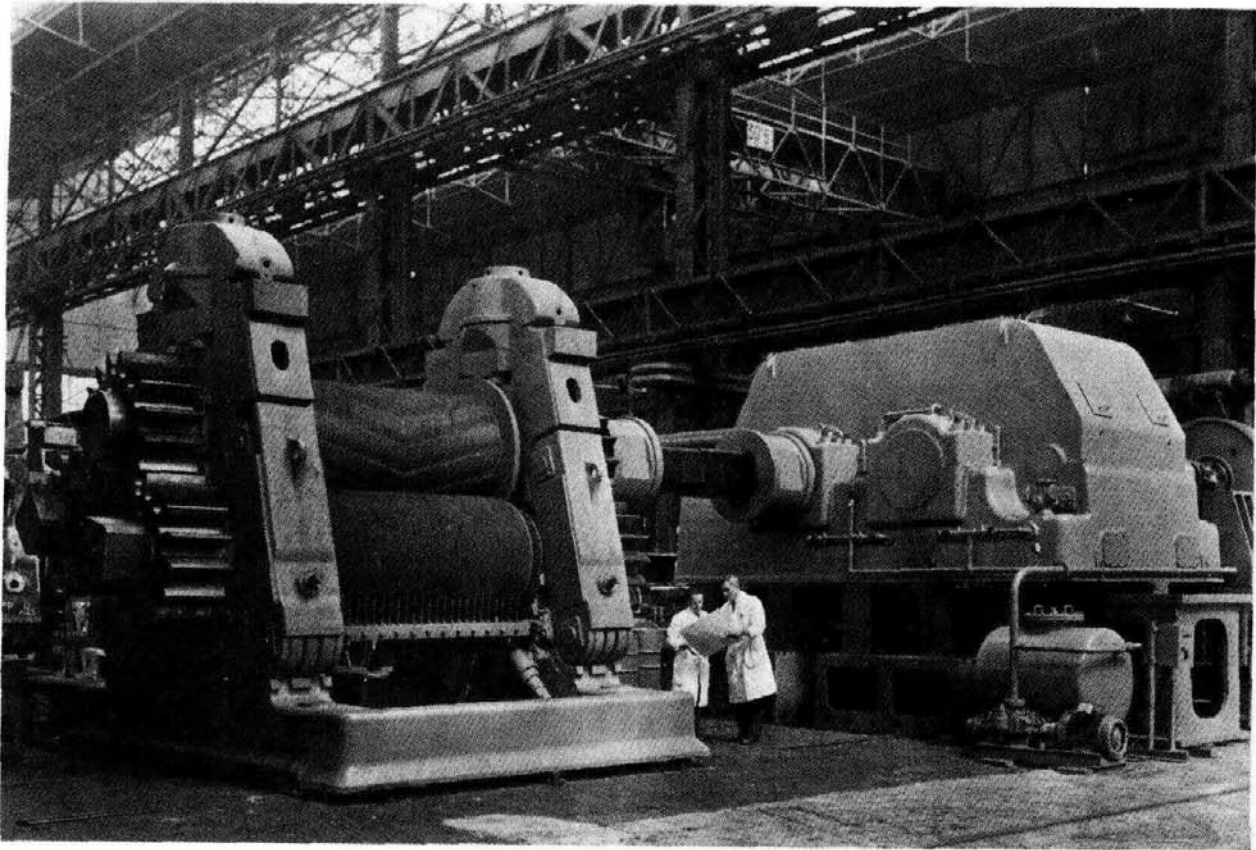


Fig. 17.4. Mill drive by tail-bar and coupling (Fives Cai - Babcock). 1,100 x 2,134 mm mill being one of a tandem of six similar mills, hydraulic load 800 tonnes, speed 3.25 – 6.5C r.p.m., capacity 8,000 tonnes/day driven by 1,500 h.p. turbine.

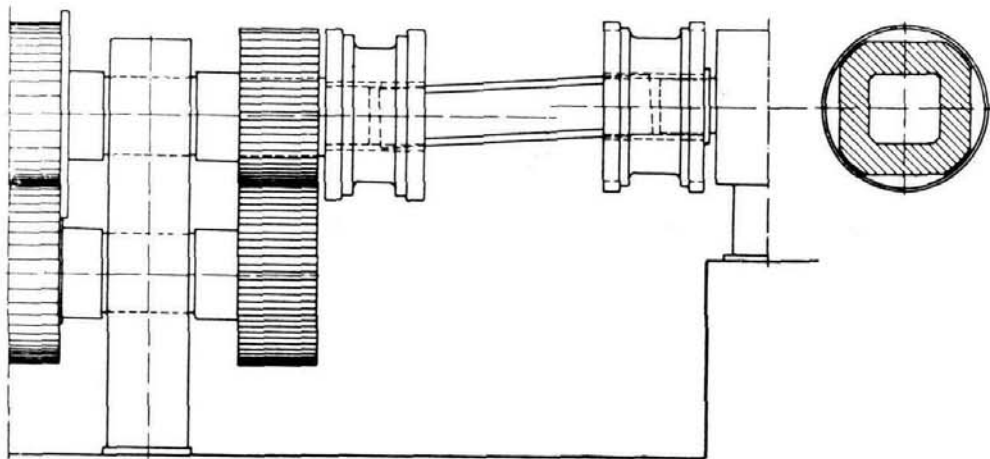


Fig. 17.5. Tail-bar and couplings.

gearing and mill. German manufacturers sometimes make this connection by cardan shaft or universal joint (Fig. 17.6, cf. also Fig. 18.16). This method is more flexible and furnishes a solution which is more mechanical, more elegant and, to sum up, less crude. It avoids the strain, the binding, the warping, and the breakages due to the rigidity of the assembly of tail-bar and couplings. It is more expensive than the standard method of drive, but avoids its worries.

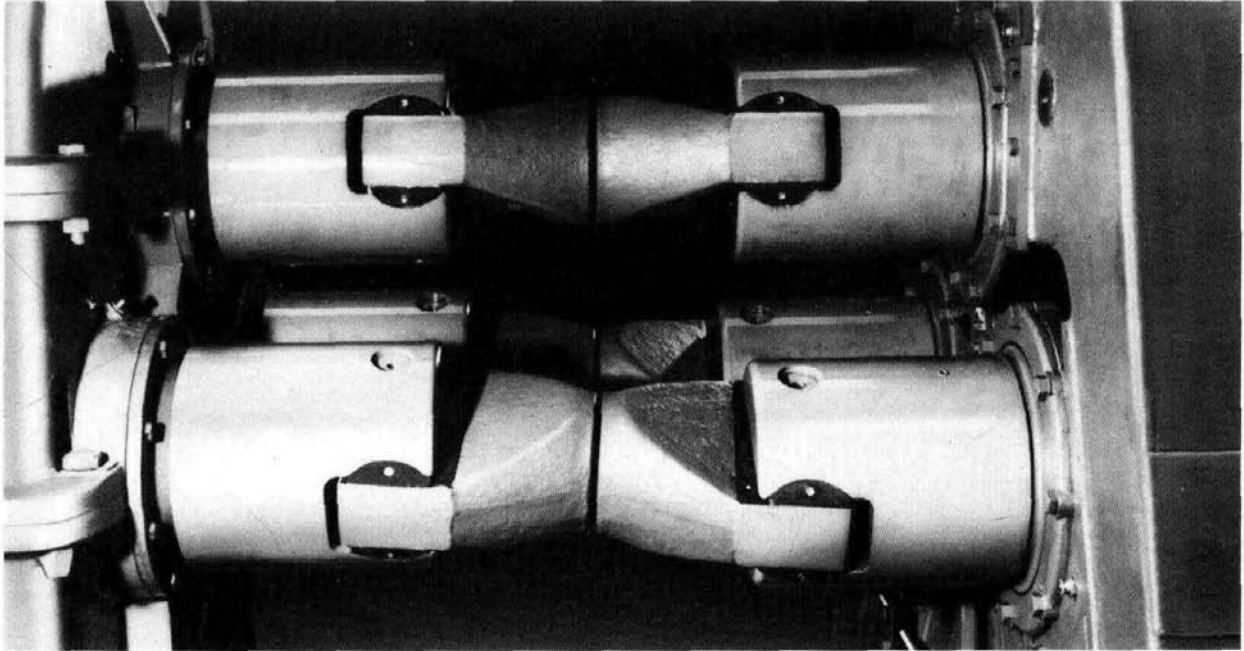


Fig. 17.6 Cardan shafts

Changes in height

At the time of erecting the mill, the levels must be fixed so that the centre line of the final gear shaft is at one-third or one-half of the possible movement allowed for the top roller. If the maximum lift of the roller is 30 mm, the centre line of the final gear shaft should be set at 10 or 15 mm above the lowest position of that of the roller. Since connection by tail-bar lacks flexibility, this precaution is necessary to avoid binding at the extreme upper or lower positions of the roller. When the mill is working at maximum lift, the load on the bearings is greatly increased by this binding effect, and the life of the bearings suffers.

REFERENCES

- 1 L. A. TROMP *Machinery and Equipment of the Cane Sugar Factory*, Norman Rodger, London, 1936, p. 191.
- 2 *Sugar Azucar*, 59 (5) (1964) 68.

18. Construction of Mills

HOUSINGS

Many different types of mills have been built throughout the world, and we have no intention of describing them all. We shall keep to the principal types.

The side frames of mills are designated by the name " housings " or " mill cheeks ". The classic type of housing (Fig. 18.1) carries 2 long bolts set near the vertical, called " kingbolts ", which oppose the lifting force of the top roller, transmitted to the mill caps by the hydraulic plunger and the oil pressure.

Next, designers succeeded in eliminating the kingbolts (Fig. 18.2), but the housing then had a weak spot, between the openings provided for the shafts of the top and delivery rollers. Squier then introduced a new type of housing (Fig. 18.3) in the form of a V enclosing the

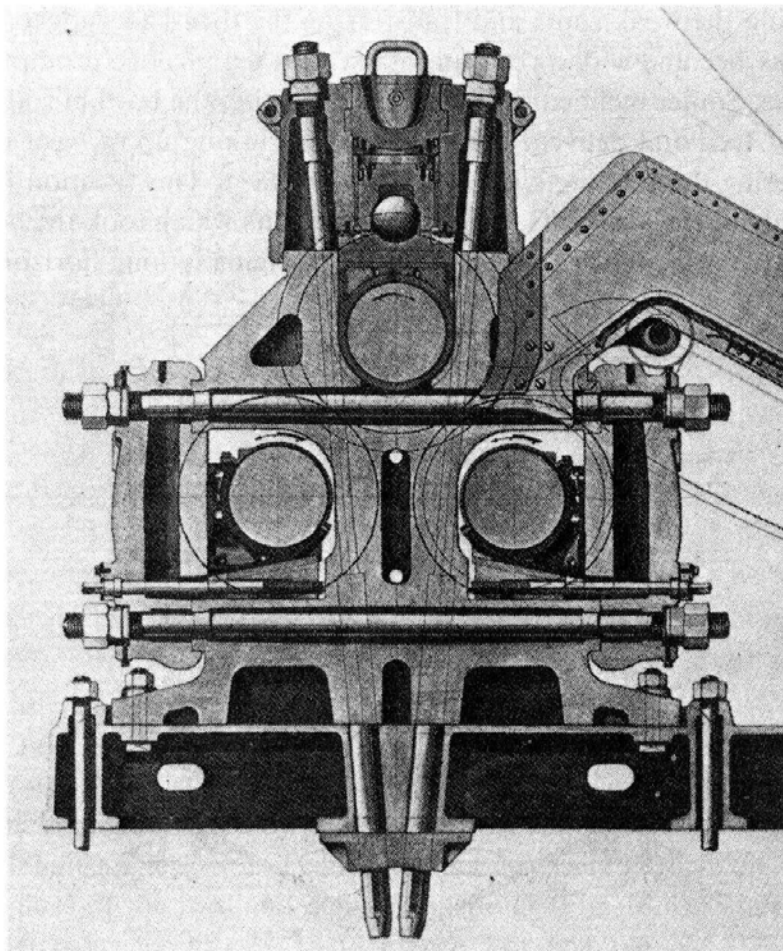


Fig. 18.1. Housing with kingbolts and horizontal bolts.

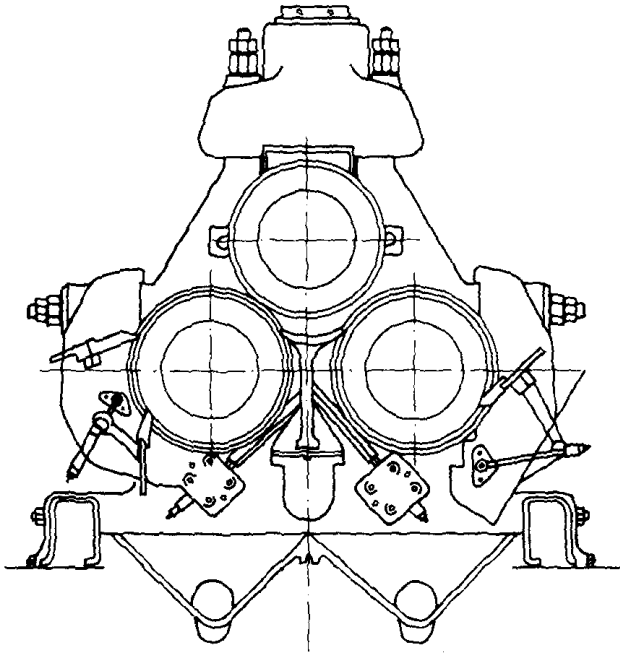


Fig. 18.2. Standard housing.

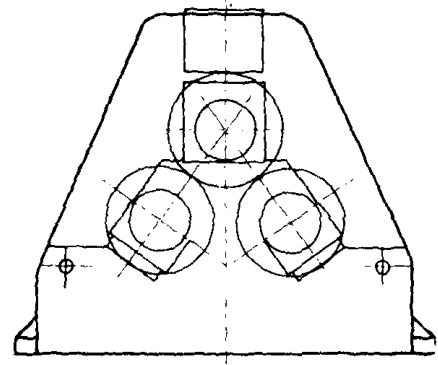


Fig. 18.3. Squier housing.

3 shafts, eliminating the weak spots and transferring the thrust to gudgeons fixing it to the base. This idea has become widespread and most new designs are prompted by it.

At the same time, Squier inclined the plates for adjusting the bottom rollers in such a way that adjustment of feed and delivery openings and the taking up of wear of these rollers is made without altering the top angle of the mill (Fig. 18.3). This solution has permitted the elimination not only of the kingbolts, but also of the bolts which took the horizontal component of the thrust from the lower rollers; these were originally long horizontal bolts passing

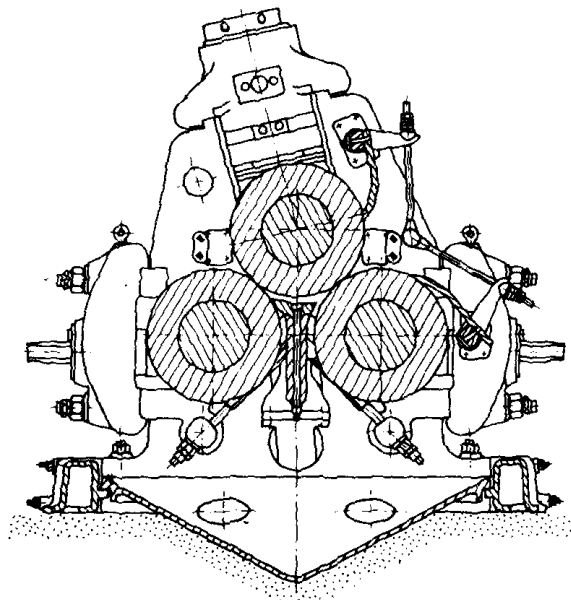


Fig. 18.4. Standard inclined housing.

through the mill cheek from one bearing cap to the other (Fig. 18.1) and later took the form of short bolts which fixed the bearing caps to the main frame of the housing (Fig. 18.2).

The standard housing is symmetrical. If it fractures, it is nearly always on the delivery side. Certain designers take into account the difference between the reactions at the delivery and feed rollers (*cf.* p. 161) by constructing housings with inclined mountings for the top roller (Fig. 18.4). Fives has had the original idea, after inclining the cap by 15° towards the feed side, of turning the whole design forward by 15° , so as to bring the mill cap vertical again. They have thus obtained the housing of Fig. 18.5, the two lower rollers of which are at different heights. This rotation through 15° has the theoretical disadvantage of increasing correspondingly the travel of the juice on the rising face of the lower rollers, but in practice this disadvantage does not show up in the extraction. It has the advantage, moreover, of improving appreciably the feeding of the lower rollers, which gravity then assists at the feed roller and hinders less at the delivery.

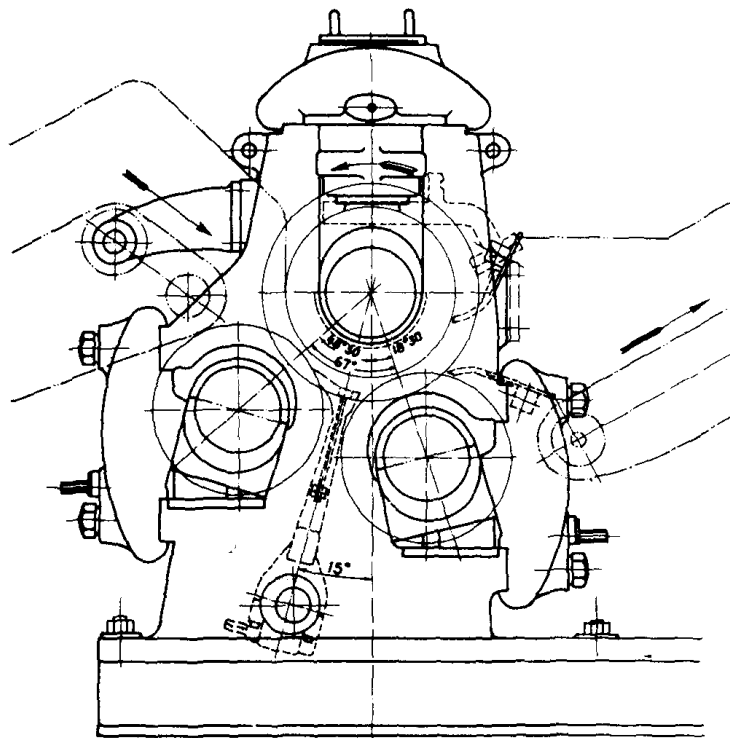


Fig. 18.5. Fives housing, model C46.

Top angle of the mill

This angle, which we have just been discussing, and which we have previously discussed, is the top angle of the triangle formed by the 3 axes of the rollers seen in elevation. The vertical angle consists of 2 half-angles, α_1 and α_2 , which are generally slightly different, on account of the different values of the feed and delivery openings (Fig. 18.6). Since $AB > AC$, $C > B$, and $\alpha_2 < \alpha_1$. There is generally 2-3° difference. For example: $\alpha_1 = 37^\circ$; $\alpha_2 = 35^\circ$; $a = 72^\circ$.

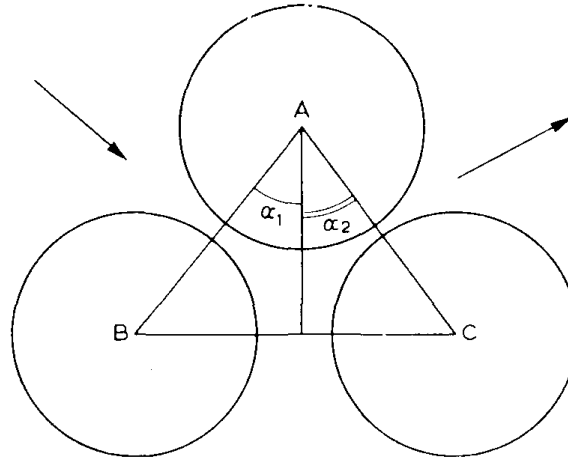


Fig. 18.6. Top angle of mill.

In mills of standard design, the angle a varies from 70 to 85° . A good value for a modern mill is 72° .

Designers aim to reduce the top angle as far as possible. The smaller the angle a , the narrower is the trashplate, the lower is the friction on this trashplate, and the power lost due to this friction (term P_3 in our calculation of Chap. 14) becomes lower.

What limits the angle a is, in the first place, the necessity to allow clearance between the pinions of the 2 lower rollers. This difficulty has been overcome:

- (a) By the system of alternate pinions (*cf.* p. 167)
- (b) By the system of a separate train of pinions (*cf.* p. 167).

Consequently, the angle a is now limited only by the necessity for accommodating the trash-bar between the 2 lower rollers. The smallest angle which we know is that attained by Fives (Fig. 18.5) with 67° , by means of alternate pinions or the "cage a pignons".

Fives mill F63 with fixed ratio

The Fives mill of Fig. 18.5 is the model C46. This mill has been succeeded by model F63, which has an interesting special feature.

We have seen (p. 210) that an effort is made in a mill to establish a certain ratio m between the feed and delivery openings while operating. This ratio varies according to the position of the mill in the tandem, but is generally not far from the value 1.9 or 2.

This ratio, in other words the mill setting, must be based on the tonnage at which the mill is intended to operate. The tonnage, however, is never fixed; it varies between fairly wide limits from one minute to the next, and these changes are expressed in a repeated rise and fall of the top roller. Now the ratio m chosen is valid only for a given position, and one position only, of the top roller.

In a mill of classical design with symmetrical housings, if the setting has been established for a ratio of 2, with a delivery opening $e_A = 15$ and the feed opening E_A of 30 mm, the mill will oscillate continually, for example, between $e_A = 5$, $E_A = 20$ ($m = 4$) and $e_A = 20$, $E_A = 35$ mm ($m = 1.75$).

In a mill with inclined housing, we have seen (p. 210) that the feed opening will alter to a smaller extent than the delivery opening; it will increase or decrease by about 70% only of

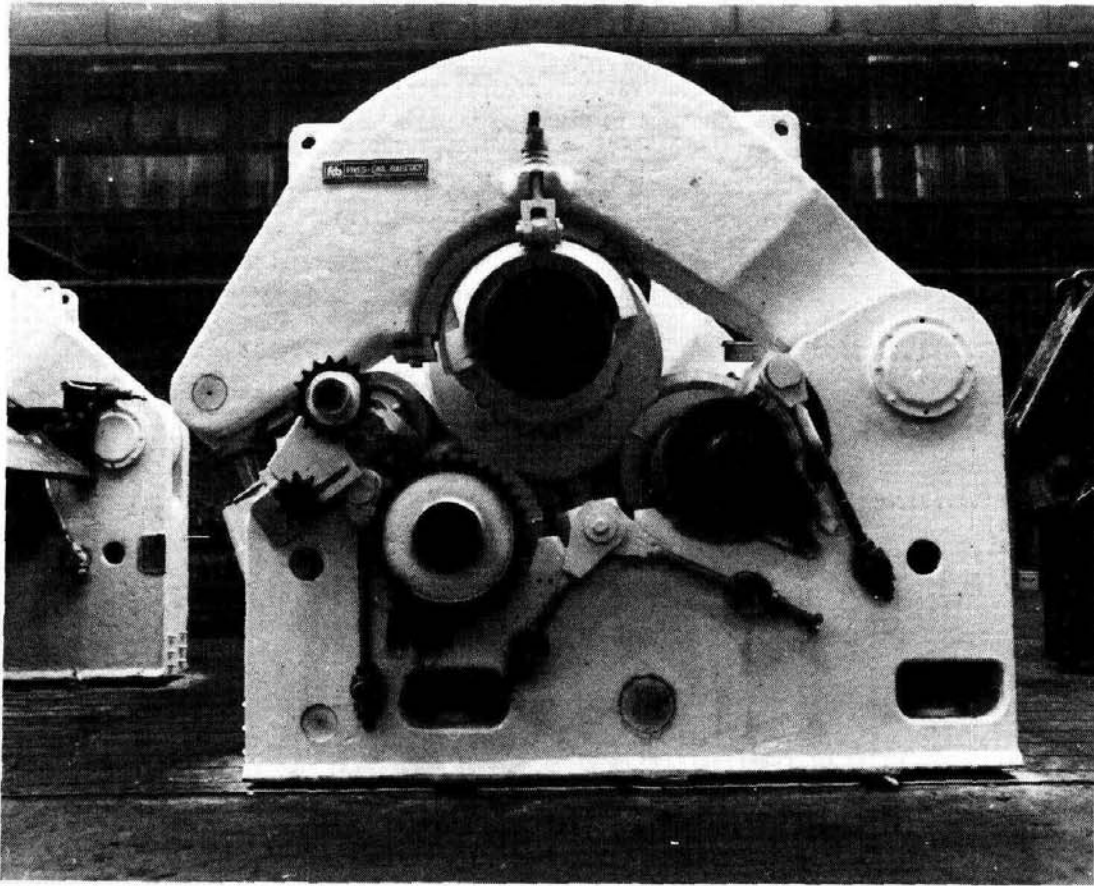


Fig. 18.7. Constant-ratio mill housing (Fives Cai - Babcock).

the increase or decrease of the delivery opening. For the same setting as above, the mill will oscillate between $e_A = 5$, $E_A = 23$ ($m = 4.6$) and $e_A = 20$, $E_A = 33.5$ mm ($m = 1.675$).

Instead of allowing the plunger to rise or fall in a straight line, Fives has conceived the idea of loading the top roller by means of a cap which pivots about an axis situated on the delivery side of the mill and located in such a way that the feed opening opens or closes by an amount approximately double the change in the delivery opening (Fig. 18.7). Since the ratio m is equal to 2, or in the neighbourhood of that figure, it does not vary, or varies much less than with classical housings, where the displacements of the two rollers are equal, or with inclined housings, where the displacements are in the ratio 0.7. The hydraulic plunger is at the same time located on the feed side of the housing; thanks to the benefit of the lever arm, this permits of a reduction of its cross-section by half, but obviously means that its movement is doubled.

It is of interest to ascertain what condition must be fulfilled to realise this fixed-setting mill. Let us fix as the desired value of m , not the approximate value 2, but the value used in Java for all intermediate mills (cf. Table 13.2) or that which we have recommended for the second mill (cf. Table 13.6), which is intermediate between those recommended for the first and for following mills: $m = 1.9$. A diagram completely analogous to that of Figs. 13.4 and 13.5 shows that the ratio of the respective lifts, dE at feed and $d e$ at delivery, depends

only on the angle θ between the plane of the pivots and the centre line of the top roller and the plane (Fig. 18.8) of the top and delivery rollers.

Using dL to designate the arc of the circle described by the mill cap when it rises, and θ the angle made with $O\omega$ and OO_2 , a being the top angle of the mill, and replacing the arcs of the circles with their tangents at O , we have:

$$\frac{dE}{dL} = \cos \gamma = \cos \left[(\alpha + \theta) - \frac{\pi}{2} \right] = \sin (a + B)$$

$$\frac{de}{dL} = \sin \theta$$

and we require:

$$\frac{dE}{de} = \frac{\sin (a + \theta)}{\sin \theta} = 1.9$$

or:

$$\sin (a + \theta) = 1.9 \sin \theta$$

$$\sin a \cos \theta + \cos a \sin \theta = 1.9 \sin \theta$$

$$\sin a + \cos a \tan \theta = 1.9 \tan \theta \quad \tan \theta = \frac{\sin a}{1.9 - \cos a}$$

Since a is approximately 76° :

$$\tan \theta = \frac{0.9703}{1.9 - 0.2419} = 0.5852 \quad \theta = 30^\circ 20'$$

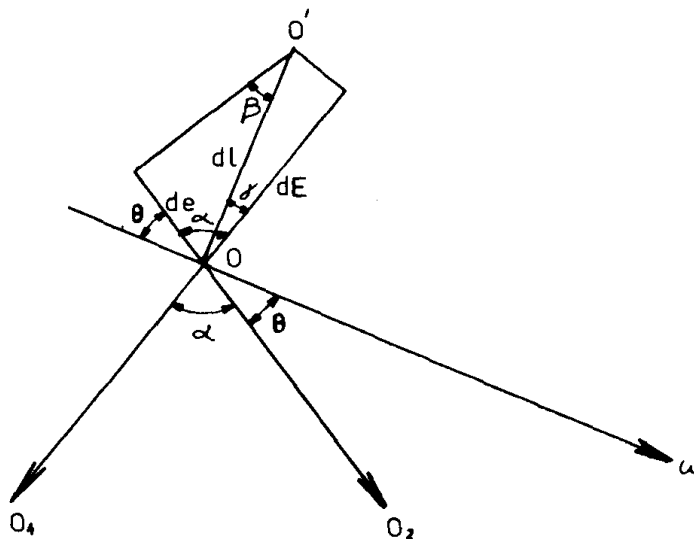


Fig. 18.8. Position of pivot in the constant ratio mill.

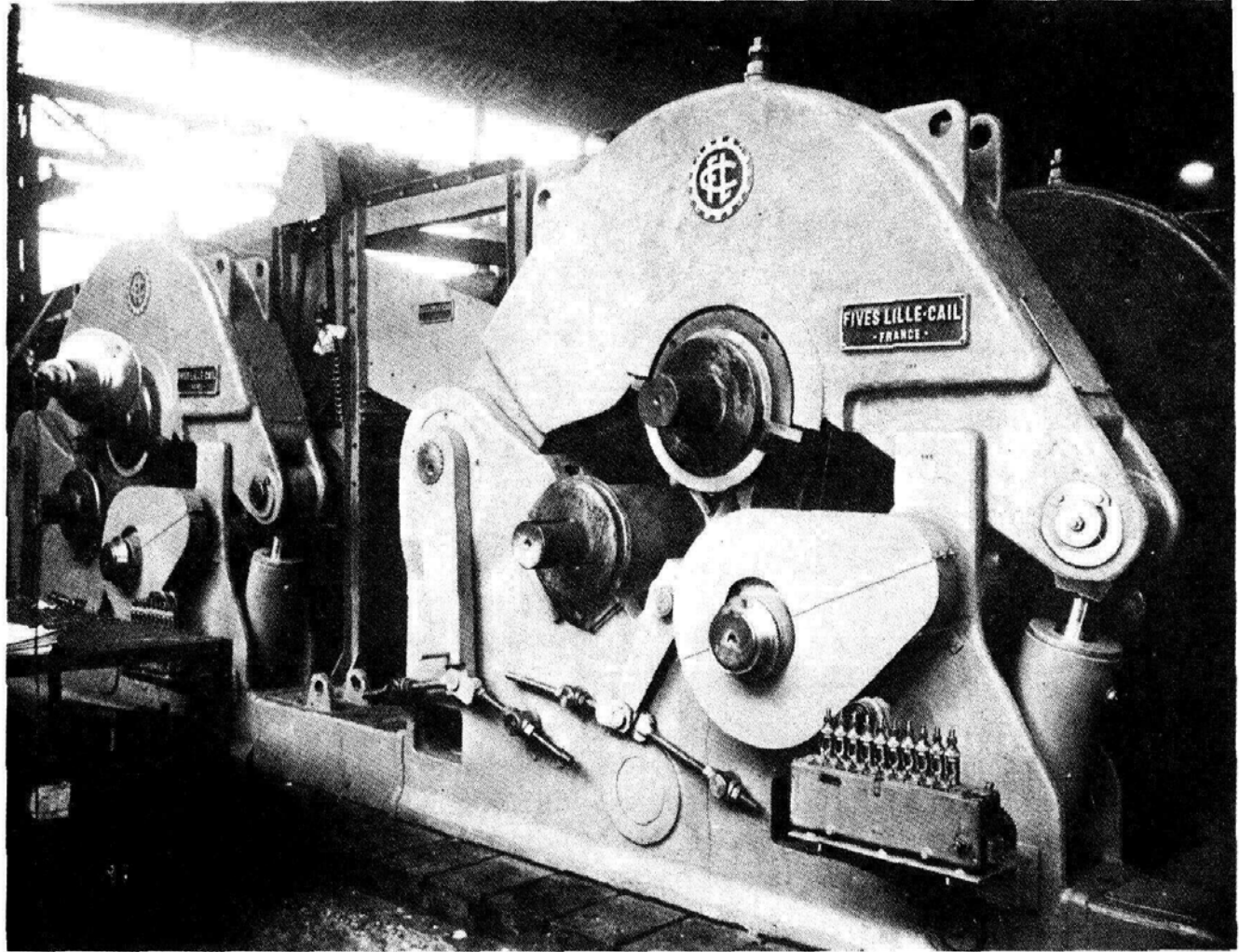


Fig. 18.9. Side view of a constant ratio mill.

Thus it is satisfactory if the pivots α are placed in a plane making an angle of $30-31^\circ$ at O with the axial plane of the two delivery rollers, in order that the entry and delivery lifts should be in the ratio of 1.9. To obtain a ratio of 2, an angle of $\theta = 29^\circ$ would be required; for $m = 2.5$, $\theta = 23^\circ 15'$. α being approximately 76° , the increases in the openings will be in the ratio:

$$dE = dL \sin (\alpha + \theta) = dL \sin (76^\circ + 30^\circ) = dL \sin 106^\circ = 0.96 dL$$

$$de = dL \sin \theta = dL \sin 30^\circ = 0.5 dL$$

An eccentric (Fig. 18.7) permits of adjusting the angle θ between 30° and 23° and thus altering the ratio within the range 1.9-2.5.

This design offers the advantage that the power absorbed is reduced, due to the fact that, with movement of the top roller, friction on the sides of the mill cap is replaced by rotation on an axis with a hinge movement.

In recent models, the feed roller is placed lower than the delivery roller, which facilitates the flow of juice extracted by the two lower rollers.

ROLLERS

The dimensions of the roller journals are determined by the limit of pressure permissible for the bearings (*cf.* p. 125). Their length l is generally made equal to f of their diameter d , and their diameter to half that of the rollers:

$$d = \frac{D}{2} \quad l = \frac{5}{8} d \quad (18.1)$$

Some recent mills make d slightly greater than $D/2$.

The hydraulic pressure should be proportional to LD , and, since d is half of D , it would be logical to make l proportional to L . Hence we should have:

$$d = \frac{D}{2} \quad l = 0.3 L \quad (18.2)$$

Designers, moreover, are compelled to approach the latter proportions with large rollers, since rule (18.1) would otherwise lead to excessive bearing pressures in large mills, the rollers of which have a ratio L/D appreciably higher than that of small mills, according to the series regarded as standard.

The fillets of the journal should have a radius sufficient to avoid the risk of starting fractures, as breakages are more frequent at these points¹. A radius of $\frac{1}{20}$ of the journal diameter is recommended: 2 cm for a 40-cm diameter journal, for example. Fracture of roller shafts most often occurs at the inner side of the mill housing on the driving side.

Fixing the roller shell to the shaft. A mill roller consists of a hollow cylindrical shell of cast iron containing some steel, placed while hot on a shaft of forged steel. The shrinkage allowed is generally one-thousandth, or sometimes 0.0008. The roller is placed on the shell by heating it to a dull red heat and lowering it onto the shaft, which is cooled with water. Perfect uniformity and control of the temperature of heating are necessary to the success of this operation.

Differences in diameter. Certain designers, including Fives Cail–Babcock, give the delivery roller a larger diameter than the other two rollers, the difference being 4–5%. The feed opening being greater than the delivery opening, this permits of retaining the same eccentrics for adjustment of the bearings of the feed and delivery rollers, the more so as the delivery roller wears more rapidly than the feed roller (*cf.* p. 114).

Bearings

The bearings are generally of bronze. The composition of this bronze is rather variable². The following is an example which may be considered typical: Cu, 84%; Sn, 10%; Zn, 3%; Pb, 3%; Brinell hardness, approximately 85.

We show (Fig. 18.11 (b) and (c)) the form of the top and bottom brasses for the top roller bearing, also that of the bearings for the lower rollers (Fig. 18.12).

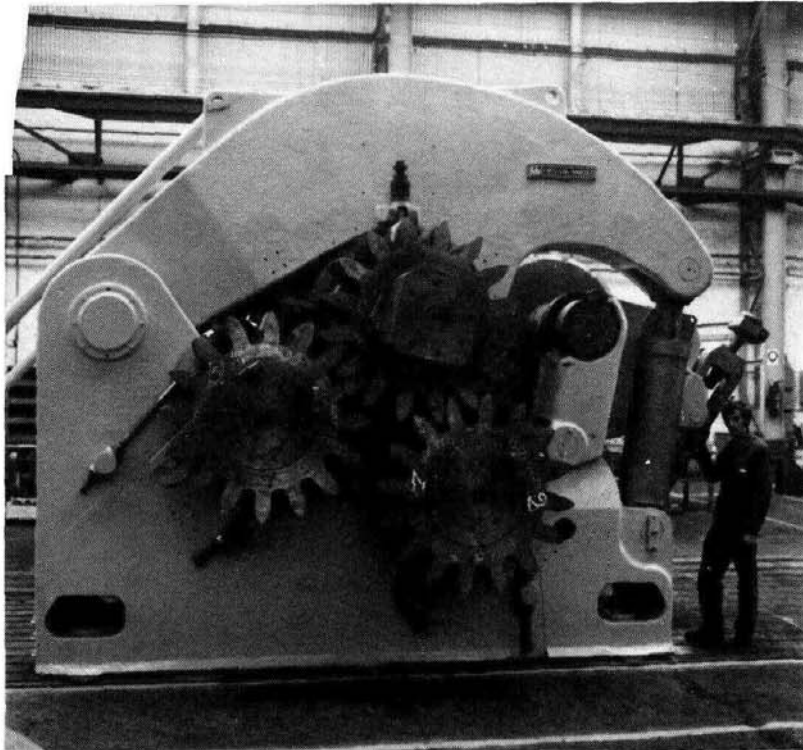


Fig. 18.10 Constant ratio mill, gear side. In shadow, feeder roller and hydraulic plunger.

These are expensive items, since they are provided with passages for circulation of cooling water, and grooves for lubrication of the bearing surface, and are consequently complicated.

Lubrication. Lubrication should be effected by a groove placed along a generatrix, at about 45° ahead of the zone of maximum pressure, bevelled on the side towards which (he shaft rotates, and wider at the centre than at the extremities (Fig. 18.13). It is necessary to feed more lubricant to the centre, since the pressure forces it towards the ends.

It is especially necessary to avoid placing oil grooves in the zone of maximum pressure: this would mean excessive oil consumption, and lubrication would be less effective.

Oil consumption varies between 0.06 and 0.10 l (0.013 and 0.022 gallons) per bearing per hour; as an average, 0.07 l/h (0.015 gallons/h).

This oil should have a viscosity of 3,000-3,200 centipoises at 40°C (175° Engler, 15,000-20,000 SSU) and contain the additives: 4.5% of a mixture of sulphur and phosphorus, 0.5% of an anti-foaming agent, 0.01% of a lubricant agent ("pourpoint depressant").

Cooling water. Cooling water for the six bearings of a mill of $915 \times 1,830$ mm (36 x 72 in.) running at 10 r.p.m. may be reckoned as 1,500-2,000 l/h (330-440 gallons/h). For different dimensions, this quantity should vary as the product nDL .

Bearing guides. It is important that the top bearings should have long guides: they must not bind when the roller tends to lift. Their top edge on the feed side should be slightly round-

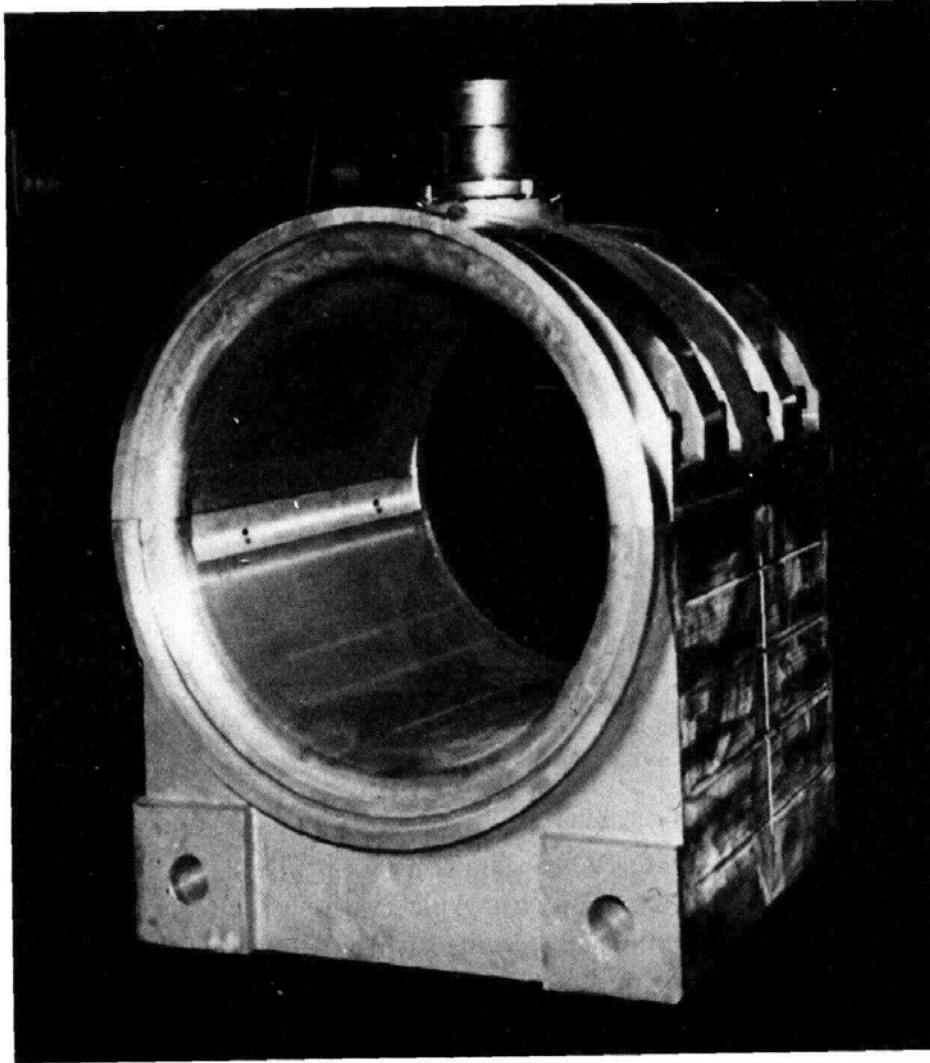


Fig. 18.11. Sliding bearing of top roller.

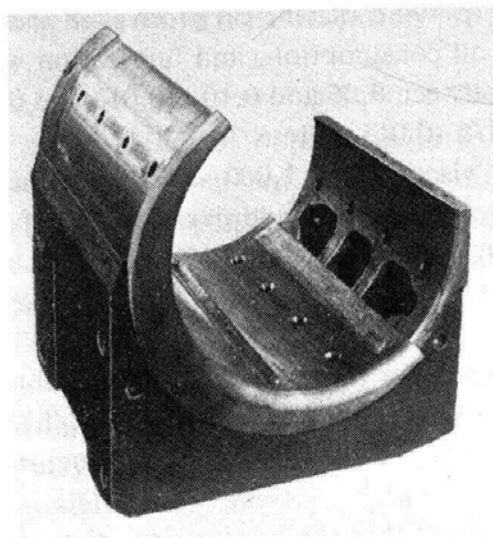


Fig. 18.12. Lower roller brass (shell slightly displaced to show water-circulation passages).

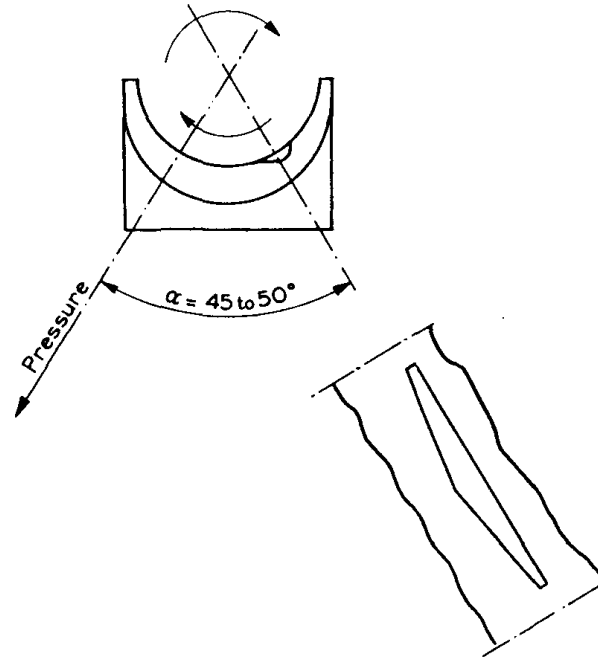


Fig. 18.13. Lubrication of a mill bearing.

ed, to avoid jamming, and the face of the brass should be very carefully finished, smooth and polished, similarly the surface of the housing on which it slides. It is of advantage to lubricate these surfaces by a special oil inlet. Some manufacturers provide inserted plates of special material, giving a very low coefficient of friction.

Antifriction bearings. White metal bearings have been tried at the Sucrieries d'Egypte. They have the advantage that they can be easily re-metalled at the factory, and that they never heat up. On the other hand, they fail without warning if the cooling water fails. It appears that they have given satisfaction in this instance, but their use has not spread to any extent.

Roller bearings. The B.M.A. organisation supplies mills in which the rollers are mounted in roller bearings instead of ordinary bearings (Fig. 18.14). This arrangement has certainly demanded the solution of several delicate problems. It has the inconvenience of requiring a relatively high top angle of the mill, but still in the normal range, *i.e.* 73° . However, it presents valuable advantages:

(a) Large reductions in power consumption, since the term for bearing friction in the equation for power is eliminated; this represents almost 20% of the total power absorbed by the mill, since friction of the rollers is negligible relative to sliding friction of a shaft on the bearing metal. We can reckon generally on a saving of one-sixth of the power consumption (*e.g.* 250 kW in place of 300).

(b) Elimination of the necessity for cooling the bearings, with economy in cooling water.

(c) Similar elimination of central lubrication of the mill. The bearings are packed with grease at the beginning of the season, and re-greased periodically.

(d) Reduction in consumption of lubricant.

The bearings are in enclosed housings and protected from contact with juice by a layer of grease. The mill housings are of the Squier type (Figs. 18.3 and 18.15), hence removal and replacement are simplified.

B.M.A. stress the long life of the roller bearings and the journals, and also the simplicity of regulating the settings of the mill rollers and trashplate. They add a claim of improved extraction. In the absence of tangible results, one cannot contradict the latter claim, which however appears ill-founded; but the other advantages are incontestable. The cost of roller-bearing mills is not stated, but it is probably rather high. Fig. 18.16 shows a mill of this type, with individual drive of the three rollers by cardan shafts, without the nest of pinions, which is replaced by a triple-reduction gear which takes up little space.

"Lotus" roller. The Wen Hsing Company of Taiwan has placed on the market a roller designed by John Bouvet which has been named the "Lotus" roller. It replaces the ordinary top roller, and consists of a cast steel roller with longitudinal cylindrical passages or pipes,

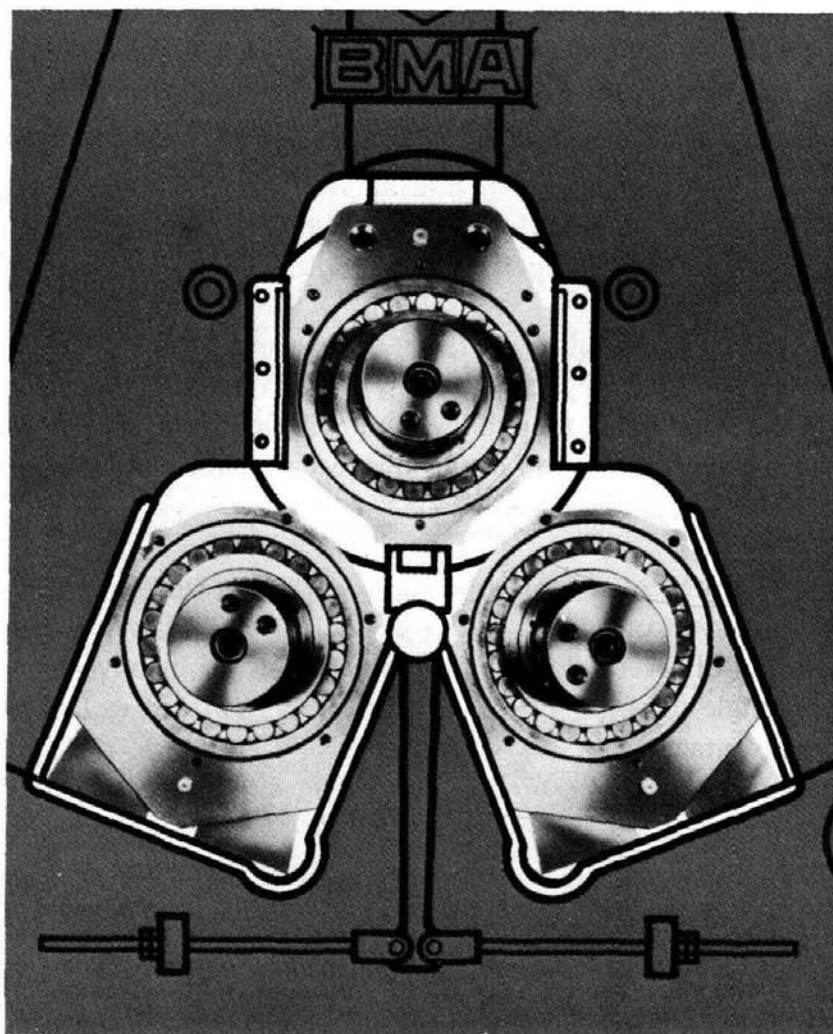


Fig. 18.14. Mill with roller bearings (B.M.A.)

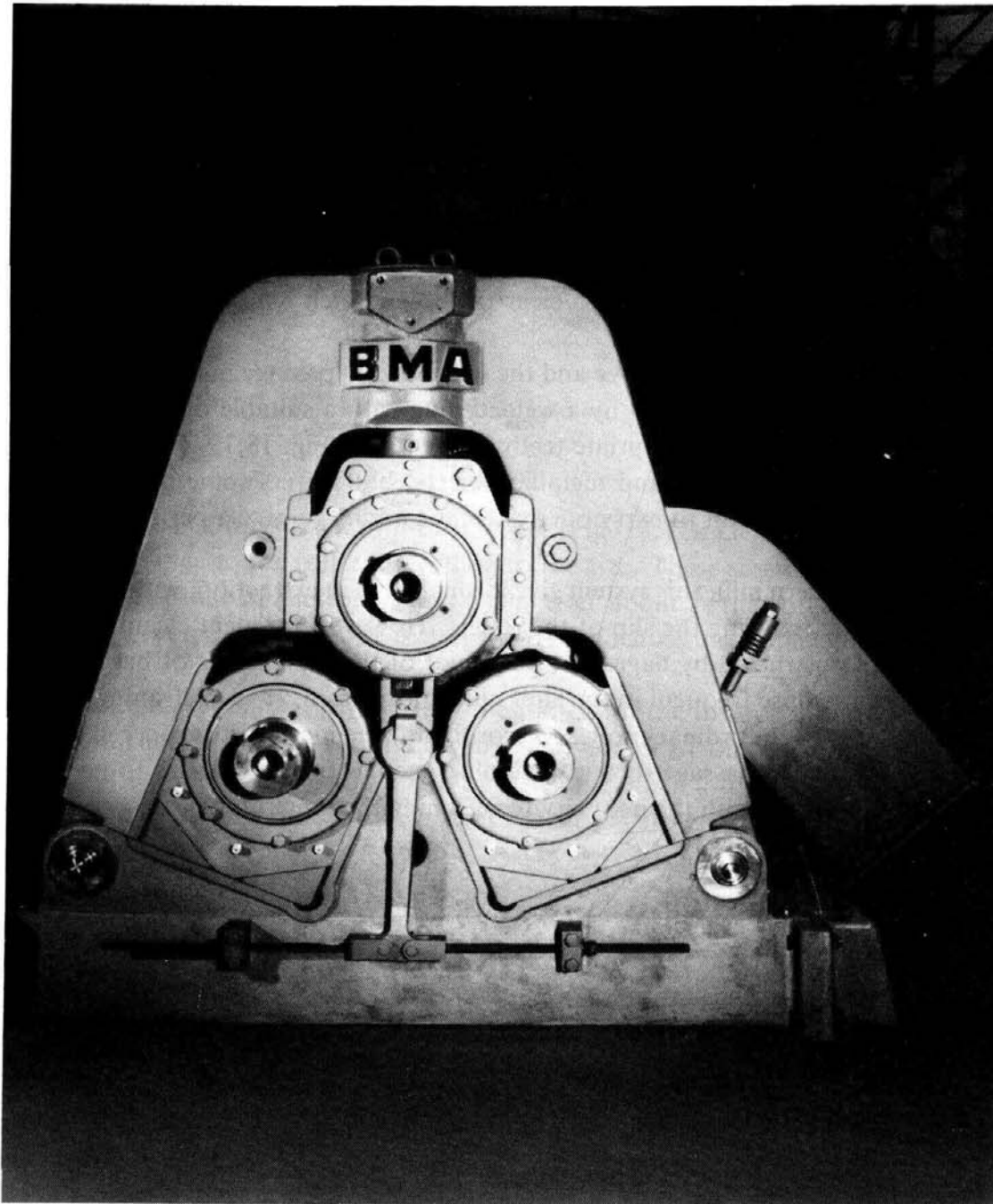


Fig. 18.15 Housing of a roller bearing mill (B.M.A.)

about 12 in number, located concentric with the axis of the shaft, and located a few centimeters below the roller grooves (Fig. 18.19). At the bottom of each groove, or alternate grooves, and just above each passage, is located a hole connecting the groove with the longitudinal pipe. When these holes arrive in the zone of maximal pressure, they give an outlet for the juice which then leaves the roller by the flanges at the end of the roller.

Such a system has the object of allowing ready escape of juice from the zone of maximal compression, with consequent reduction of re-absorption, permitting of higher imbibition, improving the extraction and reducing the tendency to mill slip. It is the last effect which

is more marked in practice. There is a gain in extraction, but it is not large in modern mills. In some tandems where the extraction is low, the gain would be more important. On the other hand, the increase in capacity due to the reduction of slip is appreciable and can attain in certain cases 30%, which would thus allow an increased tonnage to be obtained from a mill of given dimensions. To materialise this increase, all mills of the tandem should be equipped.

By examining the space between the flanges and mill cheeks by the aid of an electric torch, it may be verified that the passages deliver juice. Blockage of the passages can occur, and these can be cleared by blowing steam or compressed air into the end of the passage. During the weekly shut-down, it is necessary to clear the holes which may be blocked and to clear the longitudinal passages.

Communication between the groove and the longitudinal passage may be by a simple hole in the roller, but has been replaced by a welded plug with a suitable opening, placed every second groove, between the appropriate teeth of the roller (Fig. 18.19). The steel roller suffers less breakages from stones and metallic objects, but suffers some deformation which must be rectified. It is useful to carry out systematic arcing to give a permanent roughened surface on the roller.

Installed on a modern mill, this system gives a small gain in extraction, with a more definite gain in capacity, by reducing the slip of the rollers. However, frequent cleaning is necessary to clear passages obstructed by bagasse; on account of the dimensions of bagasse particles, these blockages are more frequent in the later mills of the train. If such cleaning is neglected, the roller functions as an ordinary roller. According to the manufacturer, a recent improvement in the form of the passage through the plug enables such blockages to be completely avoided, thus dispensing with the necessity for cleaning the longitudinal passages with steam or compressed air.

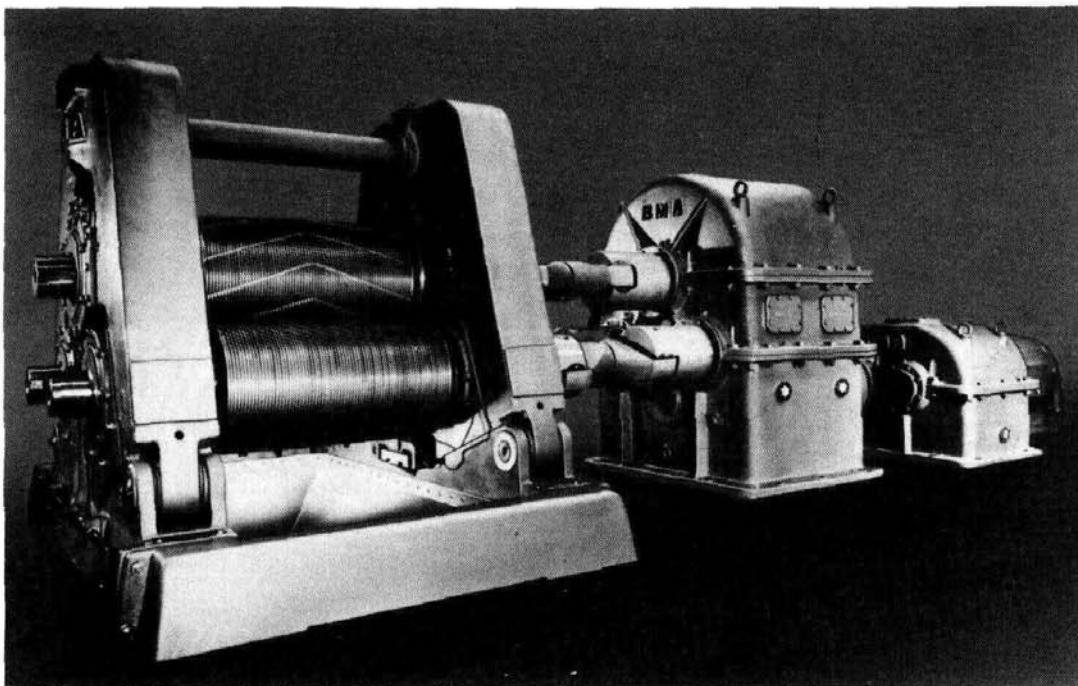


Fig. 18.16. Mill with roller hearings driven by cardan shafts (B.M.A.).

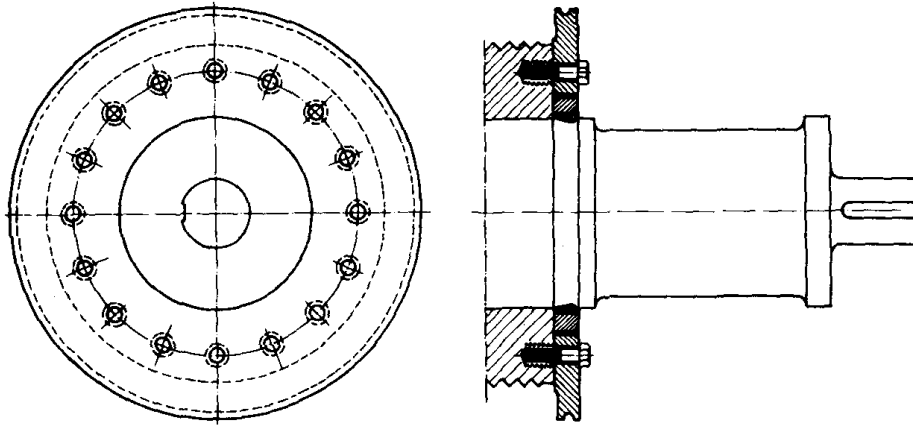


Fig. 18.17. Attachment of flanges.

Flanges

The flanges are now always fixed to the top roller (Fig. 18.17). Generally, 12- 18 screws are used to fix the flange to the roller, according to the size. These screws should have a useful length of about $\frac{1}{10}$ that of the roller, and a diameter about $\frac{1}{30}$ that of the roller.

The flanges become worn, and it is necessary to machine their inner faces at the end of the season.

The clearance to be maintained between the flanges of the top roller and the lateral faces of the bottom rollers is about 1.5 mm ($\frac{1}{16}$ in.). The object of this clearance is to permit of unequal lift of the top roller, without risk of jamming, when a foreign body passes towards one end of the rollers. For the same reason, it is advisable to give a slight conicity to the side of the flange facing the roller.

Scrapers

The surface of the scrapers should make an angle of $45 - 80^\circ$ with the surface of the roller (Fig. 18.18).

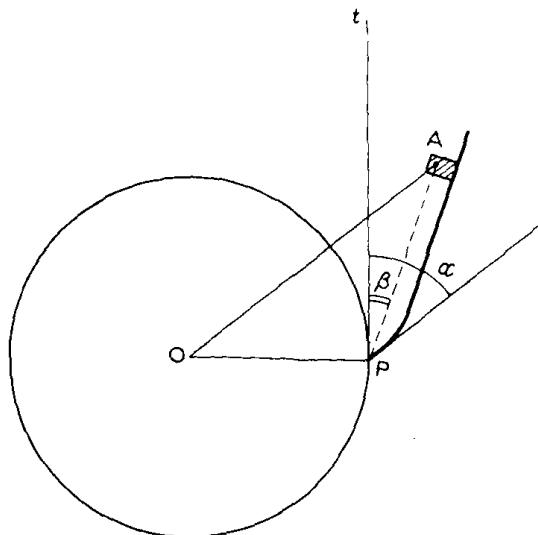


Fig. 18.18. Position of scrapers.

When the scrapers become worn, the point P of contact approaches the plane AO formed by the plane of the axis of the roller and the shaft of the scraper. There is a degree of wear for which there is a risk of the scraper being turned back by jamming against the surface of the roller, and the angle β must not be allowed to increase beyond a maximum of $35 - 40^\circ$.

Standard dimensions

Although some manufacturers do not follow them, there exists an international series of roller dimensions which is very generally adopted. It is of American origin and consequently based on dimensions in inches.

The lengths are in multiples of 6 in., increasing in steps of 6 in. The diameters are in multiples of 2 in., increasing in 2-in. steps. There are 2 successive diameters corresponding to each length, and 2 successive lengths corresponding to each diameter. Table 12.3 (p. 192) gives the combinations so obtained.

A mill dimension is specified by the figure for the diameter followed by that for its length. The dimensions most frequently employed are given in Table 18.1.

TABLE 18.1

COMMONEST STANDARD DIMENSIONS OF ROLLERS

<i>(Inches)</i>	<i>(Millimetres)</i>
24 x 48 =	610 x 1,220
28 x 54 =	710 x 1,370
30 x 60 =	760 x 1,520
32 x 66 =	810 x 1,675
32 x 72 =	810 x 1,830
34 x 78 =	863 x 1,980
36 x 84 =	915 x 2,134

The smallest roller dimension met with in the sugar factory is 508 X 915 mm (20 x 36 in.). However, some very nice little tandems of 355 x 508 mm (14 x 20 in.) are found in cane juice distilleries.

With large mills, there is a very definite trend towards large diameters, even departing from the standard series. In very large mills in particular, several tandems of 1,066 x 2,134 mm (42 x 84 in.) are in existence, in factories such as Central Ledesma in Argentina, and at Darnall in South Africa. The biggest mills known to the author are in Australia³, measuring 1,250 X 2,500 mm (49 x 98 in.); another being installed for the 1983 season⁴ of 1,375 x 2,750 mm (54 x 108 in.). On account of the exceptional size of this mill, it is of interest to give several details. It will be fitted with pressure feeders, rotating at 4.66 r.p.m., the mill rollers rotating at 3.59 r.p.m. (coefficient 1.3), and will be provided with a feed roller below the bagasse running at 5.17 r.p.m. It will be preceded by a shredder running at 1,100 r.p.m., driven by a steam turbine of 492 kW (660 h.p.) for approximately 50 kW/t.f.h. (65 h.p./t.f.h.). The rollers weigh 42 tonnes each, the mill and its drive totalling 600 tonnes. It is designed for a capacity of 16,000 t.c.d. (approximately 667 t.c.h., or 92 t.f.h.). It will be driven by two steam turbines of 820 kW (1,100 h.p.), each roller having its own tail-bar.

Figure 18.21 gives dimensions of rollers of the main series for three manufacturers.

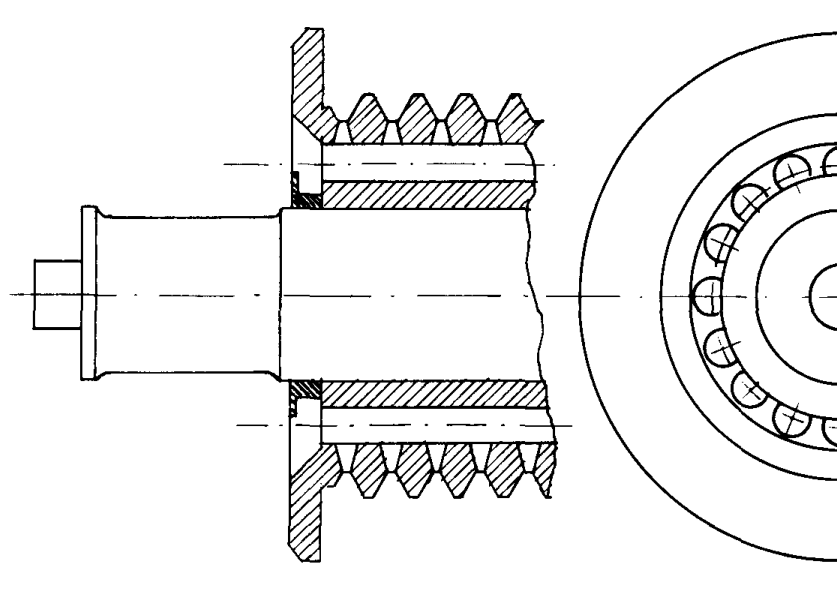


Fig. 18.19. Lotus roller.

Considerations on length-diameter ratio

We have shown on the graph of Fig. 18.20 points representing standard dimensions of rollers, the abscissae showing the diameters and the ordinates the length of the rollers. It is immediately seen that the American scale does not correspond to a constant ratio of L/D ; with large rollers, the length is relatively greater.

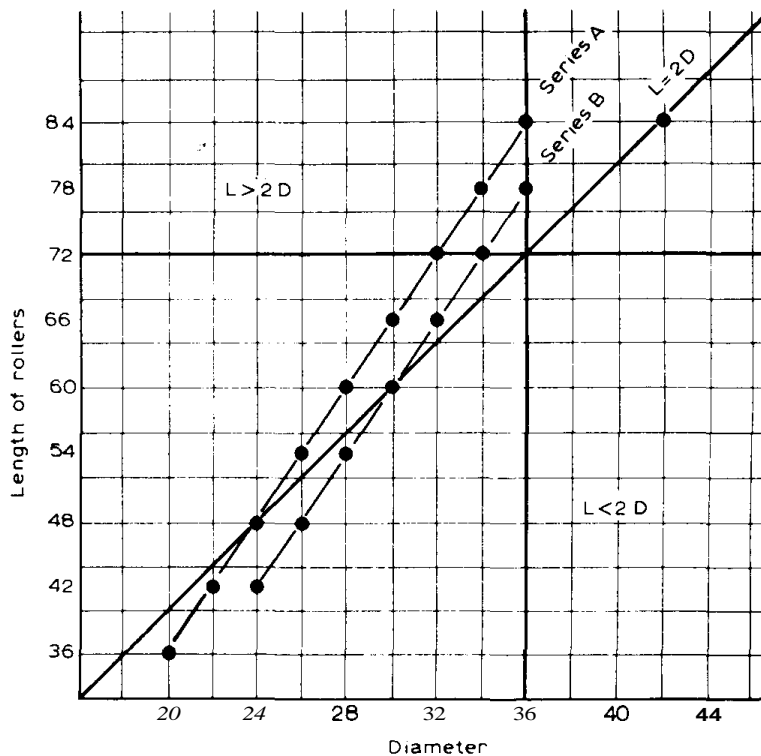


Fig. 18.20. Proportions of mill rollers of standard dimensions.

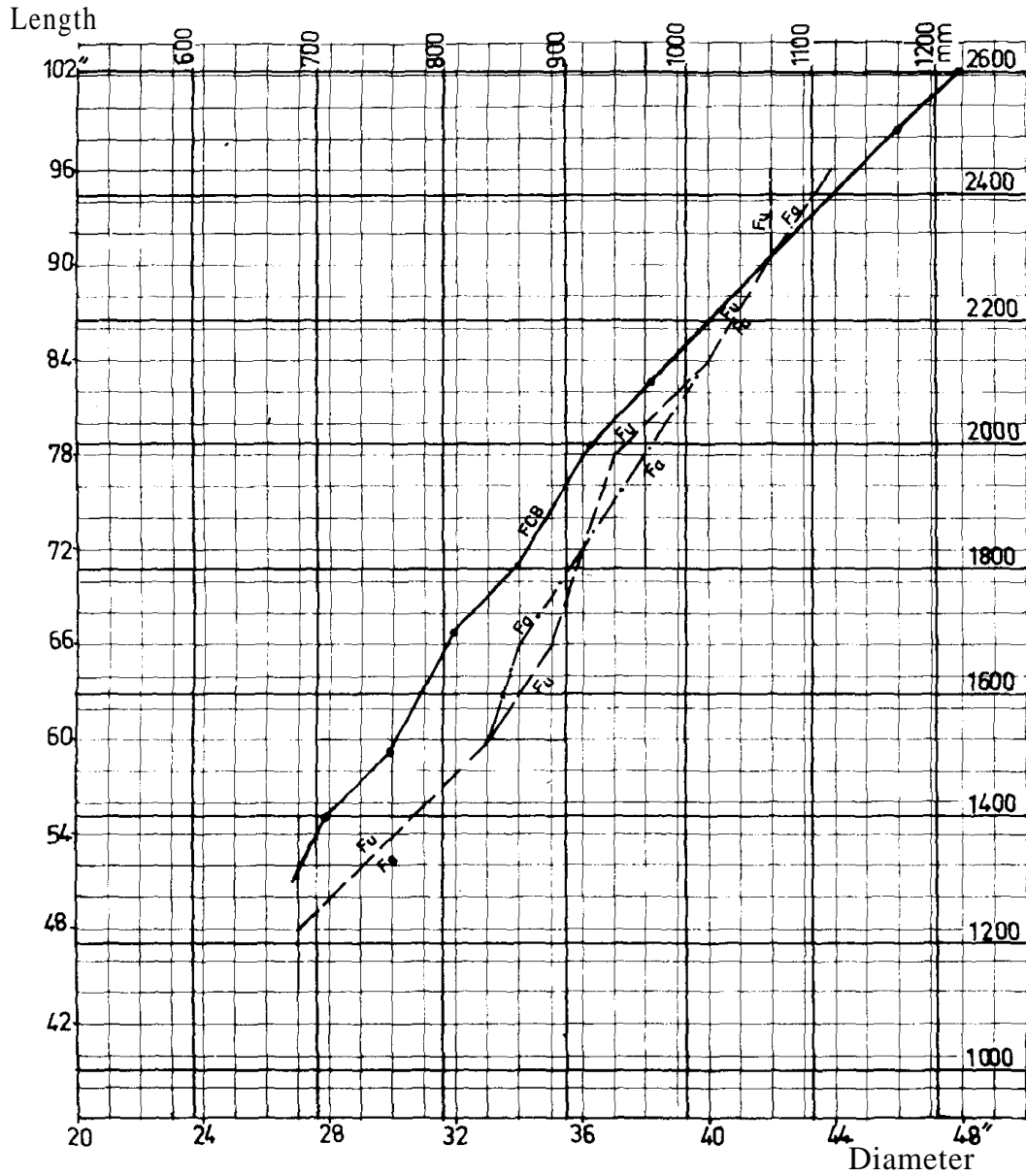


Fig. 18.21. Fulton, Farrel and Fives Cail - Babcock mill series.

Now, since the hydraulic load F should be proportional to LD , the stress due to the shearing load carried by the roller when in operation will vary, for the roller, for the shaft, or for the bearings, all the diameters of which are proportional, as:

$$\frac{P}{\lambda \cdot \frac{\pi D^2}{4}} = \lambda' \cdot \frac{LD}{D^2} = \lambda'' \cdot \frac{L}{D} \tag{18.3}$$

the X factors being numerical coefficients. The shearing force will therefore increase as the length of the roller increases or its diameter decreases.

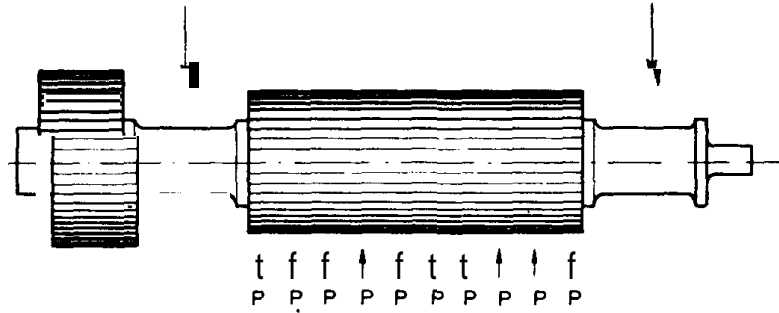


Fig. 18.22. Loads acting on a roller.

Again, the roller may be considered as a beam uniformly loaded and resting on 2 supports (Fig. 18.22). Let p be the pressure per unit length of the roller. The bending moment acting on the roller in operation varies as:

$$p \cdot \frac{L^2}{8} = \frac{P \cdot L^2}{8} = \frac{PL}{8} \cdot \frac{L^2 D}{\delta}$$

whereas the resistance to bending varies only as D^3 . The bending stress then varies as:

$$\frac{\lambda \cdot \frac{L^2 D}{8}}{\lambda' \cdot \frac{\pi D^3}{32}} = X'' \cdot \frac{L^2}{D^2} \tag{18.4}$$

the modulus of resistance to bending for a beam of circular section being $\pi D^3/32$.

Should priority be given to strength in shear and consequently to the rule (18.3), or to bending moment and rule (18.4)?

To answer this question, we shall compare the relative stresses in the metal for the two cases, commenting that:

(a) the load p distributed uniformly on the roller has a value

$$p = \frac{F}{L} = \lambda \cdot \frac{LD}{L} = \lambda D$$

As we are concerned only with relative values, we shall take $p = D$.

(b) The maximum shearing stress, which occurs at the flange of the journals, has a value $T = pL/2$.

(c) The maximum bending moment, which occurs at the mid-point of the roller, has a value $M = pL^2/8$.

(d) The maximum stress due to shearing force is given by

$$t = \frac{4}{3} \cdot \frac{T}{\pi r^2}$$

where r is the radius of the cylindrical beam.

(e) The maximum stress due to bending moment is given by $n = rM/I$, where I is the moment of inertia of the cylindrical beam.

(f) The moment of inertia of a circular section is given by $I = \pi d^4/64$, d being the diameter of the section.

(g) The diameter of the journals carrying the shearing stress is 0.9 of the shaft diameter, which in turn is half the roller diameter.

We have then:

$$(1) \text{ Shear load: } T = \frac{pL}{2} = \frac{DL}{2}$$

$$i = \frac{4}{3} \cdot \frac{DL}{2} \cdot \frac{1}{\left(\frac{0.9 \times 0.5 \times D}{2} \right)^2} = \frac{13.17}{T} \frac{L}{D} \quad (18.5)$$

$$(2) \text{ Bending moment: } M = \frac{pL^2}{8} = \frac{DL^2}{8}$$

The bending load, however, is supported partly by the shaft and partly by the roller shell, assuming that these are firmly held together by three contact areas. If it were supported wholly by the shaft, the stress in the latter would be:

$$n' = \frac{D}{4} \cdot \frac{DL^2}{8} \cdot \frac{1}{64 \times 16} = \frac{32}{\pi} \cdot \frac{L^2}{D^2}$$

If it were supported wholly by the shell, which is of cast iron, the strength of which in bending is 4.5 times less than that of steel, the stress in the shell, relative to its capabilities relative to those of the shaft, would be

$$n'' = \frac{D}{2} \cdot \frac{DL^2}{\frac{\pi}{64} \left(D^4 - \frac{16}{16} \right)} \cdot \frac{4.5}{\pi} = \frac{19.2}{\pi} \cdot \frac{L^2}{D^2}$$

We could thus expect the shaft to support the fraction $19.2/(19.2 + 32)$ of the total load, and the shell the proportion $32/(19.2 + 32)$. The stress in the shaft thus becomes:

$$n = \frac{32}{\pi} \cdot \frac{L^2}{D^2} \times \frac{19.2}{51.2} = \frac{12}{\pi} \cdot \frac{L^2}{D^2} \quad (18.6)$$

Since L/D should be of the order of 2, we see that it is the bending moment which is predominant, though the chances of breakage due to the two causes are similar.

It follows that from the two viewpoints of shearing load and particularly bending moment, the length of the roller is an element of weakness and the diameter a source of strength.

We thus see that the ratio of diameter to length should be constant for all sizes. The solution most in accord with experience, and keeping as close as possible to present practice, would consist of keeping to the line $L = 2D$. If it were desired to extend the standard series beyond 2,134-mm (84-in.) rollers, the necessity for this would be apparent, and it is no doubt this reasoning which has led the designers of recent very large mills to abandon, very fortunately, the standard series and to choose the relationship $L = 2D$ for a 1,066 x 2,134 mm (42 x 84 in.) mill.

From these considerations we can draw three conclusions:

(7) For fabrication of equal quality, and for identical metal, a standard roller of small dimensions (*e.g.* 660 x 1,220 mm (26 x 48 in.)) would be able to withstand loads, that is, hydraulic pressures, relatively higher than could be applied to large standard rollers (*e.g.* 915 X 2,134 mm (36 x 84 in.)). In other words, higher values of s.h.p. could be maintained in the former (*cf.* p. 127).

(2) It is of advantage to choose preferably a mill in the zone $L \leq 2D$ of Fig. 18.20, rather than in the zone $L > 2D$; or at least to approach the former zone. In other words, above 1,220 mm (48 in.), preference should be given to a mill of series B rather than one of series A.

(3) It is unfortunate that Continental manufacturers have not originated a metric series:

600 X 1,200 mm	900 X 1,800 mm
650 x 1,300mm	950 x 1,900 mm
700 x 1,400 mm	1,000 x 2,000 mm
750 x 1,500 mm	1,100 x 2,200 mm
800 x 1,600 mm	1,200 x 2,400 mm
850 x 1,700 mm	1,300 x 2,600 mm

This would give only 12 sizes, the range of capacities covered would be complete, and the series would be more logical and would be better received by well-informed technicians.

Stainless steel. In South Africa⁵, stainless steel No. 430 (17% chromium) is recommended for the following items: Donnelly chutes, juice trays, trays for imbibition juice and under mills, primary juice piping in diffusers, fine-bagasse conveyors, juice weighers, welded sheets to replace oxidised sheeting, and vacuum piping (6 mm ($\frac{1}{4}$ in.) thickness in place of 10 mm (I in.)).

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- 1 R. L. GEORGE AND D. M. REID, *Int. Sugar J.*, 54 (1952) 72.
- 2 *RAM*, (July – Dec. 1958) 305.
- 3 *Sugar Azucar*, (Sept. 1981) 56; (Jan. 1982) 17.
- 4 P. MAYO AND D. WRIGHT, *Proc. ASSCT*, 1983, p. 273.
- 5 *44th Conf. SASTA*, 1970, p. 60.

19. Imbibition

OBJECT

Even when bagasse is subjected to high and repeated pressures, it never gives up all the juice which it contains. It approaches a minimum moisture, 45% in general, 40% in the most favourable cases; that is, it retains a high proportion of juice, amounting roughly to half its weight.

In order to extract as much as possible of the sugar which it retains, it is therefore necessary to resort to an artifice: since this moisture content cannot be reduced, the effort will be made to replace by water the juice comprising it.

It is this artifice which constitutes "imbibition".

With dry crushing, the limit of extraction is very soon attained in practice: after the crusher and first mill, the bagasse moisture has already been reduced to about 60%. After the second mill, it is close to 50%. After the third mill, it is not far from 45%: it may be considered that the maximum extraction possible with simple pressure has been obtained. If, at this stage, the bagasse is sprayed with water, this water spreads in the bagasse and dilutes the juice which it contains. The following mill will then recover bagasse at the limiting moisture, say about 45%. But this moisture will consist, no longer of absolute juice, but of diluted juice. Hence sugar will have been extracted; and the operation may be repeated.

DIFFERENT SYSTEMS OF IMBIBITION

Actually, on account of the small difference between the bagasse moistures, there is every advantage in commencing the imbibition as early as the delivery of the first mill; the gain will be all the more rapid as the juice with which the bagasse is still filled will be more easily diluted and extracted. The quantity of juice extracted is then nearly equal to the quantity which would have been extracted without addition of water, increased by the quantity of water added. The quantity of *sugar* extracted is much greater than that which would have been obtained without addition of water, since the quantity of juice remaining in the bagasse (which, as we have just stated, is very nearly the same) now consists of diluted juice instead of absolute juice.

Simple imbibition

The simplest procedure which comes to mind is to add the water to the bagasse after each mill. This procedure is called "simple imbibition". If the water is added at one point only, between the last two mills for example, it will be called *single* simple imbibition. If it is added at 2 points, between the penultimate mill and the preceding one, and then between the

penultimate and the last, it will be termed *double* simple imbibition; similarly for *triple* simple imbibition, and so on.

Compound imbibition

This system, however, consumes much water, which has to be evaporated later. Starting from single simple imbibition, it has been observed that the dilute juice obtained from the last mill is mostly water; it has then been taken and sent back before the preceding mill. This is what is called "compound imbibition", and, in this case, it is *double*. If the juice obtained in this way from the penultimate mill is again returned before the previous mill, this becomes *triple* compound imbibition; and so on (Fig. 19.1).

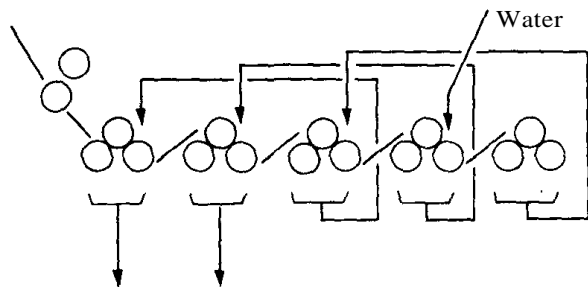


Fig. 19.1. Quadruple compound imbibition.

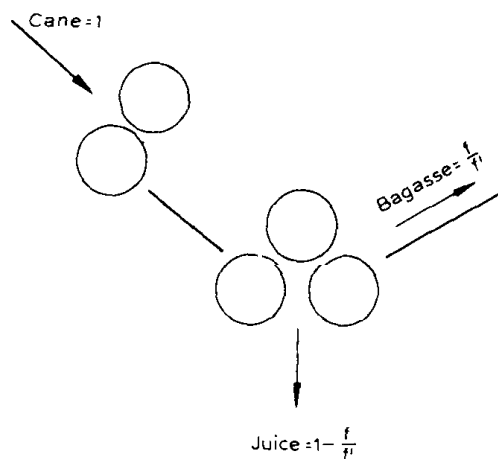


Fig. 19.2 Dry crushing.

MATHEMATICAL STUDY OF IMBIBITION

For the sake of a thorough understanding of imbibition, which is more complex than it appears to many, and which plays an important part in the financial results of the sugar factory, we shall study in order:

- (a) Dry crushing
- (b) Wet crushing
- (c) Multiple simple imbibition
- (d) Multiple compound imbibition.

(a) Dry crushing

Dry crushing consists of all the successive applications of pressure on the bagasse in a milling tandem, without previous addition of any liquid.

We shall assume the presence of a crusher, and group it with the first mill. Suppose we reserve for dry crushing a number n of mills. Let us use for each of these mills the subscripts: 1, 2, 3, ..., n .

In Fig. 19.2, let:

f = fibre per unit of cane

m_p - fibre in bagasse furnished by mill no. p

b_p = quantity of bagasse furnished by mill no. p

j_r = quantity of juice furnished by mill no. p .

Let us consider a unit weight of cane passing through the mills. We have:

$$1 \cdot / = b_1 \cdot m_1 = b_2 \cdot m_2 = \dots = b_n \cdot m_n$$

Whence:

$$b_1 = \frac{f}{m_1} \quad b_2 = \frac{f}{m_2} \dots \quad b_n = \frac{f}{m_n} \quad (19.1)$$

At the 1st mill, we have then:

$$j_1 = 1 - \frac{f}{m_1}$$

At the 2nd:

$$j_2 = 1 - \frac{f}{m_2} - \left(1 - \frac{f}{m_1}\right) = f \left(\frac{1}{m_1} - \frac{1}{m_2}\right)$$

At the 3rd:

$$j_3 = 1 - \frac{f}{m_3} - \left(1 - \frac{f}{m_1}\right) - f \left(\frac{1}{m_1} - \frac{1}{m_2}\right) = f \left(\frac{1}{m_2} - \frac{1}{m_3}\right)$$

At the n th mill:

$$j_n = f \left(\frac{1}{m_{n-1}} - \frac{1}{m_n}\right) \quad (19.2)$$

In practice, it is found that the fibre in bagasse increases rapidly towards a limit m which is always in the vicinity of 0.5. In consequence, the quantity of juice extracted by the successive mills falls rapidly and tends towards zero.

Let m be the fibre content of the bagasse from the last dry crushing (which we have called m_n in the notation adopted). The juice extraction has the value:

$$e' = \frac{\text{juice extracted}}{\text{juice in cane}} = \frac{1 - \frac{f}{m}}{1 - /} = \frac{m - f}{m(1 - f)} \quad (19.3)$$

For normal tandems, where the dry crushing is effected by the first mill only, this formula

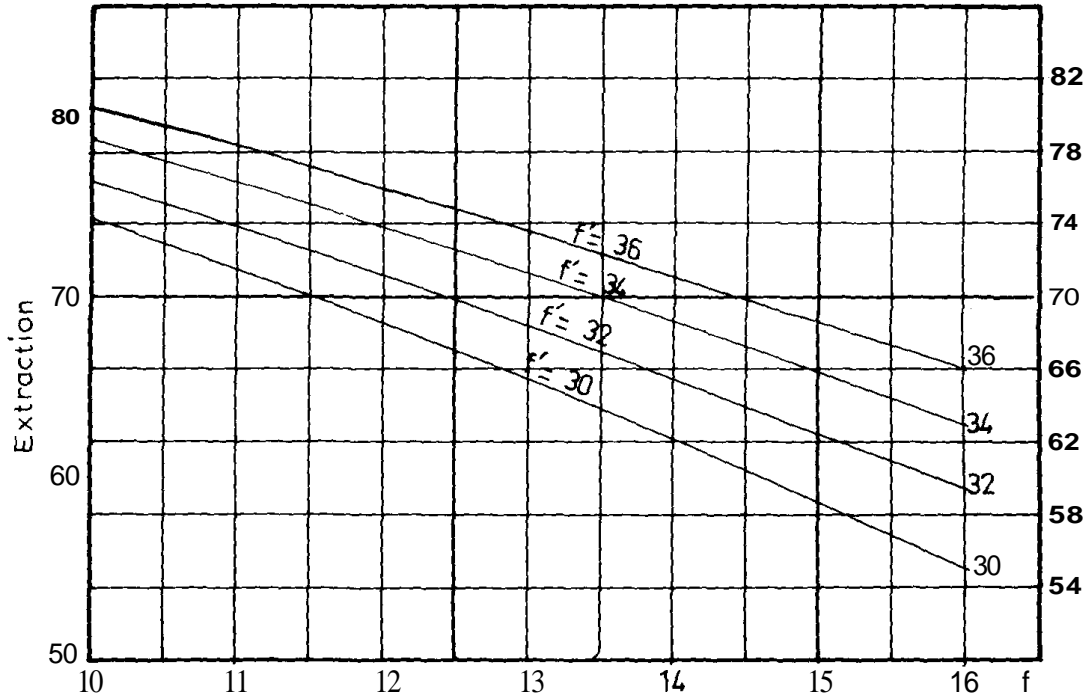


Fig. 19.3. Juice extraction in the first mill.

gives the extraction of juice by the first mill, m being the fibre content of first-mill bagasse (Fig. 19.3). We see the marked influence of fibre in cane, and the influence of the fibre of first-mill bagasse, which is normally of the order of 32–34, and varies only slightly when the fibre f of cane varies from 10–16.

The purity of the extracted juice being higher than that of the residual juice, the extraction of sugar is superior to that of juice, and has a value:

$$e = \frac{\alpha (m - f)}{m (\lambda - f)} \tag{19.4}$$

where α is a coefficient having a value of 1.05–1.10.

(b) Wet crushing

As a first approximation, we shall postulate that the dry crushing is taken to the stage where the fibre m of the last dry-crushing bagasse approaches the limit $m = 0.5$.

At this stage (Fig. 19.4), we add w of water before the following mill, which we call the first wet-crushing mill. From then on, we may assume that all the following mills reduce the bagasses to a fibre in the vicinity of m . They give therefore f/m of bagasse and:

$$\frac{f}{m} + w - \frac{f}{m} = w$$

of juice.

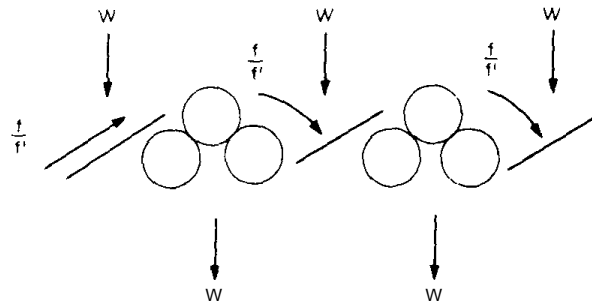


Fig. 19.4. Wet crushing.

We have then:

$$\left\{ \begin{array}{l} \text{Juice entering} = \frac{f}{m} - f + w \\ \text{Juice extracted} = w \end{array} \right.$$

The proportion of juice extracted is therefore:

$$r = \frac{w}{\frac{f}{m} - f + w} = \frac{wm}{f(1-m) + wm} \quad (19.5)$$

In practice, the proportion of sugar extracted is different:

(1) Imbibition is commenced well before the dry crushing has brought the bagasse to the fibre limit. In fact, imbibition is applied as early as the delivery from the first mill. It follows that the first wet-crushing mills give a quantity of juice greater than w . However, calculation will show that the final extraction is hardly superior to what it would have been if the fibre limit had been attained as early as the first mill.

(2) The juice extracted is always slightly higher in purity than the residual juice.

(3) On the other hand, the water (or the imbibition juice in the case of compound imbibition) mixes only in a proportion β with the juice contained in the bagasse to which it is applied. There is a proportion of this juice which is not reached nor diluted by the water, by reason of unopened cells, and of the too great affinity of bagasse for water, which causes the top layer to take the major part of the water applied; this is more readily extracted by the following mill than the juice contained in the bagasse. It follows that, of the quantity of sugar which it receives, this mill recovers only a proportion:

$$\beta r < r$$

We shall use the coefficient β to include the favourable terms (1) and (2), as well as the unfavourable term (3), which moreover has a more marked effect than the others.

Simplification of r. Let

$$w = \lambda f \tag{19.6}$$

w = quantity of imbibition water added per unit weight of cane

f = fibre content per unit of cane

X = ratio of weight of imbibition water to weight of fibre = w/f .

This term X is very useful: it is X and not w which characterises the imbibition. In fact, eqn. (19.5) may be written:

$$r = \frac{\lambda m}{1 - m + \lambda m} \tag{19.7}$$

which shows that the extraction depends on X , and not on w ; r does not alter when w and f vary in proportion.

Since m is always in the vicinity of 0.5, we have:

$$r \approx \frac{\lambda}{X + 1} \tag{19.8}$$

(c) Simple multiple imbibition

We have seen that simple imbibition means imbibition carried out by means of water only, without any return of juice to the bagasse.

Simple imbibition is called multiple when the water is applied at 2 or more points before 2 or more mills.

If there are n wet-crushing mills, it is customary to distribute equally over the n mills the quantity w of water used.

Each mill then receives w/n , and the proportion of juice extracted by each mill is obtained by replacing w by w/n in the expression for r :

$$r = \frac{\lambda m}{n(1 - m) + \lambda m} \tag{19.9}$$

and, for $m \approx 0.5$:

$$r = \frac{\lambda}{X + n} \tag{19.10}$$

For unit quantity of sugar arriving at the first wet crushing, this mill recovers βr . There remains $1 - \beta r$. If w/n of water is again applied and the bagasse is again compressed, this second operation will again furnish f/m of bagasse, and w/n of juice, and will recover $\beta' r'$ of the sugar presented to it.

But $r' = r$, since the quantity of water is the same, w/n , and since $m' = m$ by hypothesis. Similarly, we shall postulate $\beta' = \beta$, since, while the remaining juice must have become more

difficult of access, the bagasse, to compensate for this, has been better prepared by an additional pressure.

Actually, the successive values of θ should decrease along the train, since, as the bagasse becomes exhausted, the juice which remains after each pressure must be more and more enclosed in the undamaged cells, and the finer subdivision of the bagasse can compensate only slightly for this disadvantage.

Let us assume, however, that $\beta = \beta' = \beta'' = \dots$, and adopt a mean value of β . The 1st pressure will leave $1 - \beta r$ of sugar. Of this, the 2nd will recover:

$$(1 - \beta r) \beta r$$

The first 2 pressures together will therefore furnish:

$$\beta r + (1 - \beta r) \beta r$$

There remains:

$$1 - [\beta r + (1 - \beta r) \beta r] = (1 - \beta r)^2$$

Of this, a 3rd wet crushing will recover:

$$(1 - \beta r)^2 \beta r$$

Hence these first 3 pressures together will give:

$$\beta r + (1 - \beta r) \beta r + (1 - \beta r)^2 \beta r$$

and so on. After n wet crushings, there has been extracted:

$$\beta r + (1 - \beta r) \beta r + (1 - \beta r)^2 \beta r + \dots + (1 - \beta r)^{n-1} \beta r$$

a geometrical progression the sum of which is:

$$1 - (1 - \beta r)^n$$

The total extraction of the tandem, dry crushing plus wet crushing, is therefore:

$$\boxed{\frac{a(m-f)}{m(1-f)} + \left[1 - \frac{a(m-f)}{m(1-f)}\right] [1 - (1 - \beta r)^n]} \quad (19.11)$$

where:

$$r = \frac{\lambda}{\lambda + n} \quad \times = \frac{w}{f}$$

m = fibre content of bagasse from the last dry-crushing mill
 n = number of imbibitions = number of wet-crushing mills
 w = total weight of imbibition water per unit weight of cane
 f = weight of fibre per unit weight of cane

a is of the order of 1.05 – 1.10
 β is of the order of 0.60 – 0.80 for water.

(d) Compound multiple imbibition - theoretical study

Compound imbibition is the system which consists of applying water before the last mill, of returning the juice obtained at the last mill as imbibition before the second-last mill, the juice from the latter mill before the next one preceding, and so on. The juice from the 2nd wet-crushing mill is thus returned before the 1st wet-crushing mill, and only the juice from this first wet-crushing mill goes to manufacture with the dry-crushing juice.

In Fig. 19.5 let e_1, e_2, e_3, \dots be the corresponding quantities of sugar extracted by the 1st, 2nd, 3rd, ... wet-crushing mills, for unit quantity of sugar remaining after dry crushing. If each of the latter gives a bagasse of fibre m , each of them will recover w of juice and will extract a proportion r of sugar.

As a first approximation, we shall disregard the coefficient β and calculate the total extraction of sugar obtained in this way.

At the 1st wet-crushing mill, there arrives a quantity of sugar:

$$A_1 = 1 + e_2$$

The 1st wet-crushing mill extracts of this:

$$e_1 = rA_1 = r(1 + e_2)$$

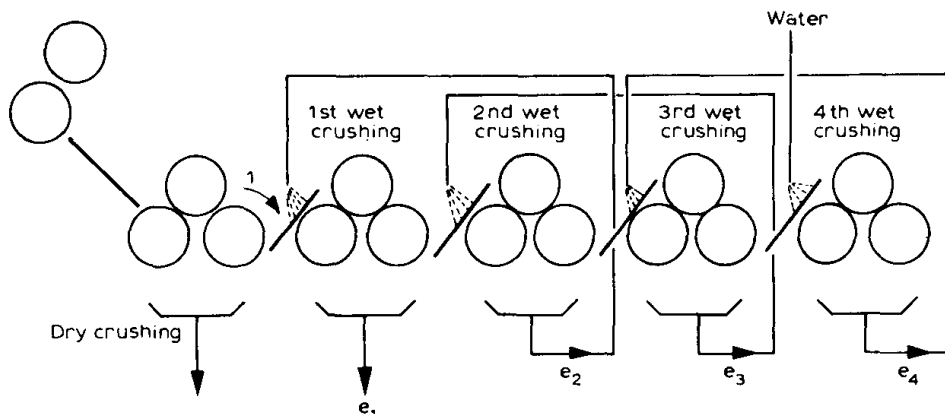


Fig. 19.5. Multiple compound imbibition.

There remains in the bagasse:

$$B_1 = A_1 (1 - r) = (1 + e_2)(1 - r)$$

At the 2nd mill there arrives a quantity of sugar:

$$A_2 = e_3 + B_1 = e_3 + (1 + e_2)(1 - r)$$

The 2nd mill extracts from this:

$$e_2 = rA_2 = re_3 + r(1 + e_2)(1 - r) = re_3 + e_2(1 - r)$$

There remains in the bagasse:

$$B_2 = A_2 (1 - r) = e_3(1 - r) + (1 + e_2)(1 - r)^2$$

The quantity of sugar arriving at the 3rd mill is:

$$A_3 = e_4 + B_2 = e_4 + e_3(1 - r) + (1 + e_2)(1 - r)^2$$

The 3rd mill extracts from it:

$$e_3 = rA_3 = re_4 + re_3(1 - r) + r(1 + e_2)(1 - r)^2 = re_4 + e_3(1 - r)$$

Similarly for the following mills, there arrives at the $(n - 1)$ th mill a quantity of sugar:

$$A_{n-1} = e_n + e_{n-1}(1 - r) + e_{n-2}(1 - r)^2 + \dots + e_3(1 - r)^{n-3} + (1 + e_2)(1 - r)^{n-2}$$

The $(n - 1)$ th mill extracts from this:

$$e_{n-1} = re_n + (1 - r)e_{n-2}$$

There remains in the bagasse:

$$B_{n-1} = e_n(1 - r) + e_{n-1}(1 - r)^2 + \dots + e_3(1 - r)^{n-2} + (1 + e_2)(1 - r)^{n-1}$$

There arrives at the last mill a quantity of sugar:

$$A_n = 0 \text{ (water)} + e_n(1 - r) + e_{n-1}(1 - r)^2 + \dots + e_3(1 - r)^{n-2} + (1 + e_2)(1 - r)^{n-1}$$

The last mill extracts from this:

$$e_n = e_{n-1}(1 - r)$$

If we set down the various values found for the successive extractions e , and if we add the equations so obtained, a large number of terms cancel out on one side or the other, and we have:

$$\begin{aligned}
 e_n &= e_{n-1} - re_{n-1} \\
 e_{n-1} &= re_n + e_{n-2} - re_{n-2} \\
 e_{n-2} &= re_{n-1} + e_{n-3} - re_{n-3} \\
 \dots &= \dots + \dots - \dots \\
 e_3 &= re_4 + e_2 - re_2 \\
 e_2 &= re_3 + e_1 - re_1 \\
 e_1 &= re_2 + r \\
 \hline
 e_n &= re_n + r - re_1
 \end{aligned}
 \tag{19.12}$$

whence:

$$e_n = \frac{r}{1 - r} (1 - e_1)$$

A similar addition may also be made omitting the first of the equations (19.12). We then obtain:

$$e_{n-1} = re_n + re_{n-1} + r(1 - e_1)$$

whence:

$$e_{n-1} = (1 - e_1) \left[\frac{r}{1 - r} + \frac{r^2}{(1 - r)^2} \right]$$

Omitting similarly 2, 3, ... equations, we have successively:

$$\begin{aligned}
 e_{n-2} &= (1 - e_1) \left[\frac{r}{1 - r} + \left(\frac{r}{1 - r} \right)^2 + \left(\frac{r}{1 - r} \right)^3 \right] \\
 \dots & \\
 e_2 &= (1 - e_1) \left[\frac{r}{1 - r} + \left(\frac{r}{1 - r} \right)^2 + \dots + \left(\frac{r}{1 - r} \right)^{n-1} \right]
 \end{aligned}$$

But:

$$e_1 = r(1 + e_2) \quad e_2 = \frac{e_1 - r}{r}$$

We have then:

$$\frac{e_1 - r}{r} = (1 - e_1) \left[\frac{r}{1 - r} + \left(\frac{r}{1 - r} \right)^2 + \dots + \left(\frac{r}{1 - r} \right)^{n-1} \right]$$

Whence:

$$e_1 = r \cdot \frac{1 + \left[\frac{r}{1 - r} + \left(\frac{r}{1 - r} \right)^2 + \dots + \left(\frac{r}{1 - r} \right)^{n-1} \right]}{1 + r \left[\frac{r}{1 - r} + \left(\frac{r}{1 - r} \right)^2 + \dots + \left(\frac{r}{1 - r} \right)^{n-1} \right]}$$

But:

$$r = \frac{\lambda}{\lambda + 1}$$

Hence:

$$\lambda = \frac{r}{1 - r} \quad (19.13)$$

Each of the expressions in square brackets in the expression for e_1 above is a geometrical progression, the sum of which is:

$$\lambda \cdot \frac{\lambda^{n-1} - 1}{\lambda - 1}$$

Hence:

$$e_1 = \frac{\lambda}{\lambda + 1} \cdot \frac{1 + \lambda \cdot \frac{\lambda^{n-1} - 1}{\lambda - 1}}{1 + \frac{\lambda}{\lambda + 1} \cdot \lambda \cdot \frac{\lambda^{n-1} - 1}{\lambda - 1}}$$

or:

$$e_1 = \lambda \cdot \frac{\lambda^n - 1}{\lambda^{n+1} - 1} \quad (19.14)$$

This expression postulates $\lambda \neq 1$. For $\lambda = 1$, we find readily, by reference to the formula in r above:

$$e_1 = \frac{n}{n + 1} \quad (X = 1)$$

Hence the total extraction for the tandem, dry crushing plus wet crushing, is:

$$\frac{\alpha (m - f)}{m (1 - f)} + \left[1 - \frac{\alpha (m - f)}{m (1 - f)} \right] \lambda \cdot \frac{\lambda^n - 1}{\lambda^{n+1} - 1} \quad (19.15)$$

where:

$$\lambda = \frac{w}{f}$$

Practical results

The same 3 causes already discussed on p. 283 intervene in compound as in simple imbibition to modify the results. If results obtained in practice are compared with those forecast by the foregoing calculation, it is found that the difference obtained may be expressed in terms of an efficiency coefficient which we shall define as follows:

$$\beta = \frac{\text{sugar actually extracted by the imbibition concerned}}{\text{sugar which would have been extracted according to the preceding theoretical formulae}}$$

This coefficient β is unfortunately variable, and depends on the efficiency of the system of imbibition used, on the condition of the rollers, the performance of the mills, the state of preparation of the bagasse, and, in compound as in simple imbibition, on the place of the mill in the tandem. However, it would not be greatly in error to adopt a coefficient of the order of 0.6-0.8.

Further, imbibition with water under pressure, as carried out at the last mill with compound imbibition, and at all mills with simple imbibition, has an efficiency slightly higher than that of imbibition with juice flowing onto the bagasse by gravity.

Similarly, water applied under pressure penetrates the bagasse better than when it is in the form of a fine spray: a distributor in the form of a perforated pipe, with holes at intervals of 25 — 40 mm (1 — 1½ in.), is more effective than a distributor with atomisers giving a spray in the form of a cone which spreads the water more uniformly over the layer of bagasse. The reason is that, in the second case, only the top layer of bagasse is wetted, whereas in the former case, the jets of water penetrate further into the interior of the bagasse and the water is distributed more uniformly as regards depth.

We may comment in this connection that MittaP, in India, has found a better efficiency with water under a pressure of 0.5 kg/cm² (7.5 p.s.i.) than under 1.0 kg/cm² (15 p.s.i.). It is actually advantageous to apply imbibition under a head of 5-6 m (15 — 20 ft.).

As a first approximation, we may assume:

For water under pressure: $\theta = 0.60-0.80$
 For return of juice: $\beta = 0.50-0.70$

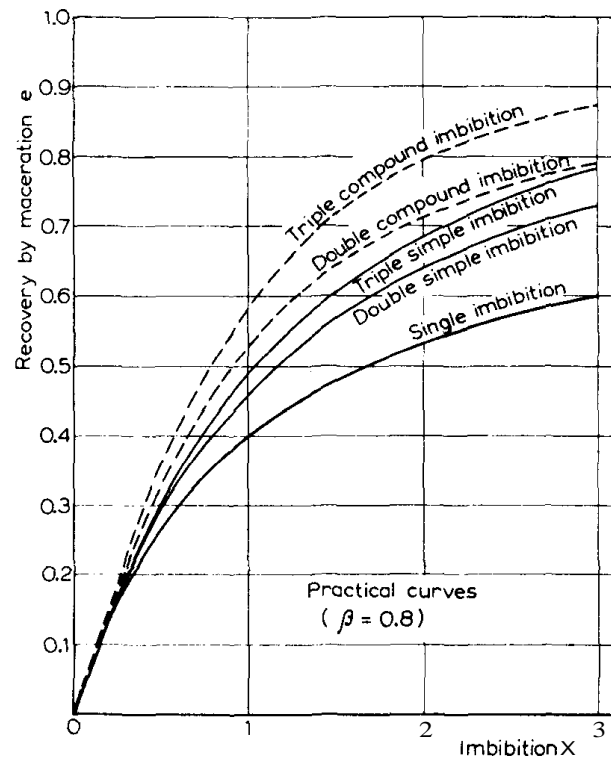
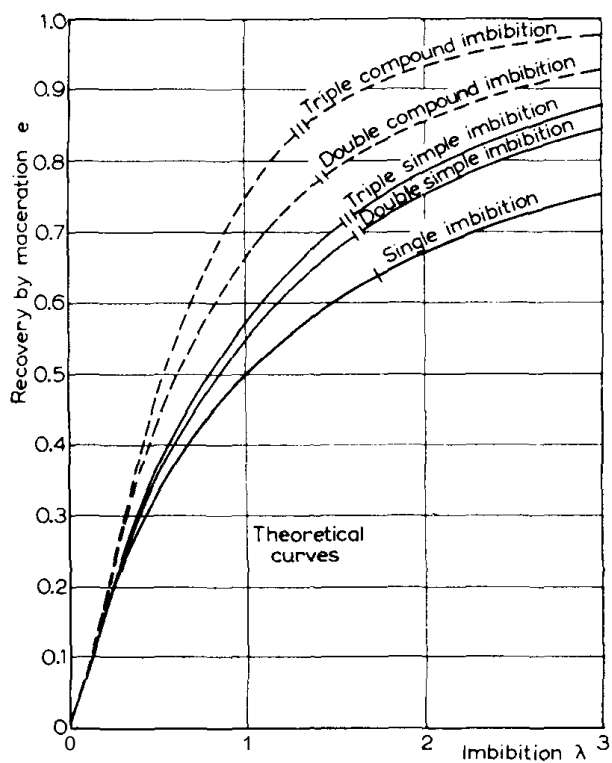


Fig. 19.6. Variation of residual extraction as a function of imbibition. Theoretical graph.

Fig. 19.7. Variation of residual extraction as a function of imbibition. Practical graph ($\beta = 0.8$).

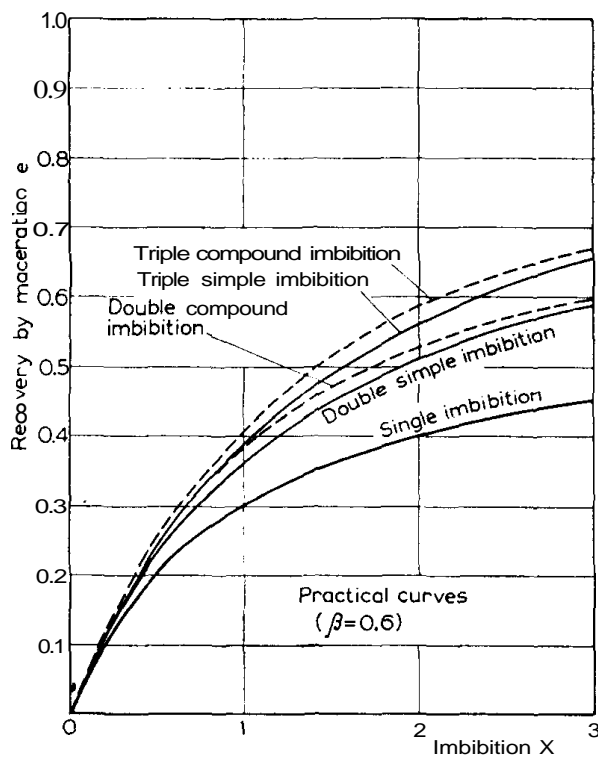


Fig. 19.8. Variation of residual extraction as a function of imbibition. Practical graph ($\beta = 0.6$).

However, to avoid complication, it will suffice in practice to adopt a single value of β , easy to calculate in each case, and which we shall take equal to 0.60 or 0.65, in the absence of more precise figures.

Case of compound imbibition. To take into account the coefficient β , it will be necessary therefore, in our formula (19.15), to replace X by:

$$\lambda' = \frac{\beta r}{1 - \beta r} \tag{19.16}$$

We show now, for the 2 methods of imbibition:

(a) Theoretical graphs obtained by the original formulae (Fig. 19.6).

(b) The same curves corrected to allow for: (1) a coefficient $\beta = 0.8$ (Fig. 19.7); (2) a coefficient $\beta = 0.6$ (Fig. 19.8).

It is seen immediately that the introduction of the coefficient β is much more disadvantageous in compound than in simple imbibition. Whereas the theoretical curves which express the effect of simple imbibition are far below the corresponding curves for compound imbibition, the relative distance of the corrected curves is much less, and if a very low value were adopted for β , these curves would end up by being approximately superimposed (or strictly speaking, by intersecting).

Conclusion. (1) The difference in value presented by the 2 modes of imbibition is much less important than the theoretical formulae would lead one to suppose.

(2) The efficacy of the system of imbibition is of greater importance in compound than in simple imbibition.

Comment. Our hypotheses (1) of a bagasse fibre of 0.5 in all cases, and (2) of a single and constant coefficient θ , are approximations only; but they are sufficient to show at what point a mediocre efficiency affects the results of compound imbibition. It is this point especially that we wish to emphasise.

Brix and sugar content of juices with compound imbibition

We have calculated the extraction with compound imbibition by taking as unity the quantity of sugar remaining after dry crushing, and calculating the successive quantities $e_1, e_2, e_3, \dots, e_n$ of sugar extracted by the successive wet-crushing mills.

This calculation assumed that the quantities of sugar extracted by the various mills were proportional to the quantities of juice extracted by them (r); in other words, that the imbibition juice mixed completely with the juice previously contained in the bagasse which received it.

Let us retain these assumptions, and consider now the sugar content of these different juices. Let:

S_0 = pol content of the juice remaining after dry crushing

$S_1, S_2, S_3, \dots, S_n$ = pol contents of the juice furnished by the 1st, 2nd, 3rd, ..., n th wet-crushing mills.

Similarly, let us consider the relative pol contents, as proportions of the pol content S_0 of the juice remaining after dry crushing:

$$s_1 = \frac{S_1}{S_0} \quad s_2 = \frac{S_2}{S_0} \quad s_3 = \frac{S_3}{S_0} \quad \dots \quad s_n = \frac{S_n}{S_0}$$

Finally, all expressed per unit weight of cane, let:

Q_0 = weight of pol remaining in the bagasse after dry crushing

$Q_1, Q_2, Q_3, \dots, Q_n$ = weights of pol extracted by the 1st, 2nd, 3rd, ..., n th wet-crushing mills

j = weight of juice present in each bagasse

w = weight of imbibition juice or water applied at each mill.

Note that:

$$j \approx f$$

Since:

$$\frac{w}{j+w} = r \quad \frac{j}{j+w} = 1-r \quad \frac{w}{j} = \frac{r}{1-r} = x = \frac{w}{j}$$

we have:

$$\begin{aligned} Q_0 &= jS_0 & \frac{Q_0}{Q_0} &= 1 \\ Q_1 &= wS_1 & \frac{Q_1}{Q_0} - e_1 &= \frac{wS_1}{jS_0} = \lambda s_1 \\ Q_2 &= wS_2 & \frac{Q_2}{Q_0} - e_2 &= \frac{wS_2}{jS_0} = \lambda s_2 \\ &\dots & & \\ Q_n &= wS_n & \frac{Q_n}{Q_0} = e_n &= \frac{wS_n}{jS_0} = \lambda s_n \end{aligned}$$

Hence we may derive (c/ p. 288):

$$\frac{S_{p+1}}{S_p} = \frac{S_{p+1}}{S_p} = \frac{e_{p+1}}{e_p} = \frac{(1 - e_1) (\lambda + \lambda^2 + \lambda^3 + \dots + \lambda^{n-p})}{(1 - e_1) (\lambda + \lambda^2 + \lambda^3 + \dots + \lambda^{n-p+1})}$$

or:

$$\frac{S_{p+1}}{S_p} = \frac{\lambda \cdot \frac{\lambda^{n-p} - 1}{\lambda - 1}}{\lambda \cdot \frac{\lambda^{n-p+1} - 1}{\lambda - 1}} = \frac{\lambda^{n-p} - 1}{\lambda^{n-p+1} - 1} \tag{19.17}$$

We have thus obtained a recurrence formula. This equation is valid only for $1 \leq p \leq n$, since:

$$\frac{S_2}{S_1} = \frac{e_2}{e_1} \quad \text{but} \quad \frac{S_1}{S_0} = \frac{e_1}{?}$$

We shall, however, verify that it is valid for S_1/S_0 , that is, for $p = 0$.

We have in fact, assuming that the mixture of juice is complete after imbibition, and that the juice extracted by the mill and that remaining in the bagasse consequently have the same sugar content:

$$S_1 (w + j) = S_0 j + S_2 w$$

$$S_1 = r S_2 + (1 - r) S_0$$

$$\frac{S_1}{S_0} = r \frac{S_2}{S_0} + (1 - r) = r \cdot \frac{S_2}{S_1} \cdot \frac{S_1}{S_0} + (1 - r)$$

$$\frac{S_1}{S_0} = r \cdot \frac{\left(\frac{r}{1-r}\right)^{n-1} - 1}{\left(\frac{r}{1-r}\right)^n - 1} \cdot \frac{S_1}{S_0} + (1 - r)$$

Hence:

$$\frac{S_1}{S_0} = \frac{1 - r}{1 - r \cdot \frac{\left(\frac{r}{1-r}\right)^{n-1} - 1}{\left(\frac{r}{1-r}\right)^n - 1}} = \frac{\left(\frac{r}{1-r}\right)^n - 1}{\left(\frac{r}{1-r}\right)^n \left(\frac{1}{1-r} - 1\right) - 1}$$

or:

$$\frac{S_1}{S_0} = \frac{X^n - 1}{\lambda^{n+1} - 1}$$

Q.E.D.

We have then:

$$s_1 = \frac{X^n - 1}{\lambda^{n+1} - 1}$$

$$\begin{aligned}
 s_2 &= s_1 \cdot \frac{s_2}{s_1} = \frac{X^n - 1}{\lambda^{n+1} - 1} \cdot \frac{\lambda^{n-1} - 1}{X^n - 1} = \frac{\lambda^{n-1} - 1}{X^{n+1} - 1} \\
 s_3 &= s_2 \cdot \frac{s_3}{s_2} = \frac{\lambda^{n-1} - 1}{\lambda^{n+1} - 1} \cdot \frac{\lambda^{n-2} - 1}{X^n - 1} = \frac{X^{n-2} - 1}{X^{n+1} - 1} \\
 &\dots \dots \dots \\
 s_p &= s_{p-1} \cdot \frac{S_p}{S_{p-1}} = \frac{\lambda^{n-p+2} - 1}{\lambda^{n+1} - 1} \cdot \frac{\lambda^{n-p+1} - 1}{X^{n-p+2} - 1} = \frac{\lambda^{n-p+1} - 1}{X^{n+1} - 1}
 \end{aligned}$$

We may therefore calculate the pol content of any imbibition juice, knowing simply S_0 and X . We have:

$$\boxed{S_p = S_0 \cdot \frac{\lambda^{n-p+1} - 1}{\lambda^{n+1} - 1}} \tag{19.18}$$

- n = number of wet-crushing mills
- p = number of the wet-crushing mill concerned (*i.e.* its position in the sequence of wet-crushing mills)
- S_p = pol content of the juice of this p th wet-crushing mill
- S_0 = pol content of dry-crushing juice = pol content of normal juice
- $X = w/f$ = weight of imbibition water per unit weight of fibre.

Note. We have just seen above:

$$e, = \backslash s, \quad s_1 = \frac{X^n - 1}{\lambda^{n+1} - 1}$$

from which we obtain:

$$e, = X \cdot \frac{X^n - 1}{\lambda^{n+1} - 1}$$

which provides a second derivation of this important and interesting equation.

Brix. The relationships which we have just shown for the sugar contents could equally well have been derived for the brixes, if we accept the hypothesis that the purities of the successive juice were the same. Actually, they decrease from mill to mill, but this scarcely more than compensates for the approximations due to our hypotheses, and we may equally well write:

$$B_p = B_0 \cdot \frac{\lambda^{n-p+1} - 1}{\lambda^{n+1} - 1} \quad (19.19)$$

B_p = brix of juice from p th wet-crushing mill

B_0 = brix of the dry-crushing juice = brix of normal juice

n = number of wet-crushing mills

p = position of the wet-crushing mill concerned, in the wet-crushing mills

A = weight of imbibition water per unit weight of fibre = w/f .

Practical conclusions

The results of the preceding theoretical discussion, and of experience, lead to the following practical conclusions:

(a) For equal quantity of water used, compound imbibition is superior to simple imbibition.

(b) Compound imbibition leads to a much greater return of juice to the bagasse: twice as much for double compound, thrice as much for triple, for the same quantity of water. For this reason, when the limiting factor is, not evaporator capacity, but choking at the mills, crushing rate is restricted much sooner with compound than with simple imbibition. In this case, one may resort to a compromise between the 2 procedures; for example, in the case of 17 rollers:

Imbibition with water before the 5th or last mill

Imbibition with water before the 4th mill

Return of 5th-mill juice before the 3rd mill

Return of 4th-mill juice before the 2nd mill

Transfer to manufacture of the juices from crusher and 1st mill (primary juice)

Transfer to manufacture of the juices from 2nd and 3rd mills (secondary juice).

For average quantities of imbibition, it is sometimes sufficient to divide the water between the 4th and 5th mills, which are the ones most susceptible to choking, to return their combined juices before the 3rd, the juice of the 3rd before the 2nd, and the juice of the 1st mill (primary juice) and of the 2nd (secondary juice) to manufacture.

If, on the other hand, it is desired or is found necessary to balance the quantities of imbibition between the various mills, as in the case of our first example above, we may comment that it is then necessary to send to manufacture the juice not only of the first 2 mills, but of the first 3 mills.

We add (Fig. 19.9(b)) the simple schemes proposed by B.M.A. for tandems of 4, 5 and 6 mills, for fibre of 12.5, imbibition 20-25, per 100 of cane.

RELATIVE WEIGHTS OF BAGASSE AND JUICE

The relative weights of bagasse and juice from the various mills in a tandem are very readily determined. We give as an example a tandem of 5 mills and consider the case of compound

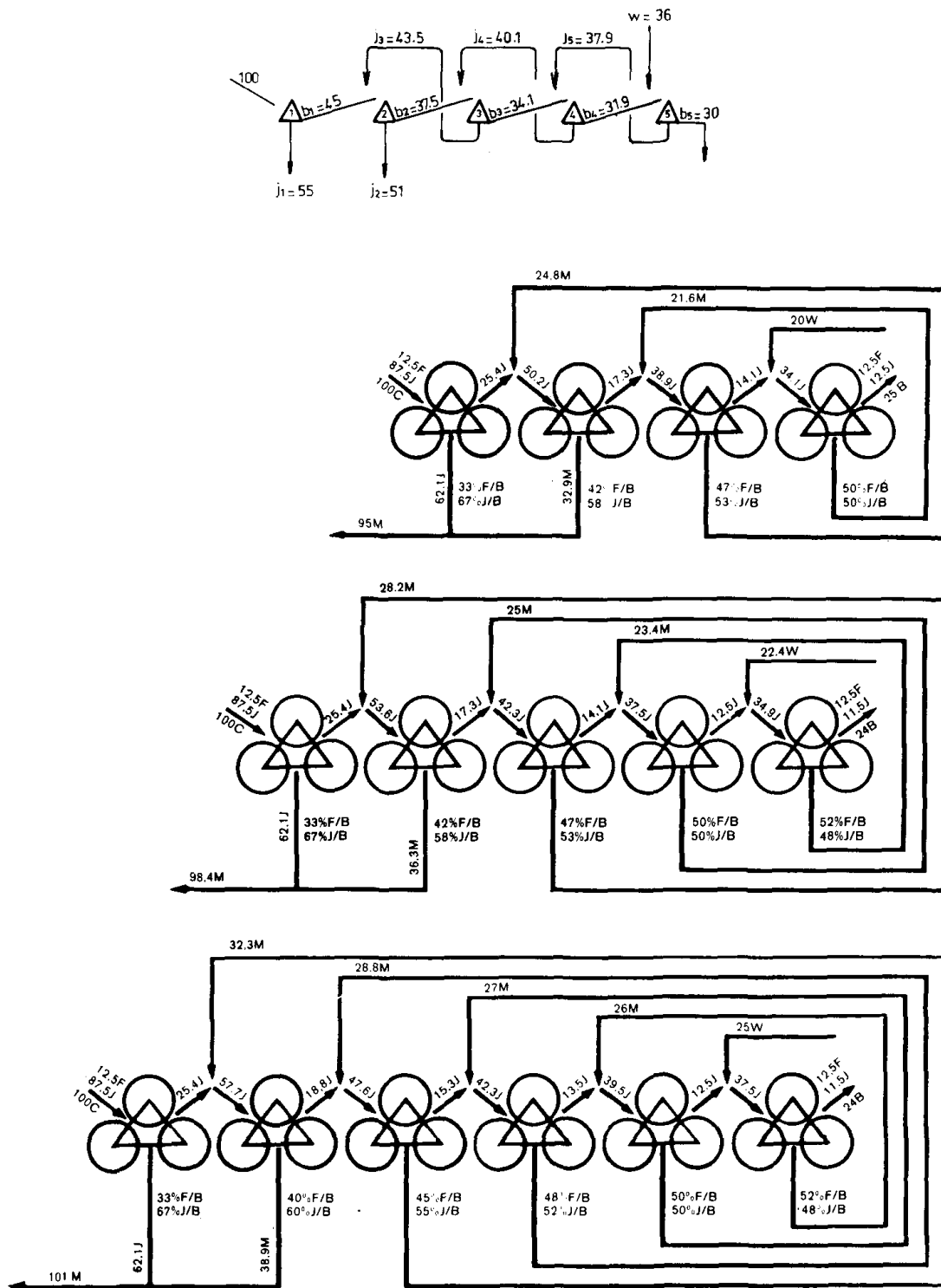


Fig. 19.9. (a) Juices and bagasses in compound imbibition ($f = 0.15$; $w = 0.36$). (b) Diagram of juice and fibre in 4-, 5- and 6-mill tandems ($f = 0.125$; $w = 0.20-0.25$) (B.M.A.)

imbibition, assuming that, as is the general case, the intermediate juices are returned by chokeless pumps without separation of fine bagasse. The calculation may be easily modified for any other case.

Notation. We assume that unit weight of cane arrives at the first mill, with fibre content $f = 0.15$ and imbibition $w = 0.360$:

j = weight of juice per unit of cane

b = weight of bagasse per unit of cane

F = fibre content of bagasse

Q = weight of mixed juice per unit of cane

1, 2, 3, . . . n = indices of successive mills.

The basic equation gives:

$$1 + w = Q + b_n$$

Hence:

$$Q = 1 + w - b_n = 1 + 0.360 - 0.300 = 1.060 \quad (\text{Fig. 19.9a})$$

Bagasses. We assume operation with the following scale of fibre figures:

$F_1 = 0.33$	$b_1 = 0.15/0.33 = 0.450$
$F_2 = 0.40$	$b_2 = 0.15/0.40 = 0.375$
$F_3 = 0.44$	$b_3 = 0.15/0.44 = 0.341$
$F_4 = 0.47$	$b_4 = 0.15/0.47 = 0.319$
$F_5 = 0.50$	$b_5 = 0.15/0.50 = 0.300$

Juice. Equating the quantities entering and leaving a group of mills, we readily have:

$j_1 = 1 - b_1 = 0.550$	$j_2 = Q - j_1 = 0.510$
$j_2 = b_1 + j_3 - b_2$	$j_3 = j_2 - b_1 + b_2 = 0.435$
$j_3 = b_2 + j_4 - b_3$	$j_4 = j_3 - b_2 + b_3 = 0.401$
$j_4 = b_3 + j_5 - b_4$	$j_5 = j_4 - b_3 + b_4 = 0.379$
$j_5 = b_4 + w - b_5$	

For intermediate mills:

$$j_i = b_{i-1} + (w - b_n)$$

The material balance is calculated starting with the minimal necessary data as follows, for which we apply by way of example typical values:

f = fibre per unit of cane:	0.15
w = imbibition per unit of cane:	0.36
B_1 = brix of primary juice:	20.73
p_1 = purity of first-mill juice:	89
e = extraction obtained:	96
p_n = purity of residual juice of final bagasse:	50

With brix of juices returned:

$$B_3 = 6.50 \quad B_4 = 4 \quad B_5 = 3$$

and purities:

$$p_3 = 80 \quad p_4 = 77 \quad p_5 = 72$$

The calculation is carried out in the following order:

Cane. Brix B_0 of absolute juice is deduced from brix of primary juice assuming:

$$B_0 = 0.965 B_1 = 0.965 \times 20.73 = 20$$

Purity of absolute juice is similarly:

$$p_0 = 0.95, \text{ or: } 0.96 p_1 = 0.85$$

Hence sucrose content:

$$S_0 = (1 - f) B_0 p_0 = 0.850 \times 20 \times 0.85 = 14.44$$

Juice of first mill. Soluble matter:

$$M_1 = j_1 B_1 = 0.550 \times 20.73 = 11.40$$

Hence sucrose % first-mill juice:

$$S_1 = B_1 p_1 = 20.73 \times 0.89 = 18.45$$

and sucrose in this juice:

$$s_1 = 11.40 \times 0.89 = 10.15$$

Final bagasse. We have sucrose in final bagasse:

$$s_{..} = s_0 (1 - e) = 14.44 \times 0.04 = 0.58$$

Hence soluble matter in the bagasse:

$$M_n = s_n/p_n = 0.0058/0.50 = 0.0116$$

Juice in bagasse:

$$g_n = (1 - F_n) b_n = 0.50 \times 0.300 = 0.150$$

and sucrose % in bagasse:

$$s_n/b_n = 0.0058/0.300 = 1.93\%$$

Second-milljuice. Soluble material:

$$M_2 = M_0 - M_1 - M_n = 0.0444$$

Hence brix of second-mill juice:

$$B_2 = 0.0444/0.510 = 8.71$$

and sucrose:

$$S_2 = 0.1445 - 0.1015 - 0.0058 = 0.0372$$

Sucrose % of juice:

$$s_2 = 0.0372/0.510 = 7.29$$

Purity:

$$p_2 = 0.0372/0.0444 = 83.8$$

Residual juice from first mill. Soluble material:

$$M = 0.1700 - 0.1140 = 0.0560$$

Sucrose:

$$s = 0.1445 - 0.1015 = 0.0430$$

Juice:

$$g = b_1 - / = 0.450 - 0.150 = 0.300$$

Sucrose % juice:

$$S = 0.0430/0.300 = 14.33$$

Purity:

$$p = 0.0430/0.0560 = 76.8$$

Third-mill juice. Soluble material:

$$M_3 = 0.435 \times 0.0650 = 0.0283$$

Sucrose:

$$s_3 = 0.0283 \times 0.80 = 0.0226$$

Residual juice from first mill after imbibition. Total juice:

$$g = \text{£}1 + j_3 = 0.300 + 0.435 = 0.735$$

Soluble material:

$$M = 0.0560 + M_3 = 0.0843$$

Sucrose:

$$s = 0.0430 + 0.0226 = 0.0656$$

Residual juice from second mill. This is the residual juice of the first mill less what leaves with the second-mill juice:

Soluble material:

$$M = 0.0843 - M_2 = 0.0843 - 0.0444 = 0.0399$$

Sucrose:

$$s = 0.0656 - 0.0372 = 0.0284$$

Continuing step by step in the same way: residual juice from second mill after imbibition, residual juice from third mill = residual juice from second mill after imbibition less juice which leaves as third-mill juice, residual juice from fourth mill, and so on: we thus find the quantities indicated in Fig. 19.10.

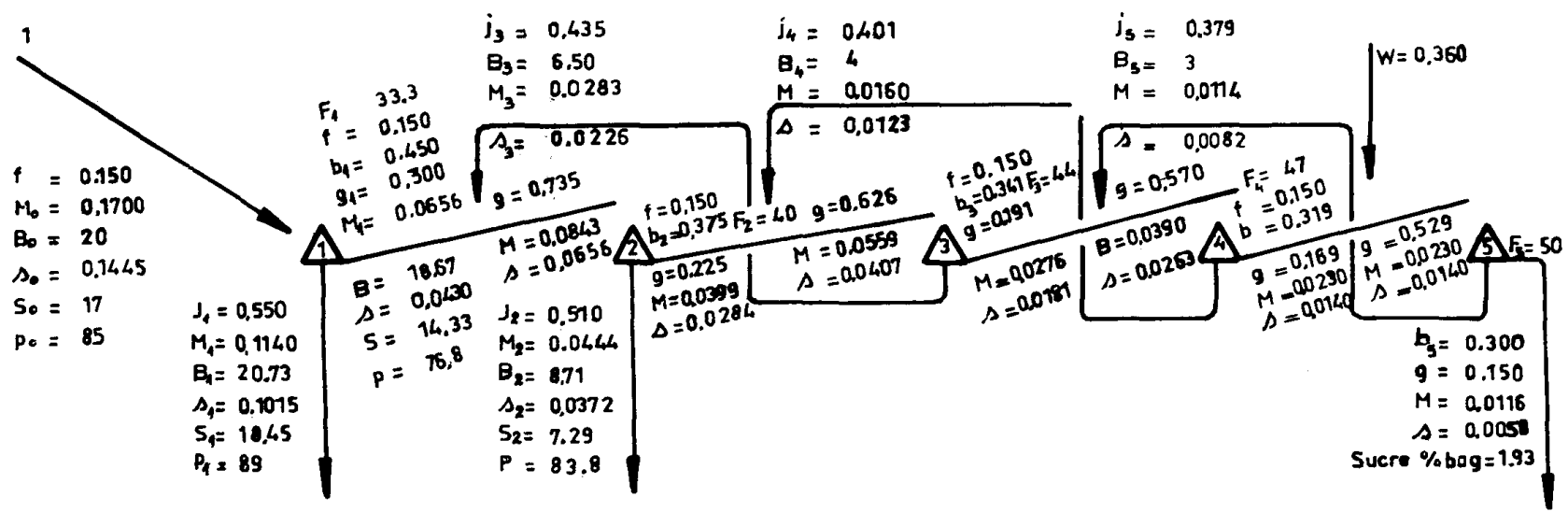


Fig. 19.10. Weight of juices and bagasses in the tandem.

GENERAL CONSIDERATIONS

Optimum imbibition

The extraction increases, rapidly at first, more slowly later, as imbibition is increased. Steam consumption increases with imbibition. If the factory is consuming purchased fuel, or if it is using its bagasse profitably, there will obviously be a quantity of water which corresponds to the maximum benefit. This quantity depends on the ratio of price of sugar to price of fuel, and should be calculated for each individual case.

The form of the graphs in Figs. 19.6 ~ 19.8 shows that generally the optimum imbibition will be reached at values of X of the order of 2.

The extraction in fact increases very rapidly as X increases from zero to 1, still fairly rapidly between 1 and 2, but more slowly above 2. There is rarely any advantage in exceeding $X = 3$, and it is normally never advantageous to exceed $X = 4$.

It is of interest, however, to note that, in the course of tests carried out at the experimental factory at Audubon Park, in Louisiana², it was found that extraction increased rapidly with imbibition up to the neighbourhood of 30% ($X = 2 - 2.5$), and attained a maximum for an imbibition of 60% ($X = 4-5$).

In Australia³ it has been found, for 3 factories studied, that the optimal value was situated at $X = 2.4$. One author⁴ reports, however, a gain up to $X = 3.2$. This would suggest that, above the economic optimum, there would exist also a technical optimum and that one cannot accept a graph of recovery by imbibition increasing indefinitely. In this case a tentative explanation could be found in the fact that the capacity of bagasse for absorbing water is limited to about seven times the weight of fibre it contains (*cf.* p. 309); there would be no advantage in going beyond that figure.

Theoretical experiments (simulations) in Australia⁵ show that there would be a maximum for $X = 3$ in a tandem of 5 mills, or for a value of 2.5 in a 6- or 7-mill tandem, and that the effect of quantity of imbibition decreases as the number of mills increases.

More recent experiments, however, also in Australia, by Crawford⁶, would tend to indicate that, if the feeding of the mill is not affected, there would be no limit to the gain in extraction obtained by increasing imbibition; this gain would become smaller and smaller, but never zero. The difference should not be difficult to determine, but its interest is purely theoretical.

The highest imbibition quantities recorded are those practised in South Africa. The mean figure for imbibition % fibre for 1980 - 81 was 344, one factory reporting 400. In that country, there were 11 diffusion plants and 11 tandems of mills. The average extraction was 96.89, with scarcely any difference between diffusion factories and factories with mills. However, these were nearly all 6- or 7-mill trains, and the only factory with imbibition less than 200% on fibre (173) reported an extraction equal to that of the others with imbibition figures of 300 - 401. We can however conclude that South African experience provides evidence in favour of high quantities of imbibition.

Optimum distribution of water in multiple simple imbibition

In studying multiple simple imbibition, we have assumed *a priori* that the imbibition water was divided equally between the various mills.

The question may be asked, whether it would not be better to adopt an unequal distribu-

tion among the different positions, and to seek the optimal proportions corresponding to the best extraction, for a given total quantity w of imbibition water.

For simplicity, we shall confine ourselves to the case of double simple imbibition, assuming we are dealing with the last two mills of any tandem. The results obtained will be applicable to the case of hybrid imbibition (*cf.* p. 296).

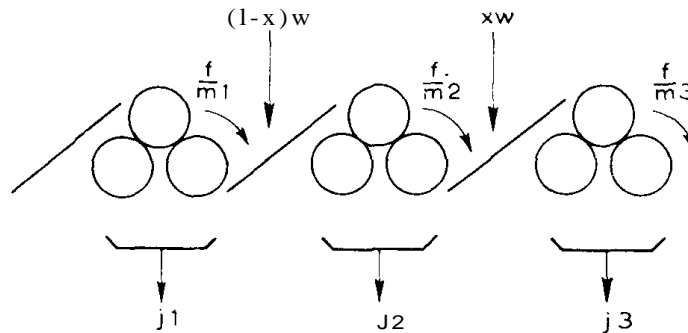


Fig. 19.11. Optimum distribution of water between two sprays.

Let (Fig. 19.11):

- w = total weight of imbibition water to be distributed, per unit cane
- xw = proportion to be applied before the last mill
- $(1 - x) w$ = proportion to be applied before the penultimate mill
- f = fibre in cane
- m_1 = fibre in bagasse from the antepenultimate mill
- m_2 = fibre in bagasse from the penultimate mill
- m_3 = fibre in bagasse from the last mill.

The penultimate mill will give a quantity of juice:

$$j_2 = \frac{f}{m_1} + (1 - X) W - \frac{f}{m_2}$$

and the last:

$$j_3 = \frac{f}{m_2} + xw - \frac{f}{m_3}$$

Let:

$$\frac{f}{m_1} - \frac{f}{m_2} = a \quad \frac{f}{m_2} - \frac{f}{m_3} = a'$$

and let us designate by b the quantity of juice remaining in the bagasse from the antepenultimate mill, before imbibition:

$$J_2 = \frac{f}{m_1} - f = b$$

The quantity of juice remaining in the bagasse from the last mill but one will be:

$$J_3 = \frac{f}{m_2} - \frac{f}{m_1} + \frac{f}{m_2} - \frac{f}{m} b - a$$

For unit quantity of sugar remaining after the antepenultimate mill, the following mill will extract a proportion of juice (or, approximately, of sugar):

$$r_2 = \frac{j_2}{J_2 + (1-x)w} = \frac{a + (1-x)w}{b + (1-x)w}$$

There remains then in its bagasse a quantity of sugar $1 - r_2$, and the last mill recovers from it a proportion:

$$r_3 = \frac{j_3}{J) + xw} = \frac{a' + xw}{b - a + xw}$$

On the total, then, we obtain:

$$e = r_2 + (1 - r_2) r_3 = \frac{a + (1-x)w}{b + (1-x)w} + \frac{[a + (1-x)w]}{[b + (1-x)w]} \frac{a' + xw}{b - a + xw}$$

We require to find the maximum value of e when x varies from 0 to 1. We have:

$$e = \frac{a + (1-x)w}{b + (1-x)w} + \frac{a' + xw}{b - a + xw} - \frac{[a + (1-x)w]}{[b + (1-x)w]} \frac{[a' + xw]}{[b - a + xw]}$$

or:

$$e = \frac{-w^2x^2 + (a+w)wx + (b-a)(a+a'+w)}{-w^2x^2 + (a+w)wx + (b-a)(b+w)}$$

Let us set down the derivative with respect to x of the function e . It has the value:

$$e' = \frac{w(a+w-2wx)(b-a)(b-a-a')}{v^2}$$

we designate the whole denominator of e by v .

For e to have a maximum value, $e' = 0$, which can occur only when:

$$a + w - 2wx = 0 \quad x = \frac{1}{2} + \frac{a}{2w}$$

and, substituting the value of a :

$$x = \frac{1}{2} + \frac{\frac{1}{m_1} - \frac{1}{m_2}}{2\lambda} \quad (19.20)$$

Comments. (7) *Common values.* In general, $1/m_1 - 1/m_2$ is of the order of 0.1 - 0.6, the smallest values being obtained in the longest tandems. For typical values of X (2 - 3), x varies then from 0.58 to 0.51. It is thus necessary always to apply more than half of the water at the last mill, but only slightly more than half and, if the last mill is liable to choke on account of excess imbibition, there will be little disadvantage in returning to a more nearly equal distribution, since the maximum of extraction is never very marked.

(2) *Limit of double simple imbibition.* When the imbibition has to be reduced to a very low quantity, a point is reached where:

$$X = 1 - \frac{1}{m_1} - \frac{1}{m_2}$$

Consequently, $x = 1$, and there is no further advantage in using double imbibition: all the water must then be applied at the last mill. However, this limiting value being very low, generally less than 0.3, this limit is not likely to apply in normal practice. It will be simply remembered that, as moreover is shown by eqn. (19.20), the proportion x to apply at the last mill should be increased as the quantity of imbibition X is decreased.

(3) *Influence of the coefficient β .* We have assumed that the last mill but one extracted r_2 and the last r_3 of the sugar received by each. Actually, we know that we shall recover at the first only βr_2 . There remains then $(1 - \beta r_2)$, of which the last mill recovers $\beta r_3 (1 - \beta r_2)$.

We should then have had to ascertain the maximum value of $\beta r_2 + \beta r_3 (1 - \beta r_2)$, which is the same as that of:

$$r_2 + r_3 (1 - \beta r_2) = r_2 + r_3 - \beta r_2 r_3$$

It is found that the coefficient β influences r_2 and r_3 *symmetrically*, and therefore should not appreciably modify the above results.

At what point to apply imbibition

Certain technicians put much emphasis on applying the imbibition as far ahead of the mill as possible. They consider that the dilution is thus more complete, the bagasse having more time to absorb the water or juice received.

Actually, the bagasse has a very great affinity for water, and dilution is very rapid and almost instantaneous. Numerous experiments have given results scarcely convincing, sometimes contradictory. In Trinidad⁷ imbibition applied just at the entry of the bagasse to the mill has even shown a slight gain relative to that applied at the delivery from the preceding mill, although the intermediate carriers were 8 m (25 ft.) in length.

There is therefore no occasion to attach much importance, from this point of view, to the position at which the maceration distributor is located.

It seems, on the other hand, that there is some advantage in directing the jets of the imbibition water spray in the plane tangential to the delivery rollers. Since part of the bagasse adheres to the top roller, this allows the water to reach the middle of the bagasse layer, and also to spray, during its fall, the bagasse dropping in a finely divided state from the top-roller scraper.

In short, care should be taken to distribute the imbibition as well as possible in the bagasse, but the best distribution is still that operating in the nip of the feed rollers; immediately pressure is applied, the excess liquid is expressed and forms a stream passing through the bagasse layer and wetting the zones which previously had remained dry. Most new factories now (1981) apply imbibition at the top of the Donnelly chute.

Hot or cold imbibition

Should the imbibition water be cold or hot? Experiments made on this question similarly show little agreement. Of the author's four factories, one has shown a fairly marked difference in extraction in favour of hot water, whereas the three others have failed to show any difference in this direction.

It appears that temperature has little effect up to the neighbourhood of 60–70°C (140–158°F). Above this point, the efficacy of imbibition is slightly superior with hot water, the gain in extraction reaching 0.4% in the most favourable cases⁸.

Supporters of hot imbibition explain the action of temperature by the destruction of the tissues of the cell walls by heat. The material of the cell walls, which is impermeable, or permeable only by osmosis, would be softened, and water thus would have direct access to the juice contained in the cells.

The following points may be noted:

(1) A slight gain in extraction is obtained with hot water, sometimes rather difficult to detect, and often negligible.

(2) The mills do not take the feed as well with hot imbibition as with cold. When the plant is operating at high tonnage relative to the normal capacity of the mills, and when heavy imbibition is used, performance is limited sooner with hot water than with cold.

In Hawaii, Haines and Hughes⁹ have experienced feeding difficulties only above 92.5°C (180°F). Between 56 and 80°C (133 and 175°F) they had, on the contrary, a slight gain, allowing an increase of the quantity of imbibition with water temperature. This temperature of 80°C would thus be an optimum. We may comment that these authors are among those who indicate¹⁰ a highly significant difference in extraction with hot water 77°C (170°F): with mill extraction 68, and that with cold water 26°C (79°F): extraction 62.2.

(3) Imbibition with hot water has the great advantage of saving cold water and of providing an excellent use for the hot condensate returned from the later vessels of the multiple effects.

Waxes. Cane carries on its rind a certain quantity of wax, most of which is found at the "wax ring", below the node. Some varieties (Co 290, B37.161, etc.) are particularly rich in waxes. Most of the waxes melt between 60 and 80°C (140–175°F), and most often between

65 and 75°C; and it has long been believed that imbibition with hot water had the serious disadvantage of increasing the proportion of waxes passing into the mixed juice. However, Honig¹¹ reports that experiments have been made on this subject in Java, and that they have led to the conclusion that imbibition with water at 85 – 95°C (185 – 203°F) does not extract any more wax than imbibition with water at 28°C (82°F). The proportion varies from 30 to 45% in the two cases, but is not appreciably affected by the temperature of the imbibition water.

On the other hand, the tests have shown that the intensity of disintegration undergone by the cane plays a favourable role, and that the bagasse *retains* an appreciably higher proportion of substances detrimental to filterability of juice and final raw sugar when it is better prepared and broken up, by knives, shredders, etc.

We may conclude then that, if it is considered that there is any advantage in using hot water as imbibition, there is no objection, from the point of view of manufacture, to doing so.

A good temperature for imbibition with hot water is 80 – 85°C (175 – 185°F).

Influence of imbibition on bagasse moisture

In all the preceding calculations, it has been assumed that the quantity of imbibition had no effect on the moisture of the bagasse. This is not strictly true.

From fundamental considerations, it would appear that the quantity of water introduced would facilitate rather than hinder the extraction of juice by the following mill. Actually, experience shows that the moisture of the final bagasse increases slightly with the quantity of imbibition. In Australia⁸ a case has even been found where the moisture increased from 47.4 to 50% when X was increased from 2 to 2.85.

Although such a difference is exceptional and is probably associated with an incomplete application of pressure at the last mill, it must be remembered that everything suggests that the last mill does not succeed in removing completely from the bagasse all the excess of water which has been added to it.

Absorptive capacity of bagasse

Dry bagasse can absorb from 5 to 10 times its weight of water. The bagasse from the mills contains about 1 – 2 parts of water to one of fibre. Since the quantity added is only of this order, it is very far from being saturated. It is from this fact that the difficulty arises of distributing the water properly in the bagasse; the first layer which receives the water tends to absorb it. Hence the advantage of application by jets rather than fine sprays.

As a matter of fact, the distribution of water or of thin juices, always more or less defective, is fortunately corrected to a certain extent by the flow of liquid under the pressure of the rollers (zone A in Fig. 9.9). At this point the very wet portions lose their excess of liquid, to the advantage of the drier portions. This favourable effect, however, should not be allowed in any way to lead to neglect of good distribution of the imbibition, which is still important.

Maceration

In certain countries, especially in Australia, imbibition was sometimes replaced by macera-

tion. The latter consists of steeping the bagasse in the liquid to be used as imbibition; the distribution thus becomes perfect, and the penetration more effective.

This system involves a high consumption of water and a high dilution. The bagasse being saturated, its feeding becomes difficult, and it is hardly possible to use maceration without employing mechanical feeders, preferably pushers or the pressure feeder developed in Australia for this specific purpose (see p. 94).

Maceration is effected in the bottom of the intermediate carrier, which is adapted to form a bath (Fig. 19.12). Its efficacy is often increased by heating the water or thin juice nearly to boiling before transferring it to the maceration bath.

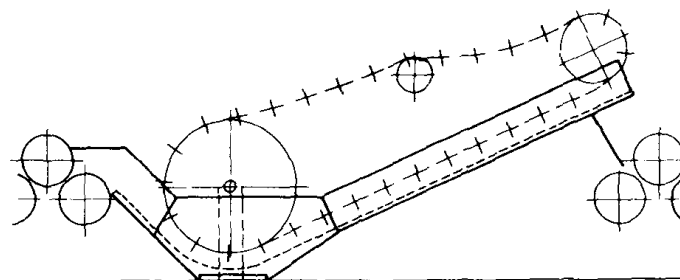


Fig. 19.12. Intermediate carrier and maceration bath.

The gain in extraction obtained by maceration is scarcely worth the complications involved. In general it is preferred to keep to a high ordinary imbibition, the results of which are hardly inferior.

Maceration bath carriers have practically disappeared in Australia.

Maximal imbibition

In modern factories, it is very rarely the evaporation capacity that limits the quantity of imbibition. Most often it is choking at the mills that determines the imbibition, and a practical limit is generally reached much below the optimal quantity. This limit, however, can be sometimes improved:

- (1) By utilising water at a lower temperature, or changing to cold water
- (2) By decreasing the hydraulic pressure
- (3) By increasing the mill speed
- (4) By decreasing the crushing rate.

Haines and Hughes¹² in Hawaii have studied the influence of these various factors and established a regression formula for the final mill at Puunene. Making use of two of their results, and expressing the others as a function of the results found in the same way in his own factories, the author gives:

$$\lambda_M = 554 - 0.9 t - 5 p + 20 v - 30 \tau$$

$$(\lambda_M = 570 - 0.5 t - 0.55 p + 6.1 v - 480 T) \quad (19.21)$$

λ_M = maximal practicable imbibition load, % on fibre

t = temperature of imbibition water, in °C (°F)

p = specific hydraulic pressure (s.h.p.) (cf. eqn. (10.6)) of the mill, in t/dm² (t./sq.ft.)

v = peripheral speed of the mill, in m/min (ft./min)

T = specific fibre loading, in kg/m²/m (lb./sq.ft./ft.) (cf. eqn. (10.19)).

Example. Taking a final mill of 812 X 1,676 mm (32 x 66 in.), crushing 130 t.c.h. at 12.5% fibre, under an s.h.p. of 25 t/dm² (230 t./sq.ft.), running at 12.5 m/min (41 ft./min), using water at 60°C (140°F), it should be possible to use an imbibition load of:

$$\begin{aligned}\lambda_M &= 554 - (0.9 \times 60.0) - (5 \times 25) + (20 \times 12.5) - (30 \times 16.00) = \\ &= 554 - 54 - 125 + 250 - 480 = 145\end{aligned}$$

Its specific fibre loading is:

$$\tau = \frac{1,000 \times 130 \times 0.125}{60 \times 12.5 \times 0.812 \times 1.676} = 16.00 \text{ kg/m}^3 \text{ (1.00 lb./sq.ft./ft.)}$$

This refers of course to imbibition applied entirely at the final mill. There is nothing to prevent increase of the determined quantity X by adding more water at the penultimate mill, the maximal quantity of this being given by the same formula.

Russell¹³ considers that the maximum imbibition admissible for any mill is $X = 3.6$. At higher values it is necessary to split the imbibition: if, for example, it is desired to apply imbibition at a value of 4.8, 3.6 would be used on the last mill, and 1.2 on the preceding mill. Moreover, not more than two-thirds of the juice can be applied to the one mill, the other one-third going to the preceding mill.

METHODS OF APPLICATION OF IMBIBITION

It is not possible to use the same method of application for water, which is free from foreign matter, and juice, which carries with it a substantial proportion of fine bagasse.

A. Water

Water is applied to the bagasse, either by means of sprays (Fig. 19.13), or by means of perforated pipes (Fig. 19.14). We prefer the second method.

According to the thickness of the bagasse layer, the water should be under a head of 3 - 5 m (10 - 16 ft.) at the pipe. The diameter of the holes in the pipe should be of the order of 3 mm (5 in.), or, better, of $0.004 D$, where D is the diameter of the mill rollers; and these holes should be spaced at intervals of 25 - 40 mm (1 - 1½ in.) along the pipe.

The holes may be made somewhat larger at the extremities of the pipe, for example, over the first and the last quarter of its length. Tests in Cuba¹⁴ have in fact shown that the polarisation of the bagasse was higher on the ends than at the centre of the roller. This is described by the Cubans as the "Betancourt imbibition system".

An excellent system for distribution of imbibition is that indicated in Fig. 19.15, which can

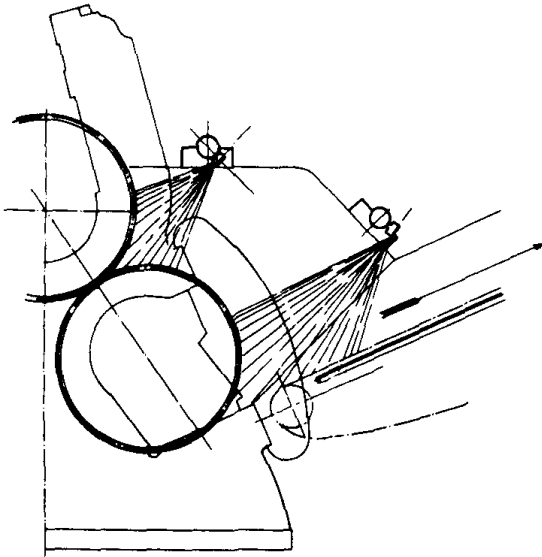


Fig. 19.13. Imbibition by fine sprays.

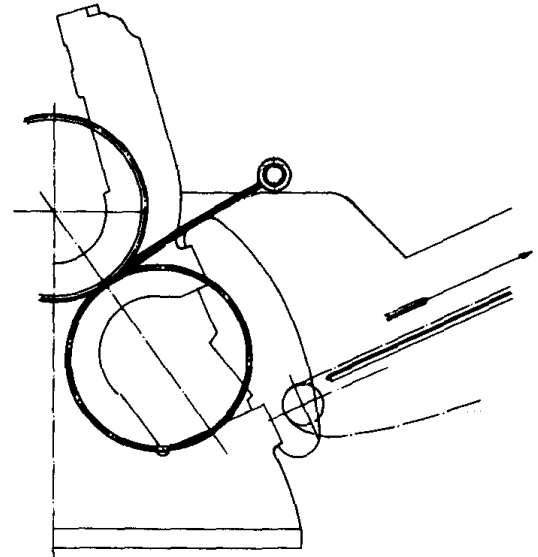


Fig. 19.14. Imbibition by perforated pipe.

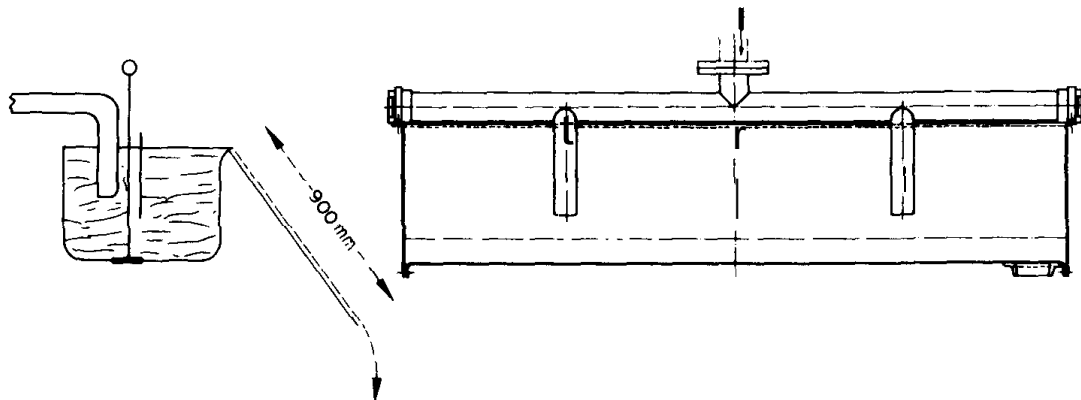


Fig. 19.15. Imbibition with juice by trough with long plate.

be equally recommended for water and for juice. It is desirable that the inclined sheet should have a length of 1 m (3 ft.), but for water it could be shorter.

B. Juice

Juice carries with it from the mills large quantities of fine bagasse, and sometimes quite large pieces. Hence it must either be screened, or handled by an unchokable pump.

For such a pump to function smoothly and not intermittently, it is essential that its suction should be taken from a vessel open to the air where the juice arrives, for example, by a swirling motion (a cylindrical tank with tangential feed) in such a way that, when the level falls, the pump sucks in a certain proportion of air without ceasing to pump.

The juice is distributed on the bagasse by one of the following methods:

(a) **Pipe and steel plate** (Fig. 19.16). The pipe discharges against a steel plate which flattens

the stream of juice and spreads it fanwise on an inclined plate, which increases in width to that of the intermediate carrier.

This method has one disadvantage: the distribution depends on the flow of juice. If it is correct for a certain flow, it will be too concentrated in the middle for a smaller flow, too strong at the ends for a greater flow.

(b) Trough open on one side. The juice discharges by a pipe at right-angles to one of the side plates of the intermediate carrier into a flat-bottomed trough set on a slight slope, and open on one side and narrowing from the inlet to the opposite end; this is set almost horizontally, but the opposite end may be slightly raised or lowered to obtain distribution of the juice across the whole width of the carrier. This system has the same disadvantage as the preceding one; it does not achieve a good distribution to the portion of the carrier situated close to the discharge, and the width of the stream varies from one moment to the next.

(c) Serrated trough. The juice pipe may terminate in a horizontal trough, the two edges of which are serrated (Fig. 19.17). Here, small pieces of bagasse become caught across the notches and hinder proper distribution.

(d) Baffled trough and wide plate (Fig. 19.15). The best system is that already indicated for water. If there is insufficient space to give the plate a length of 1 m (3 ft.), it may be left at 40 cm (16 in.) and the curtain of juice arranged to fall on another plate inclined in

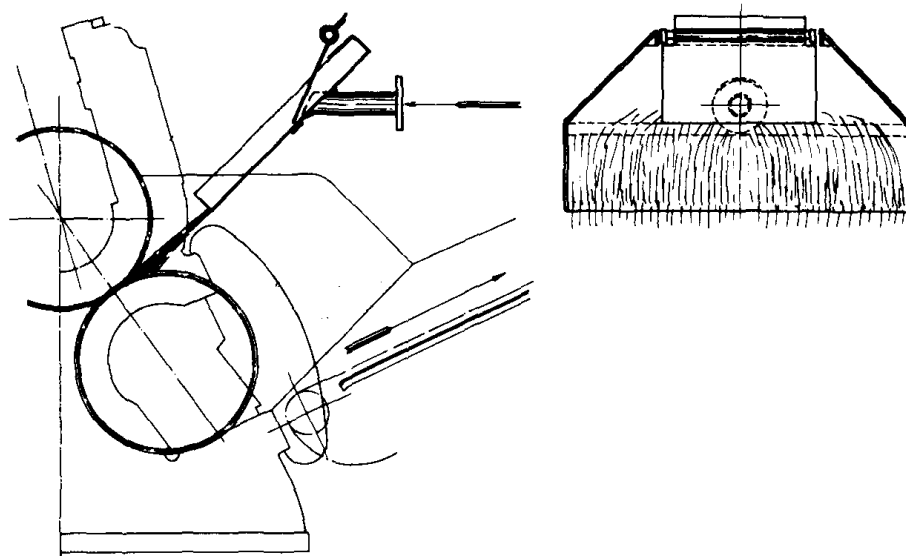


Fig. 19.16. Imbibition with juice by pipe and steel plate.



Fig. 19.17. Serrated trough.

the opposite direction, also 40 cm in length; the distribution will thus be improved. The juice flows in a uniform and continuous curtain.

(e) **Donnelly chute.** With mills which are fed by Donnelly chutes, the system generally adopted is to distribute the imbibition fluid at the top of the chute.

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20. *Extraction*

The extraction of a train of mills has a definite meaning which we shall define below. The purpose of the present chapter is to study, in a general manner, the efficiency of the milling work. We shall return to these calculations in Chapters 21 and 39 with the general factory control.

MEASURE OF EFFICIENCY OF MILLING WORK

What is a suitable standard to adopt as a measure of the efficiency of a milling train?

(1) **Sucrose per cent bagasse**

The first figure which one is inclined to suggest is that of sucrose % bagasse; but this depends not only on the efficiency of the milling train, but also on the moisture content of bagasse and, to a slight extent, on the sugar content of the cane.

(2) **Milling loss**

The moisture content of bagasse has been taken into account in Hawaii by adopting the expression "milling loss", or "sucrose/fibre coefficient":

$$\text{Milling loss} = \frac{\text{sucrose (or pol) in mixed juice \% cane}}{\text{fibre \% bagasse}} \times 100 \quad (20.1)$$

This expression corrects for the effect of moisture in bagasse; if, starting with the same cane, we compare two bagasses having the same sucrose per cent, a , and different moisture contents, this coefficient would be closely proportional to the loss at the mills, β , involved for each of them, since the weight b of bagasse as percentage of cane is inversely proportional to the fibre f' per cent of bagasse:

$$\beta_1 = \sigma b_1 = a \frac{\sigma}{f'_1} \quad \beta_2 = \sigma b_2 = a \frac{\sigma}{f'_2}$$

This expression, however, does not correct for the effect of sucrose content of cane.

(3) **Extraction**

It is more interesting to know the proportion of sucrose extracted by the mill. This figure expresses essentially the commercial efficiency, rather than the technical efficiency, of the operation:

$$\text{Extraction} = \frac{\text{sucrose or pol in mixed juice \% cane}}{\text{sucrose or pol \% cane}} \times 100 \quad (20.2)$$

Since the sugar extracted is closely proportional, other things being equal, to the sugar content of cane, this expression has the advantage of eliminating at least theoretically the factor for sucrose in cane; but it has the disadvantage that it depends on the fibre content f of the cane; the higher the fibre, the greater the loss of the sugar in the bagasse, β , since:

$$\text{Sugar loss in bagasse \% cane} = \text{sugar \% bagasse} \times \text{bagasse \% cane}$$

$$O = ob = a \frac{f}{f'}$$

For this reason, mills handling high-fibre cane are at a disadvantage. The extraction figure thus does not permit comparison of the work of two different mills, nor even of the same mill with different canes.

On the other hand, we see later (p. 318) that the sugar extracted is far from being proportional to the sugar % cane. If a given tandem, maintaining the same efficiency, treats successively a poor cane and a rich cane, the sucrose % bagasse will vary very little between the two cases. Hence the extraction figure, without application of a correction factor for the sugar in cane, will not serve to express the efficiency of a milling plant.

(4) Reduced extraction

Noel Deerr has sought to eliminate this predominant disturbing factor, f , by proposing as a figure for comparison, "reduced extraction".

This is the extraction obtained by the tandem, modified hypothetically to that which it would have been if the fibre in cane had been equal to a standard figure, chosen by convention as 0.125.

The reduced extraction is designated by $e_{12.5}$. Deerr calculates it as follows. Let:

e = extraction obtained with cane of fibre f

E = extraction which would have been obtained with cane of fibre F .

For these corresponding extraction figures, Deerr gives the following equation, postulating that both cases would have furnished the same loss of juice % fibre (cf. p. 319):

$$\frac{(1 - e)(1 - f)}{J} = \frac{(1 - E)(1 - F)}{F} \quad (20.3)$$

Hence:

$$E = 1 - (1 - e) \frac{1 - f}{f} \cdot \frac{F}{1 - F}$$

or, for $F = 0.125$:

same t.c.h.), it would obtain in the second case a better extraction. This is a consequence of the fact that there is a certain quantity of juice, hence of sugar, which is very difficult to extract by pressure and which remains in the bagasse; and that this quantity is very nearly the same in rich cane as in poor cane. Hence the rich cane shows a better extraction.

Ugale³ has thus proposed to multiply the figure for reduced extraction modified for fibre content (Deerr or Mittal) by the ratio $7.125/(7 + P)$, where P is the pol per unit of cane; this indicates that he adopts 12.5% as the standard pol % cane. From the formula of Deerr, we have for example:

$$E_{12.5, 12.5} = \left[1 - \frac{(1 - e)(1 - f)}{7f} \cdot \frac{7.125}{1 + P} \right] \quad (\text{Ugale}) \quad (20.8)$$

Rein⁴ in South Africa has proposed:

$$E_{15.5, 13} = 1 - 0.1834 - \frac{(1 - e)(1 - f)}{7f} \left(\frac{P}{13} \right)^{0.6} \quad (\text{Rein}) \quad (20.9)$$

P being pol % cane, and taking standard extraction as 95 for 15.5% fibre content and 13% pol in cane (conditions closer to mean figures for South Africa).

To compare these two formulae, we use for the Deerr – Ugale formula, the same basis as adopted by Rein ($f = 0.155$, $P = 0.13$) and see what would be the extraction of a tandem giving for example 95% reduced extraction:

	<i>Ugale</i>	<i>Rein</i>
	$e = 95 \cdot \frac{1 + P}{7.13}$	$e = 1 - 5 \left(\frac{13}{P} \right)^{0.6}$
$P = 10:$	94.60	94.15
$P = 16:$	95.40	95.59

The Rein correction is much greater, and is probably better founded. No country has adopted it so far but, in fairness, it should be taken into account.

At Tongaat, for a pol content approximating 13%, fibre content 15.4%, imbibition 360% on fibre and an extraction of 96%, it is reckoned that a change of 1 point of sugar content would lead to a change in the same sense approximating 0.2 point in extraction. This corresponds fairly well with the formula of Rein.

The formula of Rein, applied to Mittal's reduced extraction (eqn. (20.6)), adopting 12.5 as the standard sugar content, and expressing this as sucrose, designated r , becomes:

$$e_{12.5, 12.5} = 1 - \left[\frac{0.125(1 - e)}{f} \left(\frac{r}{12.5} \right)^{0.6} \right] \quad (20.10)$$

(5) Extraction ratio

This expression, brought into prominence by the Hawaiians, corresponds exactly to the preceding reasoning:

$$\text{Extraction ratio} = \frac{100 (100 - \text{extraction})}{\text{fibre \% cane}} \quad (20.11)$$

This is a correct measure, but it takes into account only the factor which is the main consideration against extraction, namely, the fibre content.

(6) Lost undiluted juice per cent fibre

This figure, originating in Java, is one which has been widely used, for international comparisons as well as for comparisons between factories:

$$\text{Lost juice \% fibre} = \frac{\text{brix of bagasse} \times 10,000}{\text{brix of primary juice} \times \text{fibre \% bagasse}} \quad (20.12)$$

where:

$$\text{Brix of bagasse} = \frac{\text{pol \% bagasse} \times 100}{\text{purity of last expressed juice}} \quad (20.13)$$

This expression possesses the following disadvantages:

(a) Equation (20.13) is inaccurate, since the purity of the last extracted juice (and *a fortiori* that of the final-mill juice, which is often utilised in its place) is higher than that of the residual juice contained in the bagasse.

(b) What is of interest in the performance of a mill is its efficiency in extracting sucrose, not in extracting brix. It would be perfectly satisfactory if the extraction of a milling plant were good for sucrose but low for the non-sucrose making up the brix. The correlation between these two quantities is not complete; it is even less so when comparing mills with diffusion.

(c) The brix of the primary juice depends on the nature and settings of the early mills of the train (*cf.* p. 345 and Fig. 21.6).

(7) Lost absolute juice per cent fibre

Another attempt at a more useful expression is that of substituting for the brix of primary juice that of the absolute juice in cane. This is certainly an improvement, but we may still comment that:

(a) The first two disadvantages persist.

(b) The absolute juice necessitates a rather lengthy calculation.

This latter expression may equally well be written in the form:

$$\text{Lost absolute juice \% fibre} = \frac{\text{absolute juice in bagasse \% cane}}{\text{fibre \% cane}} \times 100 \quad (20.14)$$

which demonstrates better the origin of this expression; but it is never calculated in this form.

Comparison of the several efficiency figures

We consider that:

(7) The milling loss offers only a secondary interest; it serves only to eliminate the influence of bagasse moisture in comparing values of sucrose % bagasse.

(2) The figure of lost juice % fibre had been adopted in Java because it did not require any weights of materials. Today, however, there is no factory of any consequence which does not have material balances and complete chemical control, and its retention is no longer justified.

(3) It is essential, for commercial reasons, to retain the extraction figure, but there would be no question of using it as a technical criterion of the work of the milling train.

(4) Noel Deerr's reduced extraction, not content with correcting the effect of fibre by placing it in the denominator, further introduces the factor $(1 - f)$ in the numerator. We shall rule it out as exceeding the stated objective.

Mittal's reduced extraction may be used for valuable comparisons, but, with the extraction coefficient, involves unnecessary repetition.

(5) The lost absolute juice in bagasse has the same form as the extraction figure. It represents:

$$\frac{\text{Brix lost \% brix in cane}}{\text{Fibre \% cane}} \quad \text{instead of} \quad \frac{\text{Sugar lost \% sugar in cane}}{\text{Fibre \% cane}}$$

In other words, it measures the brix extraction, whereas the extraction ratio measures the extraction of sugar. These two extractions are very close, as are the figures which determine them; sometimes one and sometimes the other is higher, and the ratio of the first to the second varies generally between 0.95 and 1.05 for modern mills, giving values lying between 38 and 30, the two figures tending to be equal at about 34, the extraction ratio falling more rapidly below that figure and increasing more rapidly above it. We would have for example the following equivalent values:

Extraction ratio	30	34	38
Lost absolute juice % fibre	32	34	36

The lost absolute juice % fibre would perhaps measure slightly better the mechanical efficiency of the mills, which is a matter of volume and of juice; but we prefer the extraction ratio on account of:

- (a) The relationship between the two values, which lends little value to that argument
- (b) The simplicity of the formula for extraction ratio
- (c) The greater importance of sugar, which is indeed our objective and our only preoccupation
- (d) The possible influence of certain details of the mills, such as preparation and grooving, on the extraction of sugar compared with that of brix.

We consider also that it is necessary to consider, apart from the extraction figure, the

reduced extraction taking into account not only the fibre f , but also the sugar content r , utilising Rein's formula indicated above, which can be reduced to the basic traditional values:

$$\text{Following Deerr: } E_{12.5 \ 12.5} = 1 \left[\frac{(1 - e)(1 - \gamma)}{7f} \right] \left(\frac{r}{2.5} \right)^{0.6} \quad (20.15)$$

$$\text{Following Mittal: } E_{12.5 \ 12.5} = 1 - \frac{10.125(1 - e)}{J} \left[\frac{r}{12.5} \right]^{0.6} \quad (20.16)$$

Such formulae have not been adopted in the mutual control schemes of sugar countries, mainly on account of the complication of the power 0.6. This reason however is not valid now, since all laboratories are equipped with pocket calculators giving this calculation immediately, several makes of calculator being available.

FACTORS IN EFFICIENCY OF MILLS

The principal elements which influence the efficiency of milling work as measured by extraction ratio, or reduced extraction, are the following:

- State of preparation of the cane
- Specific pressure employed
- Length of the train (or number of rollers)
- Speed of rotation of the rollers
- Specific fibre loading
- Imbibition

to which should be added certain factors less easy to evaluate, such as:

- Correct setting of the mills
- Technical worth of the personnel in charge of the mills
- Mechanical condition of the plant.

TABLE 20.1

GAIN IN EXTRACTION OBTAINED BY VARIOUS CANE PREPARATORS

(a) Knives	$\epsilon = \frac{3}{N - 6} \%$
(b) Searby-type shredder	$\epsilon = \frac{7.5}{N - 8} \%$
(c) Maxwell-type shredder	$\epsilon = \frac{2.5}{N - 8} \%$

N = number of rollers in the train.

(1) Cane preparation

We have already given some figures in Chapters 3 and 6 and shall complete them here. For the common types of preparator. Table 20.1 gives the gains in extraction e , reckoned in the

same unit as the extraction (that is to say, if the extraction rises from 90 to 91%, the gain $e = 1\%$).

Moor⁷ quotes an experiment made at Tongaat where bagasse pieces of various sizes separated by sieving were analysed separately: pieces larger than 25 mm, then 25 - 12, 12 - 6 - 3, and below 3 mm. The mean pol % bagasse was 1.3; the finest fraction showed 0.8^t whereas the largest fraction showed 3%. This shows in spectacular fashion the effect preparation on extraction.

(2) Specific pressure (cf. p. 130)

The extraction increases rapidly as the pressure increases from zero to 16 t/dm² (150 t./sq.ft) and more slowly thereafter. The values most interesting from an economic point of view range from 20 to 25 t/dm² (180-230 t./sq.ft.).

(3) Length of the tandem

As we have seen for capacity (cf. Table 12.4), the figures published annually for Java permitted comparisons to be made of the extractions of the various combinations of mills existing in that country. On referring to Table 12.4 (p. 193), we see a regular variation of extraction and lost juice per cent fibre, as a function of the length of the tandem. We give in Fig. 20.1 a graph derived from these and various other figures, which, however, does not profess to show more than the general trend of the relationship.

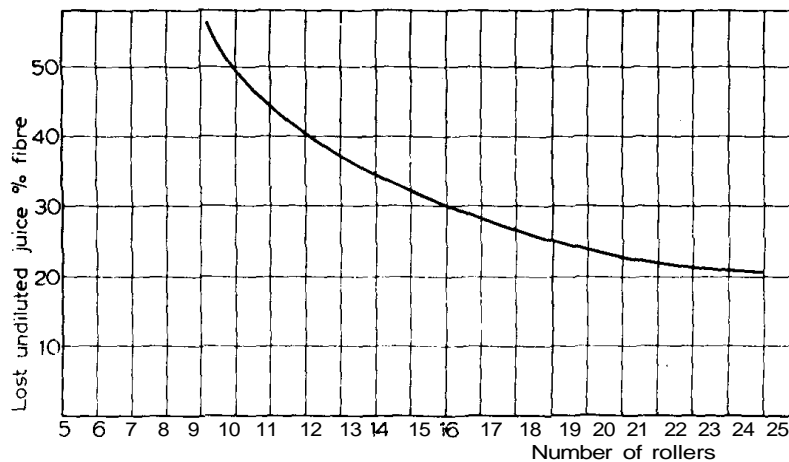


Fig. 20.1. Lost undiluted juice % fibre as a function of number of rollers ($X = 2$; $\beta = 0.6$).

It can be seen that the lost juice % fibre is represented approximately by the equation:

$$j = \frac{500}{N} \quad (20.17)$$

j = lost undiluted juice % fibre for the tandem

N = number of rollers in the tandem.

In the same way, the extraction may be expressed approximately by the formula:

$$e = 100 - \frac{100}{N} \quad (20.18)$$

which assumes an average fibre of approximately $f = 12.5\%$.

It may also be of interest for a technologist dealing with a short or medium tandem of mills (4 or 5) to judge the work of his tandem in relation to the results given by longer tandems, of 6 or 7 mills, for example.

On this question we have suggested comparing the extractions, the juice extractions, extraction coefficients or simply the sucrose lost in bagasse by considering that the work of the tandem should vary, as a function of the number of mills and the sugar content of cane (other things being equal), as the N th root of the sugar content R (N being the number of mills). To compare, for example, the sucrose in bagasse a for a tandem of 5 mills with that which should be obtained with a 7-mill tandem, we would have:

$$\frac{\sigma_5}{\sigma_7} = \frac{\sqrt[5]{R_5}}{\sqrt[7]{R_7}} \quad (20.19)$$

if $R_5 = R_7 = 14$, we would have:

$$\frac{\sigma_5}{\sigma_7} = \frac{1.695}{1.458} = 1.16$$

that is, the difference between 2.32 and 2, for example.

It should be pointed out that this rule, which sometimes applies in practice, has no experimental basis. Table 20.2 gives values of the appropriate roots ($\sqrt[N]{R}$).

TABLE 20.2

ROOTS OF R FOR EXTRACTION BY TANDEMS OF VARIOUS LENGTHS

R	$\sqrt[4]{R}$	$\sqrt[5]{R}$	$\sqrt[6]{R}$	$\sqrt[7]{R}$	$\sqrt[8]{R}$
11	1.821	1.615	1.491	1.409	1.350
12	1.861	1.644	1.513	1.426	1.364
13	1.879	1.670	1.533	1.443	1.378
14	1.934	1.695	1.553	1.458	1.391
15	1.968	1.719	1.570	1.472	1.403
16	2.000	1.741	1.587	1.486	1.414

(4) Roller speed

The influence of roller speed depends on whether the fibre loading or the crushing rate is assumed constant.

In the former case, that is, if the same thickness of bagasse blanket is maintained, the extraction decreases as the speed of rotation increases; imperceptibly at low speeds, more definitely, for the usual roller diameters, as a speed of 5 r.p.m. is approached, and rapidly above about 6 r.p.m.

In the latter case, that is, if the crushing rate is kept the same, the extraction increases with speed of rotation, but very slightly, and only provided the limits expressed by eqns. (11.9) have not been reached. Crawford⁵ considers that there is a gain at the feed roller which more than compensates for the slight loss at the delivery roller. However, in both cases, the influence of speed is slight.

(5) Specific fibre loading

The specific fibre loading T is a measure of the ratio

$$\theta = \frac{\text{tonnage actually crushed}}{\text{theoretical capacity of the tandem}}$$

as we have seen on p. 196. The study of the influence of specific fibre loading r on extraction reduces to a study of the influence of the capacity ratio B .

The extraction, or, to be more precise, the efficiency of the mill work, decreases as the capacity ratio (or, what amounts to the same thing, the specific fibre loading) increases.

In India it is estimated that the reduced extraction of a milling plant falls by 0.17 when it is operated at 10% above normal rate⁶. For example, if a mill is capable of crushing 100 t.c.h., and if its reduced extraction is 95, this should drop to 94.83 if the rate is increased to 110 t.c.h.

This value of 0.17 appears low, and would be applicable only for crushing rates in the neighbourhood of the theoretical capacity of the plant or slightly below; but it demonstrates afresh the flexibility of a milling plant as regards capacity.

(6) Imbibition

This is the most interesting factor to study from the point of view of what use can be made of an existing milling plant. It is also the one which can be most easily controlled, and which can be most readily varied.

Dry crushing will generally give an extraction of about 86 – 90%. If we assume 90%, which is really a maximum, the graphs in Fig. 19.8 enable us to read off directly the final extraction corresponding to a given imbibition. For example, with double compound imbibition:

$$\text{for } X = 1: \quad e = 90 + 3.8 = 93.8\%$$

$$\text{for } X = 2: \quad e = 90 + 5.2 = 95.2\%$$

In fig. 20.2 is given a graph which, like that of Fig. 20.1, should be regarded as indicative only, and which gives the general trend of a series of tests made in South Africa on the variation of extraction as a function of imbibition. It enables an estimate to be made of the magnitude of the change in extraction to be obtained with a given alteration in imbibition.

At Tongaat, in conditions already indicated (see p. 318), it is reckoned⁷ that a change in imbibition % fibre of 1 point (from 360 to 361, for example) involves a change in extraction in the same direction of 0.01 point approximately (96-96.01).

Similarly, for the commonest types of tandems, Fig. 20.3 gives the change in lost juice %

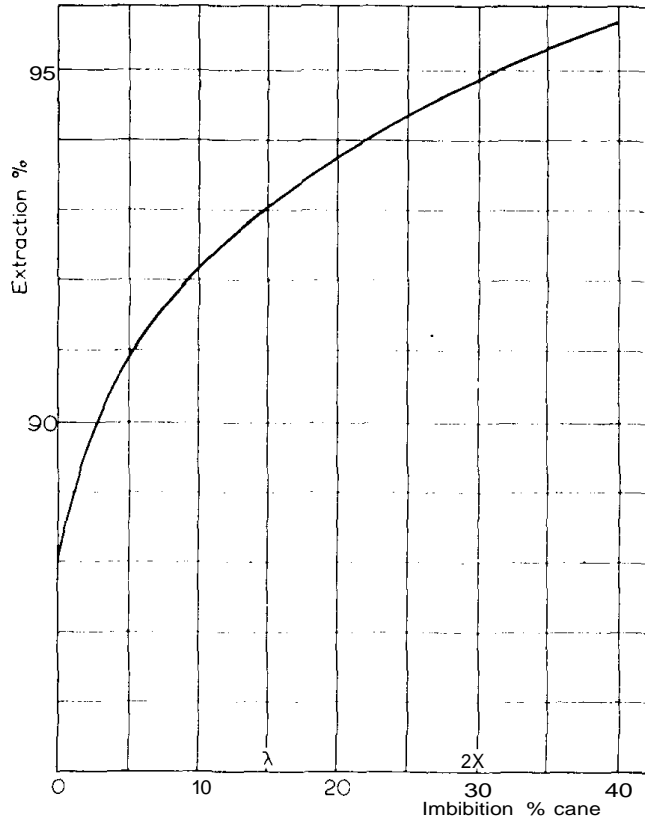


Fig. 20.2. Variation of extraction as a function of imbibition (simple double imbibition) ($f = 0.15$).

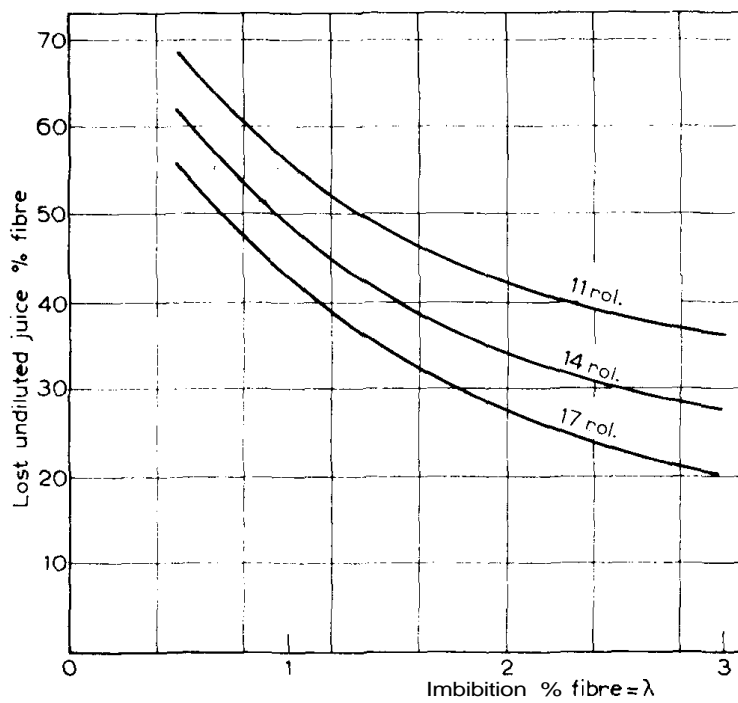


Fig. 20.3. Variation of lost juice % fibre as a function of imbibition % fibre (compound imbibition) ($\beta = 0.6$).

fibre, as a function of imbibition % fibre, X . These graphs, again, express average results only, and assume the efficiency of imbibition $\theta = 0.6$, with compound imbibition; double for 11-, triple for 14- and quadruple for 17-roller tandems.

To compare the work of the same tandem with two different values of imbibition or of different tandems working with different imbibition values, it has been proposed⁸ that the influence of imbibition W may be eliminated by considering that the extraction obtained should be proportional to:

$$100 - \frac{130 + W}{10 + W} \quad (20.20)$$

when W varies; where W = imbibition % on cane.

SANITATION AT THE MILLS

The milling control (*cf.* p. 328) does not detect physical or chemical losses suffered during the passage of the cane through the milling train. The sugar content of the cane is indeed assumed equal to the sum of sugar contained in the mixed juice % on cane and of that lost in the bagasse % on cane. In this way, all spillage from the mill trays to the floor, and all destruction of sucrose taking place during the movement of the cane from the carrier till it leaves the last mill, remain unknown. In these conditions it is all the more important to take all possible precautions to avoid losses which would not even be detected.

Destruction of sucrose is due to three causes⁹: acid inversion, enzymatic inversion, and microbial infection. These causes have been estimated as causing respectively 16, 22 and 62% of the total losses due to infection. When sanitation at the mills is inadequate, the total of these losses of sucrose can attain 2.5 kg (5.5 lb.) per t.c. In a factory kept clean and well disinfected, the losses from microbial origin can be appreciably reduced, without, however, eliminating them completely. It is accepted that an improvement of 0.9 kg (2 lb.) of sucrose per t.c. can be realised.

Several factors reveal inadequacy of sanitation: acid odours around the mills, presence of slime or of colonies of *Leuconostoc* ("frog spawn"). Laboratory control may show an abnormal purity drop between first expressed juice and mixed juice.

Apart from the sucrose losses involved, certain strains of bacteria produce gums (dextran, etc.) which affect performance at the pans and lead to the production of elongated crystals.

In order to maintain good sanitation, it is necessary to practise effective cleaning of equipment as well as utilisation of a suitable disinfectant in appropriate quantities.

The principal requirements of a bactericide are:

- (a) Effective disinfectant power at the pH of cane juice (5.5)
- (b) Inactivity towards inorganic matter
- (c) Degradability in the course of manufacture; absence of traces of the disinfectant in the molasses, if intended for fermentation or for animal feed
- (*cf.*) Stability during storage; the efficacy of certain products decreases rapidly when stored at ambient temperature in tropical countries.

The manufacturers (SICCA, OLIN, HODAG, ROHM and HAAS, TATE and LYLE) offer various products, varying widely in composition, efficacy and cost: halogen derivatives, organic sulphur compounds, formaldehyde, quaternary ammonium salts. The latter appear very effective, but their use in food manufacturing processes is forbidden in certain countries. It has been shown, however, in South Africa¹⁰ that only traces remain after the evaporation process. The organic sulphur compounds would be at least equally effective, but their cost is much higher. Formaldehyde used at the mills has the disadvantage of releasing irritating vapours. Halogen derivatives (hypochlorites essentially) present the serious disadvantage of reacting with organic material, which decreases their efficacy. Moreover, they are corrosive.

The ideal disinfectant does not exist as yet. It is up to each factory to check the materials employed in terms of purity drop, increase in activity, and glucose ratio between first expressed juice and mixed juice. The dosage will be adapted to requirements, which vary with the state of the cane and the ambient temperature. The material will preferably be added continuously if it possesses no special properties which lead it to remain in the milling plant. Care will be taken that the whole tandem is disinfected, and this is realised in practice by distributing the material equally on the fibrated cane and in the tray of the last mill. At least once per shift a complete cleaning of the tandem should be carried out with hot water (condensate), the sterilising and cleaning properties of which are definitely superior to those of steam.

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21. Milling Control

In this chapter we discuss only matters peculiar to the control of the milling plant. Further reference to this control will be made in Chapter 39 in its relationship to the general control of the factory.

EXTRACTION BY DRY CRUSHING

In order to discuss fully the question of efficiency it will be necessary, as we have done at the beginning of the study on imbibition (p. 280), to distinguish between dry crushing and wet crushing.

In practice, dry crushing corresponds only to:

- (a) For tandems with a 2-roll crusher, the crusher and 1st mill
- (b) For tandems without a 2-roll crusher, the 1st mill (sometimes described as a mill-crusher).

If a large number of mills were available, the efficiency of this first unit or of this first combination would not have great importance. The dry crushing could then be pushed to its limit (*cf.* p. 281), which moreover is rapidly attained. It would not, however, be desirable to lose even one mill to dry-crushing purposes in this way, since nothing would be gained, and one must be content with the combination or the unit mentioned above, for dry crushing (we shall call this the 1st mill, on the understanding that the term includes the crusher where there is one). Now, the dry-crushing extraction has a very marked influence on the overall extraction of the tandem, as eqns. (19.11) and (19.15) demonstrate. In practice, it varies between wide limits from one tandem to another. The greater the extraction of sugar by dry crushing, at the head of the tandem, the less difficult is the task left to the following mills to recover more sugar by the troublesome step of wet extraction, and the better is the overall extraction of the tandem. It is thus important to attain the maximum extraction possible by the dry crushing. It is considered in Queensland, for example, that a gain of 1 % in the extraction of the 1st mill gives a gain in total extraction of:

0.12% in 4-mill tandems

0.10% in 5-mill tandems

Since the 1st-mill extraction may vary between 60 and 80%, we see the huge influence of this extraction on the final result, for the same work of the subsequent wet crushing.

It is desirable, therefore, to follow carefully the 1st-mill extraction, and to report the figure in the weekly chemical control bulletins. A tandem of 3-roller units should attain a minimum

of 60% extraction at the 1st mill; it may readily reach 70%, and can go as high as 75 - 80%, according to the fibre content of cane, if the preparation is good (knives and shredder), and if all necessary attention is given to feeding and setting of the 1st mill. With a crusher, these values may even be exceeded (*cf.* Table 21.2, p. 332). For a 3-roller 1st mill without crusher, Australian workers report¹ extractions of 81 - 82% and 1st bagasse moistures of 53 or even 50% for a combination of crusher and 1st mill². Hawaiian factories³ for this combination normally reckon 77 - 80% of absolute juice (not of sugar), and in excellent conditions 80-83%.

Extraction of primary juice

In order to know the extraction of juice by dry crushing, it is not necessary to weigh it separately. A ready calculation may be made, in the general case where the primary juice is sampled and analysed separately and where all the rest of the juice is obtained at the second level (general case of compound imbibition) forming the secondary juice.

Let:

S_1 = sugar % primary juice

S_2 = sugar % secondary juice

S_m = sugar % mixed juice

e_1 = primary juice extraction = sugar in primary juice % sugar in cane

e = total extraction of the tandem

Q_1 = primary juice per unit of mixed juice

x = sugar in primary juice per unit of sugar in mixed juice.

Equating sugar in mixed juice to the total of that in the primary and secondary juice, we have:

$$1 \times S_m = Q_1 S_1 + (1 - Q_1) S_2 \quad (21.1)$$

Now:

$$x = \frac{S_1 Q_1}{S_m \times 1}, \text{ hence } Q_1 = x \frac{S_m}{S_1}$$

Substituting in (21.1) and solving for x :

$$x = \frac{S_1 (S_m - S_2)}{S_m (S_1 - S_2)} \quad (21.2)$$

But the sugar in mixed juice = e % of sugar in cane. Thus the extraction of sugar in primary juice % sugar in cane is given by:

$$e_1 = x e = \frac{S_1 (S_m - S_2)}{S_m (S_1 - S_2)} \cdot e \quad (21.3)$$

This calculation, however, may only be used when it is definite that there is no mixture of 3rd-mill juice with that of the 2nd mill after the sampling point (generally at the juice screens). Otherwise the result is obviously false.

The sampling should thus be carried out at the second mill. It is preferable to operate as follows. Let:

f = fibre per unit of cane

S_0 = pol or sucrose per unit cane

f_1 = fibre per unit of first-mill bagasse

S_1 = pol or sucrose per unit of first-mill bagasse

B_1 = weight of first-mill bagasse per unit of cane.

We have:

$$\text{Pol or sucrose in first-mill bagasse per unit of cane} = S_1 B_1 = S_1 \frac{f}{J_1}$$

$$\text{Extraction of first mill} = e_1 = \frac{1 \times S_0 - S_1 \frac{f}{f_1}}{1 \times S_0} = 1 - \frac{S_1 f}{S_0 f_1} \quad (21.4)$$

This calculation assumes that the fibre is the same before and after the mill (actually, a small amount is lost from the mill with the fine bagasse).

We recommend this second method as simpler and more accurate. It applies not only to the first mill, but for all mills. For mill n , the extraction is:

$$e_n = 1 - \frac{S_n \cdot \frac{f_{n-1}}{J_n}}{S_{n-1}} \quad (21.5)$$

Order of magnitude of first-mill extraction. The first-mill extraction depends on fibre in cane, and the magnitude of this extraction could be expressed by:

$$e_1 = 1 - \frac{0.06}{0.34 - f} \quad (21.6)$$

This formula is empirical, without mathematical basis. The fibre in the first-mill bagasse depends on fibre in cane and on first-mill extraction. It increases as extraction increases, and roughly e_1 is of the order of $2f_1$.

Extraction at successive mills. Table 21.1 gives two examples from South Africa, for the Illovo and Noodsberg factories, with fibre of 14.6 and I.P. of 89 in both cases.

TABLE 21.1

EXTRACTION (UNIT AND CUMULATIVE) IN SUCCESSIVE MILLS

	<i>Illovo</i>		<i>Noodsberg</i>	
	<i>Unit</i>	<i>Cumulative</i>	<i>Unit</i>	<i>Cumulative</i>
1st mill	72.5	72.5	72.1	72.1
2nd mill	25.5	79.5	42.1	83.8
3rd mill	42.0	88.1	43.1	90.8
4th mill	28.5	91.5	33.5	93.9
5th mill	21.7	93.3	36.0	96.1
6th mill	37.3	95.8	34.7	97.4

BRIX GRAPHS

One of the best methods of controlling the milling plant is to construct a graph of the brix of the juices from the successive mills.

We have seen (eqn. (19.19)) how the brix decreases from one mill to the other in the case of compound imbibition, which is most generally adopted.

The degree to which the values obtained in the mills approach the theoretical curve indicates how well each mill is doing its work. We give (Fig. 21.1) two graphs by way of example:

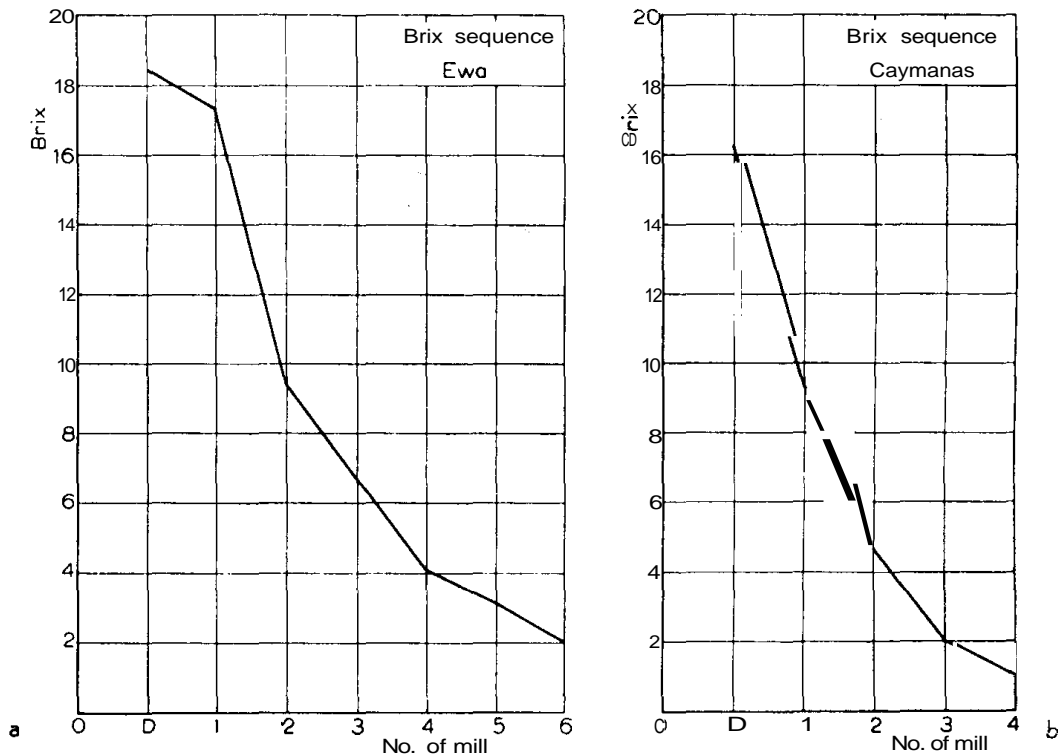


Fig. 21.1. Brix graph: (a) Ewa; (b) Caymanas.

(a) The mean of brix values obtained in the 20-roller tandem at Ewa, in Hawaii, during the 6 seasons 1928 – 33⁴

(b) A summary⁵ of the brix figures for the 15-roller tandem at Caymanas, in Jamaica, to which we have already made reference (*cf.* p. 57).

Whereas this latter curve is excellent, there will be noticed on the previous one an anomaly corresponding to the 5th mill and showing that the 4th is not extracting as much juice as it should.

It may be noted in passing that the brix values reported and the corresponding extraction for each of the mills at Ewa were as given in Table 21.2:

TABLE 21.2

EWA: BRIX AND EXTRACTION AT DIFFERENT MILLS

	<i>Brix</i>	<i>Extraction</i>
Crusher	18.53	69.92
1st mill	17.41	12.58
2nd mill	9.40	7.11
3rd mill	6.45	3.75
4th mill	4.16	1.50
5th mill	3.18	2.08
6th mill	1.96	1.22
<i>Total</i>		98.16

It may be noted also that the 15-roller tandem at Caymanas has the advantage of being able to use imbibition immediately after the crusher, whereas Ewa cannot commence imbibition until after the first mill, and thus loses one pressure from the point of view of imbibition. In addition to the reasons given previously (*cf.* p. 58), this is a further powerful argument in favour of the mill-crusher.

To establish these brix curves, the brix of back-roller juice is generally taken. In dry-crushing mills, the brix of the feed-roller juice is the higher. In wet-crushing mills, on the contrary, it is the back-roller juice which has the higher brix.

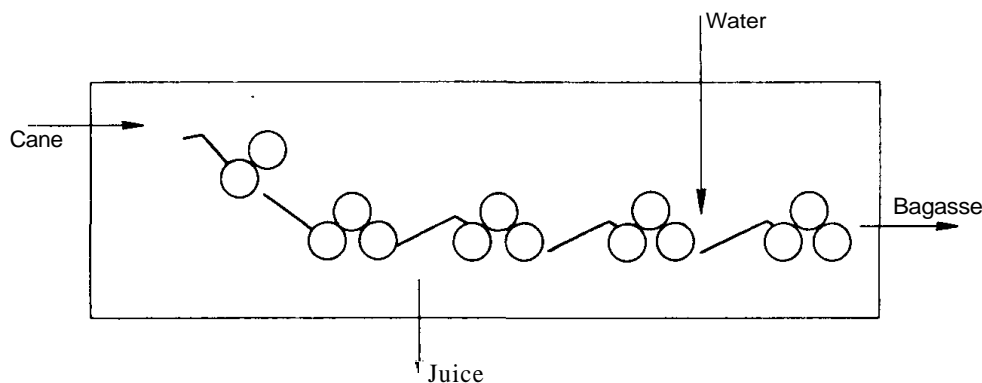


Fig. 21.2. Basic equation for mills.

BASIC EQUATION FOR MILL CONTROL

The mill control is based on an equation expressing the fact that the weight of material entering the milling plant is equal to the weight leaving (Fig. 21.2):

$$\text{Cane} + \text{imbibition water} = \text{mixed juice} + \text{bagasse} \quad (21.7)$$

or, relating all quantities to 100 of cane:

$$100 + W = Q + B \quad (21.8)$$

W = imbibition water % cane

Q = weight of mixed juice % cane

B = weight of bagasse % cane.

This equation, which is obvious in theory, does not work out strictly correct in practice. Actually:

(a) The cane is weighed some time before it enters the mills. In this interval, in particular if it remains too long in the sun at the unloading station, it loses weight, especially at the cut ends. This loss is multiplied with cane mechanically harvested and cut into short pieces.

(b) During the crushing itself, the surface on which the juice is exposed to evaporation is multiplied many times: on the bottom and sides of the feed hoppers and juice trays, on the periphery of the rollers, on the surface (very considerable) of the layer of bagasse on the intermediate carriers, and finally in the streams and trickles of juice falling from the imbibition distributors and the lower rollers.

Further, if the imbibition is hot, evaporation takes place at the spray pipes or the distributing plates, which, in this case, are always covered in a cloud of steam.

The time taken for the cane to traverse the mills is obviously much shorter than its stay on the unloading station, but the enormous multiplication of the exposed surface does much more than compensate for this difference, and effect (b) is much greater than effect (a). Assuming a mean period of 4-8 hours at the unloading platform, and a normal time of passage through the mills, between crusher and pump, we may estimate roughly:

(i) Loss of weight of cane at the platform at less than 1 %

(ii) Loss of weight of juice at the mills at 2-3%.

The former loss is very variable, and depends on the period involved, on weather conditions (day or night, sunshine or rain), on exposure of the cane pile (in the sun or in the shade of the factory), the length of the pieces of cane (canes cut by mechanical harvesters are in short pieces, 0.3 - 0.6 m (1 - 2 ft.) in length, which multiplies by 3, 4 or 5 the number of exposed ends), etc.

In South Africa⁶ a loss of 1.25% in weight of cane on the platform has been found for the first 24 hours.

Loss of water by evaporation of imbibition is fairly low with cold water, but greater with hot; and in the latter case there is a corresponding further loss from the juices which have been heated by mixing with the recirculated imbibition water.

The loss of weight of juice is relatively constant.

In experiments, particularly those made at the College of Agriculture in Trinidad⁷, a figure of at least 3–4% on weight of cane has been found for the 2nd term of eqn. (21.8). In experiments on the industrial scale in Queensland⁸, a loss of 2.06% (say 2%) on weight of cane has been found for cold imbibition. This would correspond to a difference of nearly 10% on the weight of bagasse.

Now the basic equation serves predominantly for determining the weight of bagasse, the other three materials being generally weighed.

The bagasse, for its part, also loses before analysis a small fraction of its weight, generally negligible if the necessary precautions are taken.

We have then:

$$\text{Cane} + \text{water} > \text{juice} + \text{bagasse} \quad (21.9)$$

We shall now analyse this statement of inequality. We shall consider for example the total figures for a week, as follows:

C = weight of cane given by the weighbridge

J = weight of mixed juice given by the juice scales

E = weight of imbibition water given by the water scales

c = weight of cane lost in the yard by evaporation and loss of small pieces

j — weight of juice evaporated at the mills

K = required weight of bagasse actually leaving the mills.

Assuming the scales correct, the equation becomes

$$C - c + E = J + j + K \quad (21.10)$$

Hence:

$$K = C + E - J - (c + j) \quad (21.11)$$

Expressing all the figures per 100 of cane weighed:

$$\frac{100 K}{C} = 100 + \frac{100 E}{C} - \frac{100 J}{C} - \frac{100 (c + j)}{C}$$

Returning to the notation of eqn. (21.8):

$$\bar{B} = \frac{100 K}{C} \quad \bar{W} = \frac{100 E}{C} \quad \bar{Q} = \frac{100 J}{C}$$

we have:

$$B = 100 + W - Q - (\gamma + \zeta) \quad (21.12)$$

where W and Q are the weights obtained at the scales relative to the weight of cane obtained

at the weighbridge

W = weight of cane lost in the yard % on weight of cane received

ξ = weight of juice evaporated at the mills % on cane received.

If we assume:

$$\left. \begin{array}{l} W \text{ at } 0.5 - 1\% \\ \xi \text{ at } 0.5-1\% \text{ (cold imbibition) } \\ \quad \quad \quad 1 - 2\% \text{ (hot imbibition) } \end{array} \right\} \text{ on weight of cane,}$$

the magnitude of the error ($W + \xi$) would be of the order of 2%; but B varies usually between 25 and 30. An error of 1 - 3% in eqn. (21.12) could thus lead to an error of 3 - 10% on the weight of bagasse. However, the general practice is not to make any correction to take into account these losses, and the weight of bagasse obtained is thus in error, being too high.

This error reflects on the fibre figure (if obtained by calculation) and has a slight effect on the sugar content of cane, which is generally obtained by adding sugar in mixed juice (assumed accurate) and sugar in bagasse (which is in error); it also affects the extraction and other figures for the efficiency of the milling train.

This discussion demonstrates the importance of weighing the bagasse. The author did this in one of his factories, and has found only a very slight difference relative to the classical method of calculation. Bagasse may be weighed by continuous integrating scales on the bagasse conveyor (Weiller or Syneyrco), or batch scales such as the Servo-Balans in Holland. The results obtained would be in agreement with the estimation made in Queensland⁹, according to which the evaporation at the mills would represent only 0.15% in the case of cold imbibition or 0.50% with hot imbibition. The former figure is substantially lower than that reported some years later of 2% (cf. p. 333). In South Africa, however¹⁰, an error of 7.3% has been found between calculated and actual weights of bagasse.

As a first approximation, we may take

$$Q = 1 + W - 2f$$

where quantities are expressed per unit (not %) of cane, and fibre in bagasse is assumed as 0.5.

BRIX OF ABSOLUTE JUICE

The brix of absolute juice is an important factor, but rather difficult to determine. The simplest calculation which gives it directly is the following.

Equating the quantity of material in solution entering the mills to that leaving, we have:

$$B_A (1 - f) = Q B_m + B_b \frac{f}{f'} \tag{21.13}$$

B_A = brix of absolute juice

f = fibre per unit of cane

f' = fibre per unit of bagasse
 Q = mixed juice % cane
 B_m = brix of mixed juice
 B_b - soluble matter % of bagasse.

Hence we obtain:

$$B_A = \frac{Q B_m + B_b \frac{f}{f'}}{1 - f} \quad (21.14)$$

Example. Let $Q = 1.0246$ / $f = 0.1346$
 $B_m = 16.96$ / $f' = 0.48$
 $B_b = 2.82$ / $f' = 0.2804$

Hence we have:

$$B_A = \frac{(1.0246 \times 16.96) + (2.82 \times 0.2804)}{0.8654} = 20.99$$

FIBRE

If mixed juice and imbibition water are weighed, the corresponding figure for average fibre in cane may be obtained. The quantity of fibre is substantially constant as it passes through the mills, and we have:

$$100 \times f = B \times f'$$

f = fibre cane
 f' = fibre in bagasse
 B = weight of bagasse % on cane.

Hence, substituting B from the basic equation (21.8):

$$f = f' \cdot \frac{100 + W - Q}{100} \quad (21.15)$$

The study on p. 333 has shown that, if the bagasse is not weighed, the fraction appearing in the second term, which must necessarily be taken into account, will be too high, perhaps by a small amount, perhaps by up to 10%. While it is the figure officially adopted, it is desirable to know its limitations.

VARIOUS RELATIONSHIPS IN MILLING

Relative proportions of juice from feed and delivery rollers

For a mill operating well, it is necessary that the extraction at the feed roller should be at a maximum.

Now, the feed roller extracts first the superficial imbibition water which remains on the upper or exterior portions of the particles of bagasse. Consequently, the juice obtained is of lower brix than that extracted by the delivery roller, which extracts part of the juice enclosed in the inner cells.

The proportion of juice extracted by the feed roller is easily calculated. Let:

E = brix of feed-roller juice

S = brix of delivery-roller juice

M = brix of mill juice

x = proportion of juice obtained by the front roller.

We have:

$$xE + (1 - x) S = 1 \times M$$

whence:

$$x = \frac{S - M}{S - E} \quad (21.16)$$

In Hawaii, it is considered that the work of a last mill is satisfactory if x is of the order of 70–80%.

As a general rule: the feed roller of the later mills, with wet crushing, should furnish about $\frac{3}{4}$ of the juice, and the delivery roller therefore only $\frac{1}{4}$. However, the proportion furnished by the feed roller may be proportionally greater as the imbibition received by the bagasse is increased.

Relation between pol contents of juice and bagasse from a mill

The pol of the bagasse leaving a mill should always be lower than that of the back-roller juice from that mill.

This difference, while marked at the earlier mills, decreases greatly at the later mills; but, if these mills are properly set, the pol of the back-roller juice of the last mill (last expressed juice) should remain appreciably higher than that of the bagasse.

If the opposite is found, it indicates that the mixing of imbibition water with the bagasse leaves much to be desired.

The fibre in bagasse is approximately 50%; hence, if the juice which it retained had the same composition as the juice extracted, we should have, since all the sugar retained by the bagasse is present in the juice which it contains:

$$100 a = 50s, \quad \text{or:} \quad a = \frac{s}{2} \quad (21.17)$$

a = pol % in bagasse

s = pol % juice extracted by the last roller.

Such a result is never obtained. Generally a is of the order of 0.7 s ; but there is no need to tolerate performance where it is equal to s or greater.

Sequence of purities of successive juices

The first pressure gives the highest-purity juice. As we follow through the successive pressures, the juice extracted from the inner cells is of lower and lower purity.

Since the juice of lowest purity is the most difficult to extract, it follows that the drop in purity from crusher to last expressed juice measures in a way the degree of pressure of the mills.

It increases with the extraction and gives an indication of the extent to which milling work has been pushed. This drop is of the order of 10–18 points.

Correlation between extraction and last-mill juice purity

It may be asked whether the purity of the last-mill juice will not fall more and more rapidly as the extraction approaches 100%, for a given fibre figure. Now Haines and Hughes have shown" that it tends towards an asymptotic limit (Fig. 21.3); this is encouraging when an improvement in extraction of a tandem is sought.

It may indeed be asked whether the fall in purity of the increment of juice obtained would not neutralise the gain in extraction sought. Reasoning from the graph in Fig. 21.3, we see that a gain of 1% in extraction (*e.g.* 95—96%) would give a juice which we may assume,

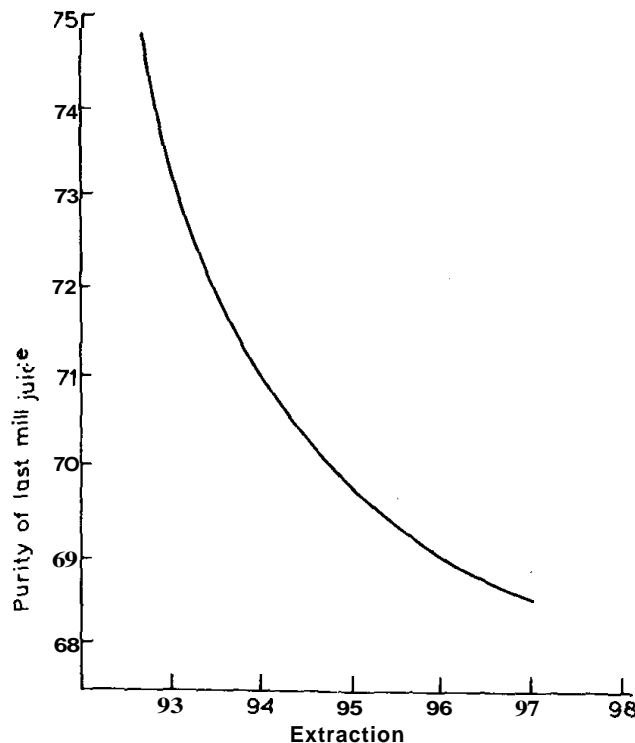


Fig. 21.3. Variation of last-mill juice purity with extraction.

at the worst, as of 69 purity, assuming that it was obtained entirely from the final mill. If the factory obtains a molasses purity of 40, the overall gain would be (eqn. (39.1)):

$$1 \times \frac{100(69 - 40)}{69(100 - 40)} = 0.70$$

or 70% of the gain obtained at the mills.

Although this purity drop has a detrimental effect, it should not discourage the attempt to obtain an improved extraction.

South Africa¹² has found the relationship:

$$\frac{\text{Purity of cane}}{\text{Purity of mixed juice}} = (0.0053 \times \text{extraction}) + 0.46$$

using apparent purities. For diffusion factories, the ratio becomes:

$$\frac{\text{Purity of cane}}{\text{Purity of mixed juice}} = (0.0042 \times \text{extraction}) + 0.58$$

Purity of residual juice

This progressive drop in purity would continue to show up if the final bagasse were passed on to a further mill. Actually, the drop in purity between juice and bagasse at the last mill is very considerable, and the purity of residual juice is of the order of 45 or 50 in general, rarely being as high as 60; for this reason, it should not be replaced by the purity of back-roller juice of the last mill, which is much higher.

Java ratio

This expression denotes the quotient:

$$\text{Java ratio} = 100 \times \frac{\text{pol \% cane}}{\text{pol \% 1st expressed juice}} \tag{21.18}$$

It aims to deduce the pol in cane by a simple multiplication, knowing the pol of crusher juice. It is utilised with this object in certain countries where the price of cane to the growers is based on its sugar content.

Actually, this ratio represents:

$$J.R. = \frac{S(1 - f)}{S_1} \tag{21.19}$$

S = pol % absolute juice

S₁ = pol % of 1st expressed juice

f = fibre per unit cane.

Its value varies from 77 to 90, for different factories and different countries. It obviously

depends essentially on the fibre content of cane. Innes¹³ gives the value:

$$J.R. = 98.5 - 1.34 f \quad (21.20)$$

Kerr¹⁴ interprets this equation as indicating that the difference between 98.5 and 100 could be due to the water of constitution of the cane ("brix-free water"), which is of the order of 1.5% on cane; and that a certain proportion of the "hygroscopic" water, equal to 34% on fibre, would remain attached to the fibre, and would not be extractable.

It appears that this ratio has furnished a rough but practical method for cane payment according to sugar content, and that there is no reason to expect anything more of it.

Hawaii ratio

This is the quotient:

$$k = \frac{\text{brix of absolute juice}}{\text{brix of 1st expressed juice}} \quad (21.21)$$

This ratio is useful for calculating easily the sugar in cane in factories where neither imbibition water nor mixed juice are weighed (see first edition, p. 259).

It is generally of the order of 0.96-0.97 but, unfortunately, it also is rather variable from one factory to another; and in the one factory, from one variety to another, though fortunately to a smaller extent.

It is readily determined by a test with dry crushing.

Arceneaux¹⁵ has found in Louisiana that the fibre content of cane had an influence on the coefficient k . After numerous observations carried out on several varieties of cane, he has reported figures which, except for slight variations peculiar to certain varieties, allow the following equation to be used with a good degree of approximation:

$$k = 1 - 0.002 / \quad (21.22)$$

/ = fibre % cane.

Innes¹⁶ gives:

$$k = 1.003 - 0.004 / \quad (21.23)$$

an expression which, as will be seen, diverges markedly from that proposed according to Arceneaux, and furnishes smaller values of k . The two authors, however, are in agreement in indicating a decrease in the coefficient k with an increase in fibre.

Imbibition and dilution

It is important to avoid confusing these two terms as is sometimes done.

Imbibition is the total quantity of water added at the mills, per 100 of cane.

Dilution is the quantity of imbibition water which enters the mixed juice, per 100 of cane.

The difference between these two figures represents the quantity of imbibition water remaining in the bagasse, per 100 of cane.

It is of interest to note that, for normal values of imbibition, the quantity of water remaining in the bagasse varies but little. It generally lies between 8 and 12. The remainder of the imbibition passes into the mixed juice. When the imbibition is increased, a weight of diluted juice approximately equal to that of the additional imbibition passes into the juice, and it is this quantity of "mixed juice % cane" which absorbs almost wholly the variations of imbibition.

Shannon^{16a} (personal communication) considers that the ratio of "dilution % primary juice" to the "imbibition" should be between 0.95 and 1.05. If the ratio is below 0.95, the imbibition figure is too high; if above 1.05, the imbibition figure is too low.

Example:

$$\left. \begin{array}{l} \text{Brix of primary juice} = 19.16 \\ \text{Brix of mixed juice} = 14.98 \end{array} \right\} \begin{array}{l} \text{Dilution \% primary juice} \\ = 100 \times \frac{19.16 - 14.98}{14.98} = 27.90 \end{array}$$

$$\text{Imbibition} = 29.06 \qquad k = \frac{27.90}{29.06} = 0.96$$

SUCROSE EXTRACTED % CANE

We have:

$$c = 100 (R - \beta)$$

c = sucrose extracted % cane

R = sucrose content of cane

β = sucrose in bagasse % cane.

Or:

$$c = 100 \left(R - \sigma \frac{f}{f'} \right) \qquad (21.24)$$

a = sucrose % bagasse

f = fibre % cane

f' = fibre % bagasse.

Or:

$$c = 100 f \left(\frac{R}{f} - \frac{\sigma}{f'} \right)$$

Evaluation of cane

In a sugar factory, it is necessary constantly to determine the value, in terms of sugar, of various batches of cane, either to determine the sugar content of one variety or of a field and hence its sugar content per acre, to analyse the various parcels of cane from field experiments, or to determine the quantity of sugar in the cane from each of the growers or from cooperative groups, etc.

Recoverable sugar

One requirement must be first realised: the interesting thing to determine is not the sugar content proper (that is, the quantity of sucrose contained in 100 parts of cane) but the *recoverable sugar*, in other words, the quantity of sugar or of sucrose which the factory will be able to recover from 100 parts of the cane under consideration. As we shall see later, two canes of the same sugar content do not give actually the same yield of sugar, if their fibre contents or the respective purities of their juices are different.

We shall attempt then to determine the yield of sucrose from a given cane. Let:

R = sugar % of cane under consideration

S_0 = sucrose % absolute juice

f = fibre per unit of cane

X = sugar recoverable from the cane

B = brix of primary juice furnished by the 1st mill (or by the combination of crusher and 1st mill)

S = pol % primary juice

P = apparent purity of primary juice = $(S/B) \times 100$

P' = Clerget purity of mixed juice obtained from this cane

M = usual Clerget purity of the molasses from the factory (assumed obtainable with the cane under consideration).

The sugar entering the factory, % on cane, is:

$$R = S_0 (1 - f) \quad (21.25)$$

Being unable to extract the absolute juice, we must be content with analysing the primary juice, or dry-crushing juice, but we know that the sucrose % primary juice bears the ratio k , approximately constant and approaching 1, to the sucrose % of absolute juice. We may therefor write:

$$R = S_0 (1 - f) = k_1 \cdot S (1 - f) \quad (21.26)$$

k_1 = coefficient slightly less than 1, and relatively constant.

The factory recovers only a fraction of R . Assuming that the losses in muds and the undetermined losses are small, we may introduce a second coefficient k_2 to take these two small losses into account. It remains to calculate the two major losses: loss in bagasse and loss in molasses.

(1) **Loss in bagasse.** Let:

O = sugar lost in bagasse % cane

a = sucrose % bagasse

f' = fibre per unit of bagasse.

We have:

$$\beta = \sigma \frac{f}{f'} \quad (21.27)$$

If we require the value of this loss corresponding to a standard extraction e_0 (for example, 95%) and for a standard fibre f_0 (for example, 0.125), we should have:

$$\frac{\beta}{R} = \frac{\sigma}{f'} \cdot \frac{1}{R} = 1 - e, \quad \text{hence} \quad \frac{\sigma}{R} = \frac{f' (1 - e)}{f} \quad (21.28)$$

and we require:

$$\frac{a}{R} = \frac{f' (1 - e_0)}{f_0}$$

The following must then hold:

$$\frac{O}{R} = \frac{f' (1 - e_0)}{f_0} = \frac{(1 - e_0) f}{f_0}$$

The sucrose obtained after the milling plant will be:

$$k_1 \cdot S (1 - f) k_2 \left[1 - \frac{(1 - e_0) f}{f_0} \right] \quad (21.29)$$

It is necessary to choose standard values for f_0 and e_0 . We shall take $f_0 = 0.125$, the classical and general figure, and $e_0 = 0.975$. The latter figure is chosen on account of the preference for basing the calculations on ideal or maximal values, and from the fact that certain milling tandems approach or exceed 97%, and since diffusion makes such extractions readily attainable.

Hoarau¹⁷ has shown in Reunion that k_1 is a function of f_0 and he has given the range of variation for all the factories in the island. The good correlation between the graph obtained and the individual values found for widely different tandems and factories allows the results to be generalised and expressed with very good precision by the formula:

$$k_1 = 1 - 0.57/f_0$$

Substituting the values of k_1 , f_0 and e_0 in eqn. (21.29), we have:

$$k_2 \cdot 5(1 - 0.57 f) (1 - f) (1 - 0.2 f)$$

or:

$$k_2 \cdot S (1 - 1.77 f + 0.884 f^2 - 0.114 f^3)$$

The terms in f^2 and f^3 have little effect. If we retain only the term in f and modify it in order to give it the same value as the whole polynomial in f would have for a mean value of f , the sucrose obtained after the milling plant becomes:

$$k_2 \cdot 5(1 - 1.65 f)$$

and it is readily shown that the approximation thus adopted involves an error not exceeding 0.4% (say 4 parts per 1,000) for extreme values of f such as 0.10 and 0.16.

(2) **Loss in molasses.** If we take a standard purity M for the final molasses, the final quantity of sucrose recovered from the cane under consideration will be (cf. eqn. (39.1):

$$X = k_2 \cdot 5(1 - 1.65 f) \frac{100 (P' - M)}{P' (100 - M)}$$

or:

$$X = k_2 \cdot \frac{100}{100 - M} \cdot S (1 - 1.65 f) \frac{P' - M}{P'}$$

But:

$$S = B \frac{P}{100}$$

Hence:

$$X = k_2 \cdot \frac{B}{100 - M} (1 - 1.65 f) P \cdot \frac{P' - M}{P'}$$

Now:

$$P \cdot \frac{P' - M}{P'} = P - \frac{P}{P'} M$$

If we choose a standard purity of molasses M such that:

$$M \frac{P}{P'} = 30$$

which corresponds closely to $M = 29$, a value very close to the classical figure $M = 28.57$, we shall have:

$$X = k_2 \cdot \frac{100}{100 - M} \cdot B (1 - 1.65 f) \frac{P - 30}{100}$$

Putting:

$$k_2 \frac{100}{100 - M} = k$$

and commenting that:

$$B \cdot \frac{P - 30}{100} = \frac{BP - 30 B}{100} = S - 0.3 B \quad (21.30)$$

we then have:

$$X = k (1 - 1.65 f) (S - 0.3 B) \quad (21.31)$$

k is the efficiency coefficient for the factory. It varies with the extraction at the mills and with the recovery in the boiling house. However, it does not measure rigorously the efficiency of the factory since it depends also on:

(a) *The proportion of primary juice obtained.* When the primary juice is derived from the combination of crusher and 1st mill, or indeed from a particularly effective 1st mill, this proportion is high; the primary juice consequently approaches the absolute juice (k_1 approaches unity) and the coefficient k is higher. When the primary juice represents only a small fraction of the absolute juice (as in the case of a low extraction at the 1st mill), the coefficient k is slightly lower. This is equally the case when the first expressed juice is sampled (crusher juice, or front-roller juice of the 1st mill) instead of the primary juice. This practice moreover is not to be recommended, since the proportion of first juice extracted depends to a much greater extent on the instantaneous volume of cane supplied to the mill (in other words, on whether the mill is more or less loaded at the moment of sampling), and its sugar content varies in proportion; it is then more variable than that of the primary juice relative to the sugar content of the absolute juice.

(b) *The cane preparation.* The phenomenon just discussed is in fact greatly accentuated by the degree of preparation of the cane. Munro¹⁸ has obtained the graphs given in Figs. 21.4

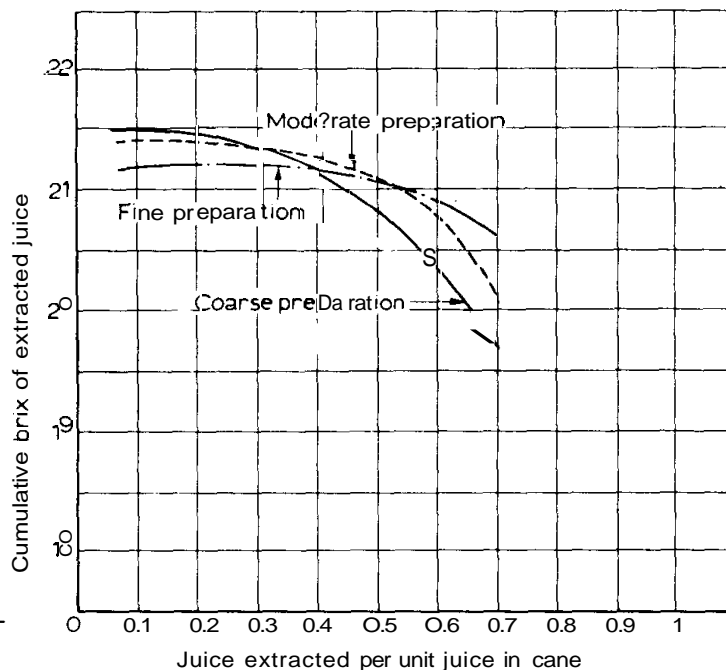
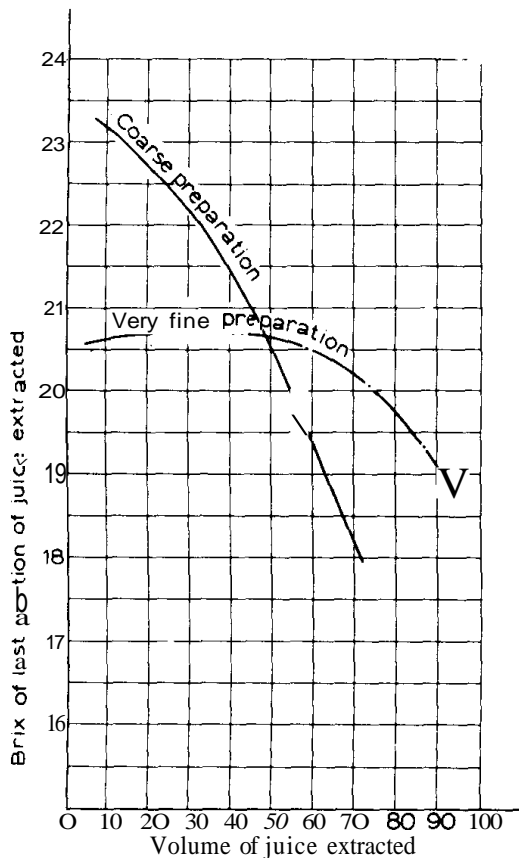


Fig. 21.4. Brix of extracted juice relative to preparation and proportion of juice extracted.

Fig. 21.5. Brix of primary juice as a function of proportion extracted.

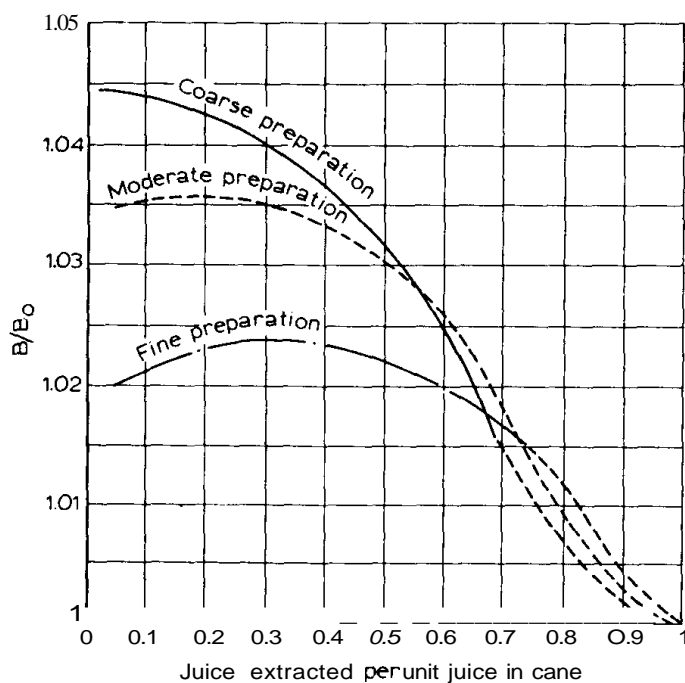


Fig. 21.6. Ratio B/B_0 of brix of primary juice to brix of absolute juice, as a function of cumulative proportion of primary juice extracted.

and 21.5, which demonstrate the effect of fineness of preparation on the variations in brix of juice extracted. In Fig. 21.4, the ordinates represent the brix of the last portion of juice extracted, for each supplementary fraction obtained under the press. In Fig. 21.5, they represent the mean brix of the whole of the juice already extracted; these are the curves which are of interest here. In Fig. 21.6, they represent the ratio B/B_0 of the brix of the fraction of juice extracted to the brix of absolute juice. This shows the very marked influence of 1st-mill extraction on the brix of primary juice, for the same cane or for identical canes, and emphasises the caution which should therefore be used when comparing the primary juices of two different tandems. It will be noted also that the difference due to preparation decreases substantially for a proportion in the neighbourhood of 60–70% of primary juice; hence the advantage of analysing the primary juice rather than the first expressed juice, which can only with difficulty amount to more than 50% of the absolute juice, and often amounts to only 40%.

This equalising effect on preparation is readily understood when it is realised that the cells of the rind and the nodes contain the juice of lowest brix and of lowest purity; hence incomplete preparation leaves them to a great extent intact, in which case they retain their juice when the first pressure is applied; whereas more intensive preparation breaks the cells and they can thus release their juice with the first application of pressure.

We may comment that both effects (a) and (b) are still more marked on the purity than on the brix, which accentuates their effect on the coefficient k .

(c) *The exhaustibility of the molasses* (cf. p. 678). If the ratio reducing sugars/ash is high, the factory will more readily obtain a low-purity molasses and consequently a high value of k ; if this ratio is low, it will have to be content with a relatively high molasses purity, which will give it a relatively low coefficient.

(d) *The method of expressing recoverable sugar*. For determination of sugar content of growers' cane, the recoverable sugar is generally reckoned not in sucrose but in sugar "tel quel" or standard (94 net titre, 96° polarisation, etc., according to custom and the country concerned). In this case, the coefficient k will be higher according as the sugar is reckoned at a lower polarisation or titre.

DETERMINATION OF THE VALUE OF THE COEFFICIENT k

For the coefficient k the mean value obtained during the previous season is adopted, or the "to date" results for the current season. If q is the average yield (in "tel quel" sugar, in 96° polarisation sugar, or in sucrose, according to the figure used) for the season, we have:

$$k = \frac{q}{(1 - 1.65f)(S - 0.3B)} \quad (21.32)$$

the values of f , S and B being the general averages for the factory for the season or the period under consideration.

NORMAL VALUES OF EFFICIENCY COEFFICIENT k

Assuming a sugar of 99° polarisation, a very good extraction and a molasses of high exhaustibility, a factory should theoretically obtain, relative to typical values of fibre, brix and sucrose % primary juice as furnished by its chemical control, an efficiency coefficient k of the order of 1.46; this figure assumes no losses in muds and no undetermined losses. Such a figure, however, is to our knowledge never attained, and most well-conducted factories maintain a figure between 1.38 and 1.42. Many others remain in the neighbourhood of 1.35.

The value of k deduced from the factory control ("factory k ") must not be confused with that obtained by analysing fresh cane ("planters' k ") as obtained from analyses of cane from the fields, of experimental lots, or of planters' cane for payment purposes. Fresh cane, which has not been stored in the yard, is in fact always superior, other things being equal, to canes of the same origin which have spent some hours in the yard. For factories having a relatively well-controlled cane supply, but for 12 hours a day only, and consequently having to make provision for overnight supply, when they crush for 24 hours per day, it is necessary to reckon a planters' coefficient k_p (fresh cane) about 2 points lower than the factory k_u furnished by the chemical control (*e.g.* 1.38 instead of 1.40). The average k for a factory during a season, calculated from the analysis of fresh cane from all its suppliers and its growers, is thus different from the k obtained from the general average figures of the chemical control, and appreciably lower. The difference originates to a small extent from the increase in fibre content during storage, but to the major extent from the drop in purity; the brix remains appreciably unchanged.

USE OF THE COEFFICIENT

We strongly recommend the use of the coefficient k . It is the best means of measuring the value of any batch of cane to the factory and of relating it to real sugar produced. It is also the simplest and easiest means of ascertaining the sugar produced per acre by each variety in each field under the control of the factory, or the sugar per tonne of cane from any outside supplier.

SPECIAL USE FOR FACTORY CONTROL

It is interesting to make use of another means of determining k , to assess the work of the factory. This second method is applicable only to the average performance of the week or of the season. It consists of applying the same formula (21.32), but by writing:

$$k_c = \frac{100 \varrho_G}{(1 - 1.65/) B (P' - 30)} \quad (21.33)$$

in which B is indeed the brix of primary juice, but P' is the Clerget purity of the mixed juice and ϱ_G the yield in E.S.G. (*cf.* p. 842). This equation avoids the uncertainties regarding

purity of primary juice (difference $P' - P$ variable) and of sugar quality, and the only uncertainty remaining is that of brix of primary juice (*cf.* Fig. 21.6).

For factories having a complete chemical control, we may advance a step further towards precision by calculating:

$$k_A = \frac{e_G}{(1 - 1.65 f) (S_A - 0.3 B_A)} \quad (21.34)$$

where S_A and B_A are the calculated values of sucrose and brix of the absolute juice. The k_A thus determined gives, more simply and rapidly, a better criterion of the general reduced recovery ($e_{12.5} \cdot r_{85}$ - *cf.* eqn. (39.14)). Good operation then gives a value of k_A lying between 1.47 and 1.53.

Comment. We may be tempted to replace the factors 1.65 and 0.3 in eqn. (21.31) by particular figures corresponding to the actual extraction and the actual molasses purity obtained by the factory. The value of k obtained by an analysis of plantation cane or suppliers' cane would then theoretically be more accurate. In practice, the accuracy of the analysis would not justify this, improvements in the factory would necessitate changing the factors each

TABLE 21.3

FORMS AND USES OF FACTORY COEFFICIENT k

	<i>Symbol</i>			
	k_P	k_U	k_C	k_A
<i>Cane</i>				
Cane for analysis	Batch of fresh cane	Average of cane entering mills		
<i>Bases</i>				
Brix to be determined	Primary juice (batch for analysis)	Primary juice (mean for week or season)		Absolute juice (mean for week or season)
Juice to be polarised or calculated (for pol or purity)	Primary juice from batch (pol or app. purity)	Primary juice (mean pol or app. purity)	Mixed juice (mean Clerget pty.)	Absolute juice (mean Clerget pty.)
Fibre	Direct analysis of batch	Calculated figure	Calculated figure	Calculated figure
Recovery	Raw sugar	Raw sugar	E.S.G.	E.S.G.
<i>Use</i>	Evaluation of varieties or fields Payment for cane	Basis of cane payment	Chemical control (factory performance)	Chemical control (factory performance)
<i>Normal values</i>	1.32 - 1.40	1.34-1.42	1.36-1.44	1.40 - 1.53

year, and coefficients for the factory for different periods, or coefficients for it and other factories, would no longer be comparable. It is much preferable to use a standard and general formula, for which tables for the two values concerned may be used, giving without calculation the value of the term $k (1 - 1.65 f)$.

Example. If we take a factory giving average values for a season of:

f	= fibre in cane	0.134
B_A	= brix of absolute juice	17.36
S_A	= sucrose % absolute juice	14.54
q_G	= yield in E.S.G.	10.82,

its efficiency coefficient has a value:

$$K_A = \frac{10.82}{[1 - (1.65 \times 0.134)] [14.54 - (0.3 \times 17.36)]} = 1.4886$$

This same factory, operating on fresh cane, obtains a mean "farmers' k " of 1.390. What sugar yield could it obtain from a field which gave an analysis of cane and primary juice of the following values:

$$\begin{aligned} f &= 0.136 \\ B &= 21.40 \\ S &= 18.83. \end{aligned}$$

We have:

$$X = 1.390 [1 - (1.65 \times 0.136)] [18.83 - (0.3 \times 21.40)] = 13.38$$

Summary. The use of the factory coefficient k is very useful, but it is necessary to distinguish clearly between its several forms, as given in Table 21.3.

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22. Fine-Bagasse Separators

Many particles of bagasse drop from the mills, falling through the space between feed plate and feed roller, or being extracted from the Messchaerts by the scrapers, or dropping between the trashplate and delivery roller.

The quantity of such fine bagasse is very variable, but generally amounts to between 1 and 10 g of dry material per litre of juice (taking into account, obviously, only the fine bagasse in suspension, and not large pieces).

The fine bagasse is known as “bagacillo” or “cush cush”, and the separator is an apparatus placed after the mills, serving to screen the mill juices, and to return to an intermediate carrier the pieces of bagasse recovered.

Juice trays

To avoid the accumulation of such bagasse, the sides of the juice pan should have a slope of at least 55°.

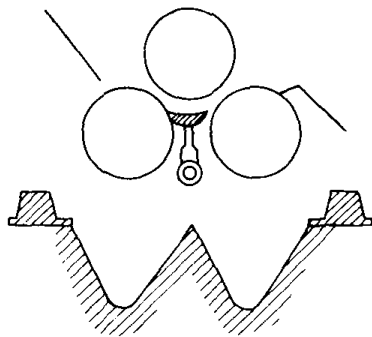


Fig. 22.1. Double transverse juice trays.

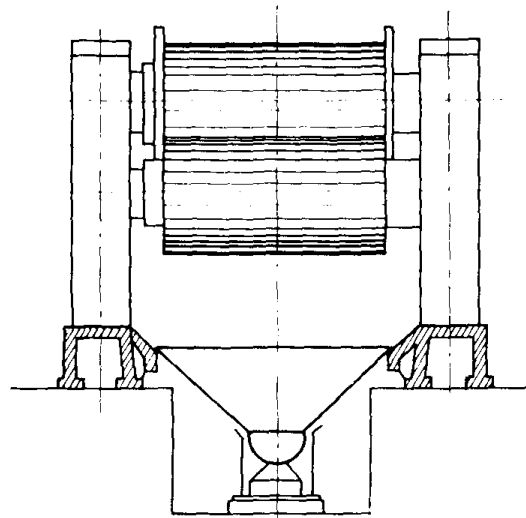


Fig. 22.2. Single longitudinal juice tray.

There are 3 principal arrangements of these juice trays:

- (1) Individual trays for each mill, flat, and in one piece; now obsolete
- (2) Double transverse trays under each mill (Fig. 22.1), permitting front- and back-roller juices to be separated
- (3) A single longitudinal tray running the length of the tandem (Fig. 22.2). The juices from the various mills are then separated by small, removable weirplates.

The gutters conveying the juice to the separator should have a fall of 1 – 2‰.

Screens

The fine-bagasse separator is a strainer or screen often termed the "cush-cush screen" (Fig. 22.3).

When compound clarification is practised, it is necessary to keep quite separate the part of the screen reserved for the primary juice. In long tandems, one screen may be kept for the primary juice, and another installed for the secondary juice and the imbibition juices.

In any case, compound imbibition makes it necessary to separate the compartments into which the juice from the later mills is delivered. Each of these compartments should be able to discharge its overflow into the compartment corresponding to the juice from the preceding mill; when one mill feeds badly and chokes, the common practice is to decrease the imbibition to that mill, and it is thus necessary, with compound imbibition, to divert the juice being used as imbibition.

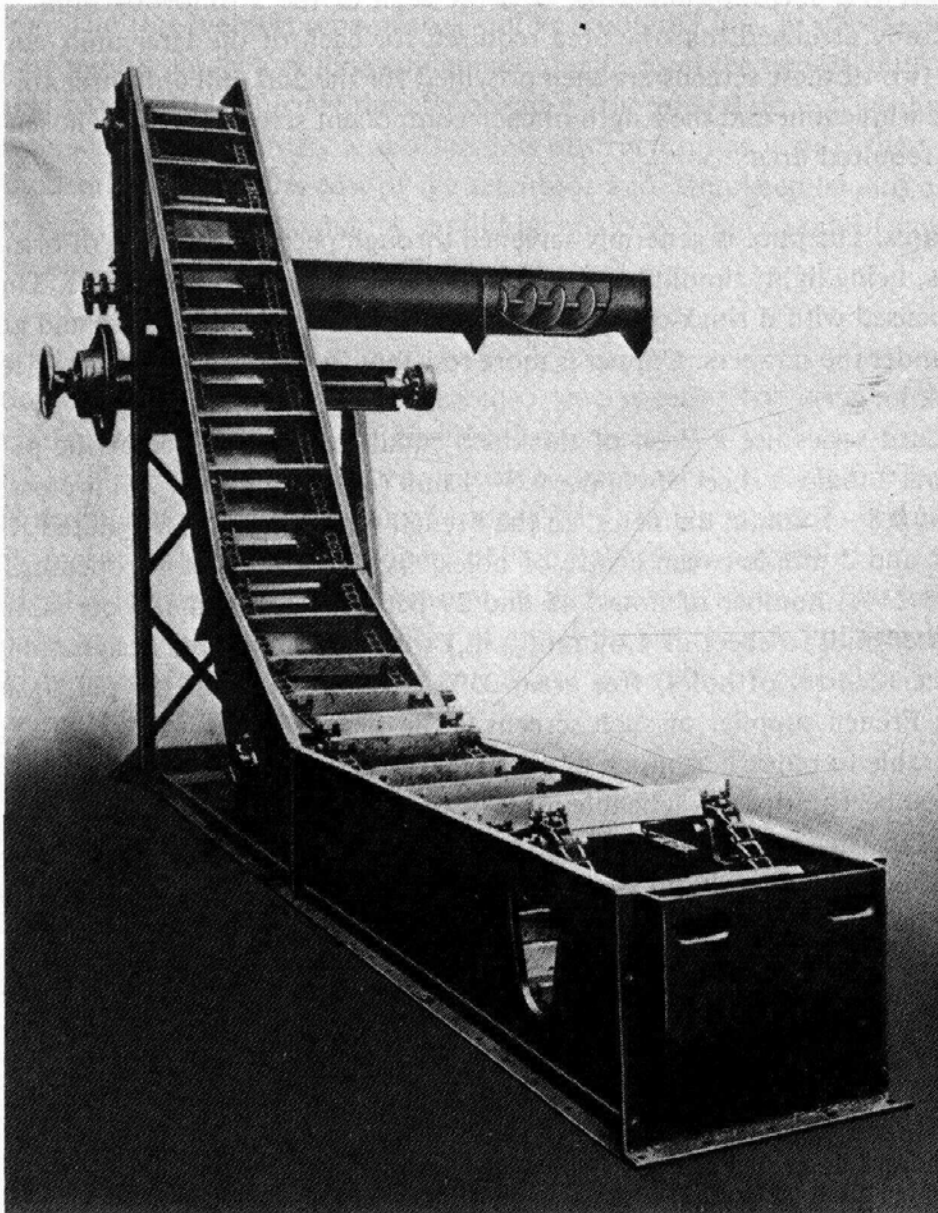


Fig. 22.3. Combined screen and elevator for fine bagasse.

Area. Generally, an area of $0.05 - 0.1 \text{ m}^2$ ($0.5 - 1 \text{ sq.ft.}$) of perforated screen per t.c.h. is provided. It is of advantage to approach the higher figure, *i.e.* 0.1 m^2 per t.c.h., or better, 0.03 m^2 per t.c.h. plus 0.01 m^2 per 3-roller mill (the crusher not being counted).

This will give:

0.07 $\text{m}^2/\text{t.c.h.}$ (0.7 sq.ft./t.c.h.) for a 12 – 14-roller tandem

0.08 $\text{m}^2/\text{t.c.h.}$ (0.8 sq.ft./t.c.h.) for a 15 – 17-roller tandem

0.09 $\text{m}^2/\text{t.c.h.}$ (0.9 sq.ft./t.c.h.) for a 18 – 20-roller tandem

This area will be distributed between the mills in the proportions: 3 for the 1st mill, 2 for the 2nd, and one for each of the following mills.

For example, a 15-roller tandem would have $0.03 \text{ m}^2/\text{t.c.h.}$ ($0.3 \text{ sq.ft./t.c.h.}$) for the 1st mill, 0.02 (0.2) for the 2nd, and 0.01 (0.1) for each of the 3 following mills.

It is generally arranged that the area **required** for each of the later mills corresponds to one screen; two of these screens are then provided for the 2nd mill and three for the 1st. The width of the whole unit and the length of each component screen are then chosen accordingly to give the required area.

Screen plates. The juice is generally screened through perforated sheets of brass or copper. These sheets, being light, should be supported by a rigid **checkerwork** frame. This, however, may be dispensed with if thick copper plates are used. Brass is more rigid and gives less risk of sagging under the scrapers. Copper is more resistant to the reaction of the juice and consequently lasts longer.

The standard series use a sheet of thickness equal to the diameter of the perforations.

For the first 2 mills, a sheet of copper $0.8 - 1 \text{ mm}$ ($\frac{1}{32} - \frac{1}{23} \text{ in.}$) thick will preferably be used, with holes of $0.8 - 1 \text{ mm}$ in diameter. In the French series they are designated RD 0.8 S and RD 1 S; 1.7 and 2 mm between centre of holes; free area, or area of openings % of total area, 20 and 23%; number of holes, 45 and 29 per cm^2 (289 and 196 per sq.in.).

For the later mills, a sheet of 1.6 mm ($\frac{1}{16} \text{ in.}$) with 1.6-mm holes; designation, RD 1.6 S; 3 mm between centres of holes; free area, 25% (13 holes per cm^2 (81 per sq.in.)).

The main French supplier of such screens is the firm of Krieg and Zivy.

It is preferable to remove as much as possible of the fine bagasse carried by the juice. For this, the finest perforations practicable must be used. In Java, with the precautions and control measures insisted upon at the mills, use of perforations of 0.4 mm ($\frac{1}{64} \text{ in.}$) was achieved¹. This is the smallest value obtainable with an ordinary screen.

The perforated sheets will be ordered in the form of rectangles to suit the frames of the separator in which they are to be used; it will be desirable to specify that they should be made with a non-perforated border of 25 mm (1 in.) along the sides of the sheets and 50 mm (2 in.) at the ends adjoining the neighbouring plates.

It is preferable to fix each screen on a frame which can be introduced or withdrawn by sliding laterally, so that the new frame pushes the other one out. In this way screens may be removed and cleaned without stopping the scrapers.

Chains. The elevator chains, working in acid juice, are subject to severe wear. It is well

to examine them every year, to renew them frequently, and especially to choose a well-designed make.

The scrapers should be of wood or rubber.

Precautions. The crush equipment is one of the items of plant where contamination is most to be feared. If it is neglected, it happens at times that the outlets become blocked with "frog spawn" (*Leuconostoc mesenteroides*).

Cleanliness of the equipment should therefore receive careful attention; it should be steam-cleaned at least twice per day, and cleaned completely each week at the same time as the mills.

For this operation, it is convenient to have the screen arranged so that the whole framework may be lifted by one end, leaving the juice tank fully accessible.

Application. The fine-bagasse separator is still encountered but, in new factories and in those which are modifying their milling equipment, the juices of the various mills are handled by unchokable pumps and returned directly ahead of the preceding mill in the train. This system is much simpler, less costly, and offers less obstruction, as it frees the whole of one side of the tandem, and saves the cost of the separator and its upkeep (chains and screens).

Juice supply to the unchokable pumps should be made by a juice tank of vortex type, to avoid trouble and losses of priming.

Secondary screening

The separation obtained by the crush screens is coarse only. The juice is often screened a second time before being sent to the clarification process.

The material still remaining in the juice consists predominantly of very fine bagasse (since it has passed through holes of the order of 1 mm ($\frac{1}{25}$ in.) in the separator screens). Strictly, it is only this material that merits the name of fine bagasse or bagacillo.

To separate this bagacillo still in suspension, it is necessary to use a fine metallic cloth. If it is arranged as a rotating screen, an area of about 0.1 m² (1 sq.ft.) of metallic cloth per t.c.h. should be provided. These woven screens are generally of phosphor bronze, Nos. 140-160 for preference.

Oliver D.S.M. screen

Like the vibrating screen, this type may serve equally well as a 2nd screen or as a 1st and single screen (Fig. 22.4).

The screen is formed of horizontal prismatic bars set close together, leaving an interval between them of 1 or 0.7 mm ($\frac{1}{25}$ or $\frac{1}{36}$ in.), and placed in a 45° arc of a circle, the tangent to which at its upper end is vertical and the length of which along the curve is 1.6 m (5 ft. 3 in.). The screen is made in 6 widths, 91, 122, 152, 183 and 213 cm (3, 4, 5, 6, 6.5 and 7 ft.). The juice is distributed at the top by means of an overflow box, and flows along the arc. The suspended particles are retained on the screen and collected at the bottom, while the juice passes between the bars.

The bars are made either of stainless steel, which has much better life, or "everdur" bronze, which is preferable where there is a risk of contamination, as bacteria grow only with

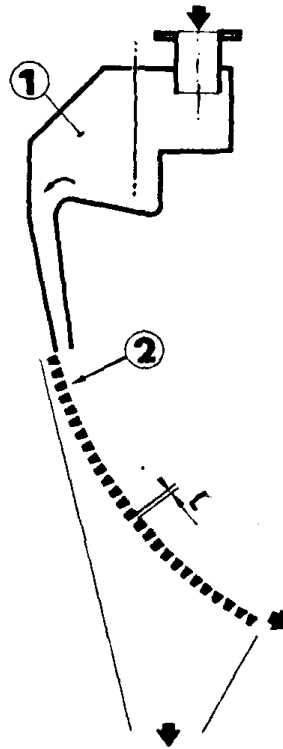


Fig. 22.4. Principle of D.S.M. screen.

difficulty in the presence of this metal. The overflow box and the juice receiver behind the screen are in stainless steel.

A choice of two spacings for the bars is available, 1 and 0.7 mm ($\frac{57}{32}$ and $\frac{1}{32}$ in.). The former is employed mostly for a first screening, and the screen may then be installed above an intermediate carrier; in this case a screen is chosen of 15 or 30 cm (6 or 12 in.) narrower than the carrier, in order to give a better distribution of the fine bagasse on the carrier.

The capacity of the screen with 1-mm spacing is 7,500 litres of juice per dm of width (23 t.c.h. per ft.). With the narrow spacing, the capacity is 80% of that figure, say 6,000 litres per dm (18 t.c.h. per ft.). Certain factories, however, report appreciably higher capacities for the 1-mm spacing, and count on 100 t/h of juice per m (30 t/h per ft.) of width.²

This screen has the advantage of being more effective than those of the crush crush type, and of having no moving parts, and thus avoids wear and the disadvantages of vibrating screens. They leave about 0.8 — 1.5 g of fine bagasse per litre of juice (with 1-mm spacing), or 0.7 — 1.2 g with 0.7-mm spacing.

The screen is cleaned by means of a jet of hot water delivered by a perforated pipe placed under the overflow box. Cleaning every 8 hours is recommended.

It is considered in **Australia**³ that factories having a careful preparation of cane should not utilise screens with openings larger than 0.8 mm ($\frac{1}{32}$ in.) if they wish to maintain the fibre content of clarified juice below a maximum of 0.2%.

To prolong the life of the screen and to improve its efficiency, it is of advantage⁴: (1) to pay attention to this sanitation; (2) to reverse the bars end for end when they become worn; (3) to sharpen the edges of the bars from time to time.

The D.S.M. screen may also be used for screening clear juice after clarification⁵. It is then installed with 0.35-mm ($\frac{1}{72}$ -in.) clearance between bars. In short, it has 3 applications:

- (a) As 1st screening, with 1-mm spacing
- (b) As a 2nd screen, with 0.7-mm spacing
- (c) As a clarified-juice screen, with 0.35-mm spacing.

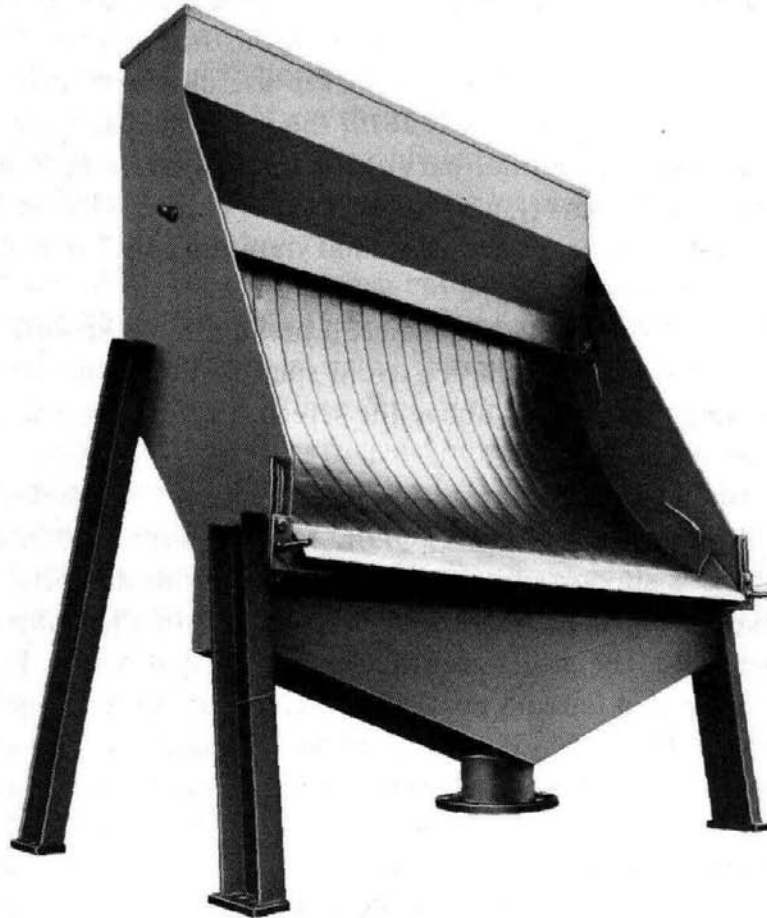


Fig. 22.5. B.M.A. - D.S.M. type curved screen.

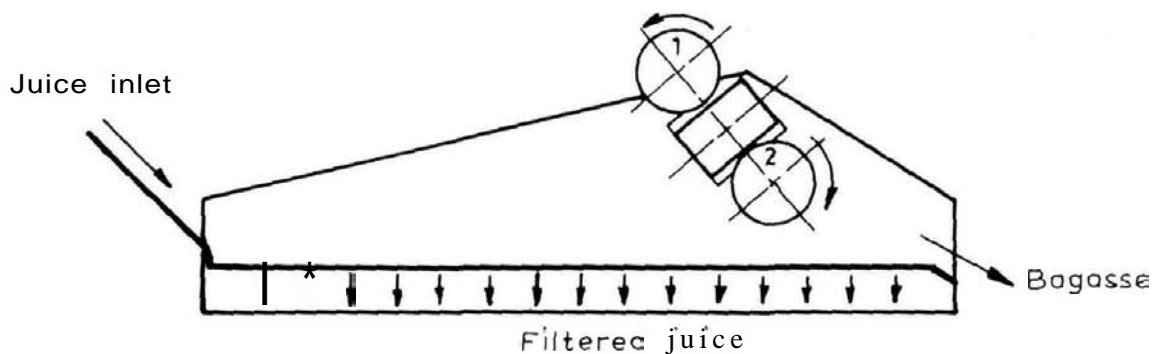


Fig. 22.6. "Vibro-crible" filtering and screening set with vibrating motors (1.250 x 3.000 m).

Vibrating screens

The second screening may also be effected by means of a vibrating screen. Vibrating screens today are generally arranged as single screens to perform the whole screening operation.

The vibrating screen permits of a reduction in the proportion of bagacillo in screened juice to 0.4 or 0.5 g per litre, compared with 0.6—1.5 for fixed screens.

Numerous types of vibrating screen are in use. The most interesting, to the author's knowledge, is the "Vibro-crible", manufactured by Fives Cail—Babcock. It consists of a framework (Fig. 22.5) set in vibration by two out-of-balance motors rotating in opposite directions and placed above the frame. The floor on this frame forms the screen surface; it measures 1.25 m (4 ft.) in width and 3 m (10 ft.) in useful length. It is of stainless steel, 6 mm ($\frac{1}{4}$ in.) in thickness, designated Z2 CN 1810; the grilles similarly are in stainless steel of 17% chromium. They take the form of bars of triangular section, apex downwards, leaving spaces between them of 0.75 mm ($\frac{1}{32}$ in.). The two motors are of 3 kW (4 h.p.) each, running at 1460 r.p.m. and giving the chassis a longitudinal vibration of 3.7 mm (0.15 in.). The frame is horizontal, but may be given a slope of $\pm 2^\circ$.

Capacity is 125 m³/h (4,400 cu.ft./h) of juice; if preparation is very fine, it is wise to allow 100 mVh (3,500 cu.ft./h) or 100 t.c.h. To avoid splashing, the juice should be fed from a plate at 45° terminating 5 cm (2 in.) above the screen, thus giving a very low fall onto the grille.

The grilles are supplied by "Filtres Crepines Johnson" (Chatellerault, France, and elsewhere), or by "Surescreen" (Zillmere, Qld.). They are very solid and durable and wear very little. The chassis is supported on 4 pads furnished with dampers in butyl rubber.

Maintenance is very simple; lubrication only at the end of the campaign or every 5,000 hours. On the other hand, the grilles require cleaning every 4 hours. This cleaning may be effected by equipment with a flat and powerful water jet, of the Arker or Symox type, under a pressure of 50 bars (700 p.s.i.). This takes about 5 minutes per screen. It may be made automatic, and then operates below the screen, with a mobile carrier and train of wheels.

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23. *Diffusion*

BRIEF HISTORICAL NOTE

In the great majority of cane sugar factories throughout the world, extraction of sugar from cane is effected by means of mills. This was the process originally employed; it has been progressively improved but never displaced. The only exception was Egypt, where batch diffusion, by a battery of diffusion vessels, as used in the beet industry, was employed for a long time and was discontinued only towards 1960.

However, continuous diffusion, originating before the Second World War, expanded rapidly afterwards, with the rebuilding of European factories which had been destroyed. From 1950 onwards several manufacturers strove to adapt to cane the type of continuous diffuser which they had designed and put into operation with beet. In the beet industry, the *simplicity* of the continuous diffuser, its efficiency and its saving in manpower had caused it quickly to replace the batteries of diffusion vessels, with their complicated installation and operation, which had been universal during the first half of the twentieth century. These were as complicated, in their way, as mill tandems; the simplicity of the continuous diffuser, on the other hand, was very attractive. It remained to adapt it to cane and to try it in operation.

LIMITATIONS OF MILLS

When one considers the great progress which has been realised in the construction and operation of milling plants, one may well ask what are the reasons which today have prompted manufacturers of cane sugar to return to this other method of extraction, namely, diffusion. Would it be simply a matter of fashion? No, There are sound reasons which have prompted them:

(a) The mill is a barbarous piece of equipment (*cf.* p. 132): a tremendous amount of energy is expended to extract the juice from the cane; but the bagasse, thus deprived of its juice, re-absorbs juice the moment it has released it, as it passes the axial plane of the rollers. Thus a large proportion of the benefit just obtained is lost and much of the trouble and power expended is wasted; no method has been found to remedy this inherent disadvantage of the principle of milling itself.

(b) The mills consume considerable power, out of proportion to the result obtained.

(c) The mills are very heavy and very expensive, both the purchase price and in costs of operation and maintenance.

(cf) Whatever the power expended, it is acknowledged that a certain proportion of the juice or liquid contained in the cane or the bagasse cannot be extracted by pressure (*cf.* pp. 134 and 135). In spite of the repeated applications of pressure, in spite of the artifice of imbibi-

tion, there is here a second handicap inherent in the equipment, which puts extraction by mills in an inferior position.

On the contrary, when it is desired to extract in one operation the whole of the sugar contained in cane or bagasse, this is effected no longer by pressure, but by physical extraction by heat, by "diffusion", as is the case, for example, in the analysis of cane or bagasse in the laboratory. The chemical control in the factory is based on the fact that this method of extraction permits of extracting 100% of the sugar contained in the sample analysed.

DIFFUSION

Diffusion is the phenomenon by which two solutions of different concentrations located side by side or, for example, separated by a membrane, exchange by osmosis across the membrane. If it is a case of two solutions of the same substance, the exchange takes place until the concentration is the same on each side of the membrane.

This assumes that the membrane is permeable in relation to the substance dissolved. If there are two or more substances in solution, the membrane may be permeable to the solvent and to certain substances, and impermeable to others; it is then described as semi-permeable. This is the case to a certain extent with the cell walls of the sugar-beet; when certain organic constituents are removed from them by heat, they allow sucrose to pass more readily than certain salts. This is why, with sugar-beet, the diffusion juice has a purity higher than that of the normal beet juice.

In the sugar factory, diffusion is therefore the phenomenon by which the cells of the beet or the cane, immersed in water or a solution of lower concentration than the juice which they contain, give up to that water or to that solution a part or all of the sugar forming the excess of concentration of their juices.

Laws of diffusion

We shall express these laws by adopting the same mode of explanation as Dubourg¹. It is evident that the area of contact between the material to be extracted and the extracting liquid has a prime importance in diffusion. The material is therefore divided into small pieces: slices or cossettes with beet, or chips with cane. Another factor is obviously the difference in concentration between the juice in the cossette and that in the liquid. It appeared logical to Fick to suppose that the extraction from the cossette operates according to the law:

$$\frac{ds}{dt} = -b \Omega \frac{dy}{dl} \quad (23.1)$$

s = quantity of sugar contained in the material to be exhausted

t = time

ds/dt = weight of sugar lost by the material in unit time, at the moment under consideration

b = coefficient depending on temperature

Ω = surface area of the cossettes or chips exposed to diffusion

$dy/dl =$ gradient of the concentration along the line of exhaustion /.

If we assume for example that the cossette is cylindrical and very long, we shall have:

$$\frac{dy}{dl} = \frac{Z - y}{\frac{1}{4}r \cdot}$$

$Z =$ concentration of sugar in the cossette

$y =$ concentration of sugar in the liquid surrounding it

$r =$ radius of the cylinder forming the cossette.

Neglecting the ends, the area of the cossette is:

$$\Omega = 2\pi rL$$

$L =$ length of the cossette.

Whence:

$$\frac{ds}{dt} = -b^2 \cdot 2\pi rL \cdot \frac{Z - y}{r}$$

But:

$$s = \pi r^2 L \cdot Z$$

Hence:

$$\frac{dZ}{dt} = -\frac{8b}{r^2}(Z - y) = -a(Z - y) \quad (23.2)$$

$a =$ coefficient depending on the dimensions (r) of the cossettes and on the temperature.

A. Fixed extraction

Let us assume the cossettes or chips are placed in a cylindrical vessel closed at the base by a perforated sheet. If we seek to exhaust them by spraying with water, the upper layer would be better exhausted since it would profit by a greater difference in concentration. We should have for it ($y = 0$):

$$\frac{dZ}{dt} = -aZ$$

If the flow of water is constant, the quantity W consumed is:

$$W = kt, \quad \text{or} \quad dW = k \cdot dt$$

and:

$$\frac{dZ}{AW} = -\frac{a}{k} Z \quad \frac{dZ}{Z} = -\frac{a}{k} dW$$

$$Z = Z_0 e^{-(a/k)W} \quad (23.3)$$

Z_0 = initial concentration in the cossettes.

Experimental values of the coefficient a show that, in order to reach a low concentration Z , a very high and prohibitive quantity of water W would be required.

B. Counter-current extraction

Accordingly, diffusion is always carried out in a counter-current manner. We shall assume that it is carried out in a pipe or tower (Fig. 23.1), and that the cossettes or chips are introduced at the bottom and the water at the upper end. The exhausted pulp or bagasse will leave at the top and the juice obtained will be removed at the bottom. Let us consider the level N , situated at a distance l from the bottom, and an adjacent level N' . If, at N , the concentration in the cane chips is Z and that of liquid is y , the respective concentrations at N' will then be $Z + dZ$ and $y + dy$. We shall seek to establish an expression for the variations in concentration from one end of the tube to the other.

We shall suppose, as is always at least approximately true, that the weight of the cane chips does not vary in the course of the operation: water is substituted for the juice in the cane in approximately equivalent quantities, in such a way that the weight of exhausted bagasse

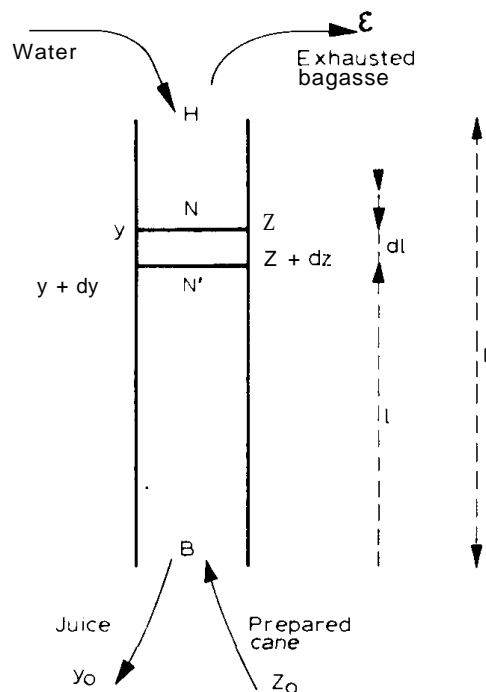


Fig. 23.1. Counter-current extraction.

is approximately equal to the weight of cane chips; and consequently the weight of juice removed is, to a close approximation, equal to that of the water introduced.

Assuming that conditions are steady, we write that the weight of sugar entering the section NH of the tube is equal to the weight leaving:

$$AZ = Qy + A\epsilon$$

whence:

$$y = \frac{A(Z - \epsilon)}{Q} = \frac{Z - \epsilon}{\frac{Q}{A}} \tag{23.4}$$

ϵ = sugar % of the exhausted bagasse

A = weight of cane feed, in kg/h (lb./h)

Q = weight of diffusion juice obtained, in kg/h (lb./h)

Q/A = ratio of weight of diffusion juice to weight of cane.

This ratio, which is termed the "draft", plays an important role in diffusion. We shall designate it by q :

$$\text{Draft} = \frac{Q}{A} = \frac{\text{weight of diffusion juice obtained}}{\text{weight of cane or bagasse subject to diffusion}} \tag{23.5}$$

We have then:

$$y = \frac{Z - \epsilon}{q} \tag{23.6}$$

On the other hand (eqn. (23.2)):

$$dZ = -a(Z - y) dt$$

Substituting for y :

$$dZ = -\frac{a}{q} [Z(q - 1) + \epsilon] dt \tag{23.7}$$

We have also:

$$A = S v \omega_A$$

S = cross-section of tube, in m^2 (sq.ft.)

v = speed of movement of the bagasse in the tube, in m/h (ft./h)

ω_A = apparent density of the bagasse in the tube, in kg/m³ (lb./cu.ft.)

ω_A is a measure of the degree of packing of the material in the apparatus.

Hence:

$$v = \frac{A}{S \omega_A} \quad (23.8)$$

and:

$$dZ = -\frac{a}{q} \cdot \frac{S \omega_A}{A} [Z(q-1) + e] dl$$

and putting:

$$\frac{a S \omega_A}{A} = \frac{\Delta}{q} \quad (23.9)$$

$$\frac{dZ}{Z(q-1) + e} = -\frac{\Delta}{q} dl$$

Integrating:

$$\log_e [Z(q-1) + e] = -\frac{q-1}{q} \cdot \Delta l + \text{constant}$$

Taking $l = 0$, we have:

$$\text{constant} = \log_e [Z_0(q-1) + e]$$

Z_0 is sugar % of the material at its entrance to the apparatus. Hence:

$$\log_e \frac{Z(q-1) + e}{Z_0(q-1) + e} = -\frac{q-1}{q} \cdot \Delta l$$

or:

$$\frac{Z(q-1) + e}{Z_0(q-1) + e} = e^{-q^{-1}(q-1) \cdot \Delta l} \quad (23.10)$$

At the upper extremity of the tube, $z = e$; thus:

$$\frac{q e}{Z_0(q-1) + e} = e^{-q^{-1}(q-1) \cdot \Delta L} \quad (23.11)$$

$$e = \frac{Z_0 (q - 1) e^{-q^{-1}(q-1)\Delta L}}{q - e^{-q^{-1}(q-1)\Delta L}} \quad (23.12)$$

This expression shows that the extraction will improve, that is to say, e will become smaller, as the absolute value of the exponent of e becomes greater. As it contains 3 factors:

(a) The draft q should be made as high as possible. With cane it is often of the order of 1.05 – 1.10, but sometimes drops to 1 and below.

(b) The coefficient of diffusion A should be as high as possible. It depends mainly on a , which increases: (1) with temperature, but a limit is soon reached here, due to the risk of inversion; temperature is accordingly maintained between 65 and 75°C (150 and 170°F); and (2) with the fineness of the cane chips or the pieces of bagasse. A also depends on the degree of packing ω_A , but a limit is soon reached in this respect by the permeability of the bagasse layer, and we must be content with avoiding channelling, that is, the presence of voids or channels in the mass. A will depend finally on S/A , but this ratio is also fixed within fairly narrow limits by the conditions of residence time and time of contact.

(c) The length L of the apparatus in which diffusion is carried out should be as long as possible. We can in fact work on this condition, but must exercise caution, since the residence time and consequently risk of inversion increase with the length of path. We may comment moreover that this factor is to a great extent identical with the time factor (*cf.* eqn. (23.8)).

DIFFUSION AND LIXIVIATION

In the preceding paragraphs we have used the term "diffusion" without comment. We have defined diffusion (p. 359) as a phenomenon of osmosis. This is correct with beet; in cane, however, the cell membrane, even when its colloids have been removed by heat, is particularly resistant to osmotic effects. In cane diffusion, the fraction of sugar extracted by true diffusion, that is, by osmosis, is very slight, being of the order of 3%, for example; it would require a considerable and prohibitive time to extract all the sugar by this means alone. Extraction is actually effected by means of rupture of the cells, and by washing of the broken cells by the water or juice in contact with them. The term "diffusion" is therefore incorrect if understood in the proper sense of transfer by osmosis; it becomes more accurate if understood in the sense of exchange between phases of different concentration in free contact (the term is used in this sense, for example, in the experiments of Berthollet on the "diffusion" of gases); but the operation described as diffusion in the cane factory is actually one of lixiviation, of washing, resembling imbibition in the mills much more closely than the method of extraction utilised in the beet factory.

Moreover, most of the laws of osmotic diffusion apply to lixiviation, only the coefficients for time and efficiency being changed; and we shall proceed with this chapter and continue to use the term "diffusion" while simply pointing out here the meaning to be given to this term.

The foregoing comments demonstrate the importance of preparation in cane diffusion; ex-

traction depends to a great extent on the proportion of cells broken and on the freedom of access of the extracting liquid to the cells.

On the other hand, while diffusion of beet permits of an improvement in purity due to the superior permeability for sucrose relative to that for impurities, with cane the gain in purity will be much smaller, since the lixiviation extracts sucrose and impurities to a similar extent.

BATCH DIFFUSION AND CONTINUOUS DIFFUSION

Diffusion of beet has long been operated in batch diffusers. The diffusion was carried out essentially in the vertical direction, as indicated by the tube in Fig. 23.1. Since the length, in this case the necessary height, was very large (of the order of 30 m (100 ft.)), this vertical tube was divided into 9, 12 or 14 sections, in other words, into 9, 12 or 14 adjacent batch-diffusion vessels; one of these would be filling and one emptying, and the others would be connected so as to act as the necessary height (Fig. 23.2, after Dubourg).

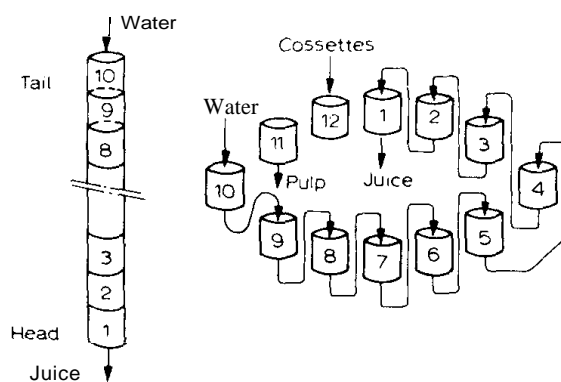


Fig. 23.2. Batch diffusion.

This arrangement was first adopted with cane in Egypt.

Due to the complications of the equipment (valves, rolls and piping) and of operations in such diffusion batteries, as well as to the success of continuous diffusers, the latter type has completely displaced the batch diffuser. This change took place at the time when cane factories were becoming interested in diffusion, and today the only diffusers installed, whether for cane or for beet, are the continuous diffusers which we shall study later.

Continuous diffusers themselves may be subdivided into two types:

(a) Those which are strictly continuous in operation, or diffusers of uninterrupted flow, for which the term continuous may be understood in the mathematical sense of the word; exhaustion in these follows a regular and continuous exponential curve (Fig. 23.3).

(b) Diffusers operating in stages, for which the exhaustion curve is replaced by a succession of steps, the extraction taking place stage by stage (Fig. 23.4).

“*Megasse*”. The spent bagasse leaving the diffuser, before being pressed, is saturated with water, its water content being of the order of 85% or reduced to 70 - 80% by pressure rollers. It is thus a material very different from the bagasse leaving the mills, and particularly final

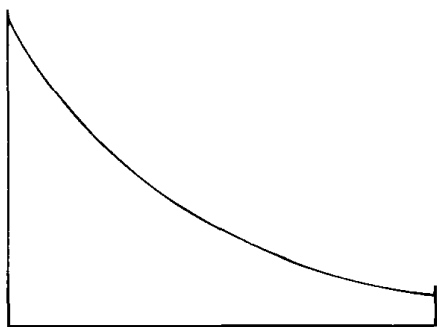


Fig. 23.3 True continuous diffusion - extraction graph.

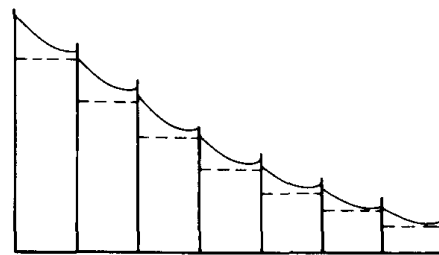


Fig. 23.4. Stepwise "continuous" diffusion. Extraction graph.

bagasse, which contains 45 - 50% of moisture. We shall designate it henceforth by the term "megasse"; this is an old name which was used many years ago as a synonym for bagasse and which we shall revive here in order to avoid confusion with the material leaving the milling plant.

CANE DIFFUSION AND BAGASSE DIFFUSION

There are two ways of extracting sucrose from cane by diffusion.

The first is pure diffusion or diffusion of cane. It consists of operating in a manner completely analogous to that with beet; the cane is prepared and broken up into small pieces by means of knives, shredder, or disintegrator, and is then sent to the diffuser. The prepared cane retains its full weight and still contains all its sugar. If the factory is treating 100 t.c.h. at 14% sugar content, the 100 tonnes of cane and the 14 tonnes of sugar pass to the diffuser.

The second method is diffusion after mills, or diffusion of bagasse. The cane, prepared as for milling, goes first into a mill (in exceptional cases, two mills), which extracts as much juice as possible, say 65 - 70% of its sugar content. It is the bagasse from this mill which is sent to the diffuser; the diffuser thus receives only 30 - 35% of the sugar in the cane, and the weight of bagasse is perhaps 40 tonnes per 100 tonnes of cane. This method of bagasse diffusion is based on the idea that, while it is very expensive to employ 4, 5 or 6 mills to extract 94 - 96% of the sugar in cane, it is a much more payable proposition to employ one mill to extract 70%; the extraction of 70% by one mill is three or four times superior to that of 16 - 23% for each mill of the train. Furthermore:

(a) Passage of the cane through a first mill with coarse grooving conveniently completes the preparation of the cane for diffusion.

(b) Losses by inversion and fermentation during diffusion no longer act on the whole of the sugar content of the cane but only on 30% of it.

(c) The diffuser can be greatly shortened, since the material treated no longer contains say 14% of sugar but only 4%, *i.e.* 30% of the original sugar in cane. It may indeed be further shortened, since a slightly lower extraction (as a proportion of this remaining 30%) may be

tolerated: if it is desired to obtain the same percentage of extraction in the diffuser (97%, for example) as in the former case, the total extraction would be:

$$70\% + (30 \times 0.97)\% \approx 99\%$$

a value which is found by experience to be otherwise unobtainable. However, in order to obtain 97% overall, it would be sufficient to obtain an extraction in the diffuser of 90%, since:

$$70\% + (30 \times 0.90)\% = 97\%$$

We have thus a much shorter and simpler diffuser, and we are justified in seeking a higher extraction.

On the other hand, the width and cross-section of the diffuser will be practically the same as in the case of cane diffusion, since the quantity of fibre introduced is the same and the affinity of bagasse for liquid causes it to absorb juice until its juice content is similar to that in the original cane; thus an equivalent weight of material passes along the diffuser. In other words, on entering the diffuser, the bagasse re-absorbs a quantity of juice equal to that extracted by the mill and takes this juice with it along the diffuser; it is from this material and from this juice that the extracting liquid flowing in the opposite direction has to remove the remaining sugar. There is, however, a slight saving in the lateral dimensions of the diffuser, since the extraction sought is less, and a higher speed of transit and a shorter time in the diffuser may be tolerated; hence a slightly narrower diffuser may be used.

In both cases, whether diffusion of cane or of bagasse, the megasse cannot be left as such; the cane-sugar factory requires fuel and it is quite impossible to burn megasse, on account of its high water content. It is therefore necessary in any case to press this megasse in order to remove the excess juice. A new and special type of press such as the French (Fig. 23.18) may be used, as in Hawaii; this uses an Archimedean screw to force the megasse through a space of decreasing cross-section. However, most factories which install diffusion already possess some mills, and the simplest procedure is to pass the megasse through one or two mills. If the rollers are oversize, one mill could be sufficient. Otherwise two mills are generally necessary to reduce the megasse from 85% to about 48% moisture, in other words, to convert it to ordinary bagasse, which can be burnt in the boiler furnaces.

We see then that cane diffusion will require normally:

$$2 \text{ sets of knives} + 1 \text{ shredder} + 1 \text{ long diffuser} + 2 \text{ mills}$$

whereas bagasse diffusion will require:

$$2 \text{ sets of knives} + 1 \text{ shredder} + 1 \text{ mill} + 1 \text{ short diffuser} + 2 \text{ mills}$$

Retention time in a cane diffuser is obviously greater than that in a bagasse diffuser. The difference is some 10 minutes.

We shall discuss later (p. 397) the economic aspect of these two methods.

The diffusion processes which can claim at present (1982) to have a place in the world sugar industry are the following:

- de Smet
- B.M.A.
- Silver
- Saturne
- F.S. van Hengel

de Smet diffusion

Description. The de Smet diffuser (Fig. 23.5) consists of a long tank enclosing a horizontal conveyor on which the prepared cane or bagasse is placed as a layer of uniform thickness. Throughout its travel from one end of the tank to the other, this layer is copiously irrigated with the extraction liquid. Below the conveyor, the bottom of the tank consists of 11 or 17 adjacent trays which receive the juice which has traversed the layer of material. Of these trays, 9 or 15 are active and 2 spares. The length of the trays in the direction of travel of the conveyor is 2.1 m (7 ft.).

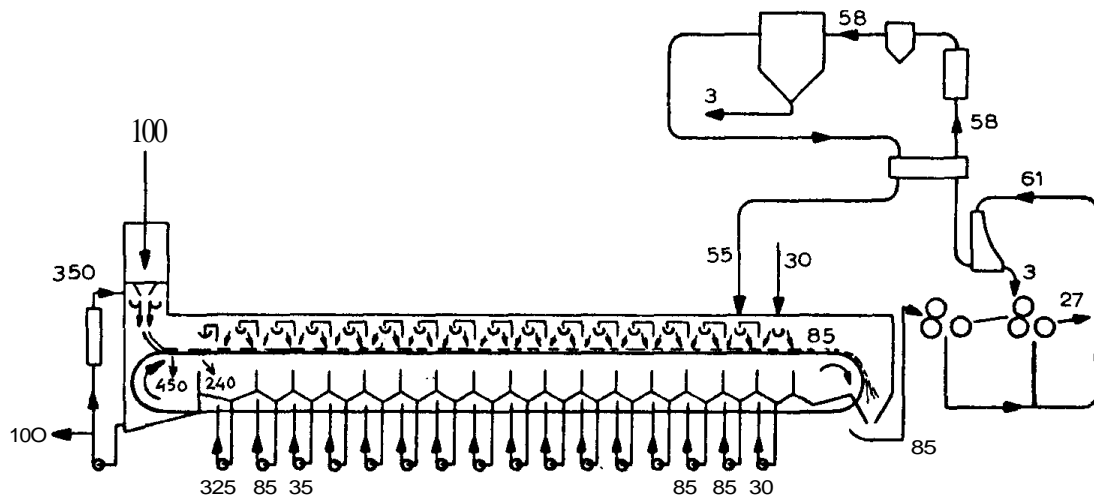


Fig. 23.5. De Smet diffuser. Material balance.

The conveyor extends the full length of the diffuser. The apron of the conveyor consists of a grille of stainless steel, which allows the juice to pass. A pump takes the juice collected on each tray and returns it to a distributor placed above the preceding tray or the next tray behind it, in such a way that the juice moves backwards from tray to tray, from the bagasse discharge end to the bagasse entry end. There is thus a counter-current extraction. The last tray at the megasse discharge end is situated below the first liquid distributor, which receives fresh water. The first hopper at the bagasse entry end is followed by a compartment which receives the diffusion juice which has traversed the megasse layer 9 or 15 times. From that compartment, the last pump in the series takes part of the juice and returns it through one

or two heaters to the hopper which receives the cane or bagasse at the entrance to the diffuser. The surplus of this juice, not heated, is sent to manufacture, the flow being controlled by the level of juice in the compartment.

At the outlet from the diffuser, a tumbler drops the megasse onto an elevator which transports it to the bagasse mill or mills. The thin juice from that mill is returned to the first tray situated below the entry of the imbibition water.

In order to reduce the number of stuffing boxes and thus reduce leakage of juice, the pumps are arranged in groups of three on the one shaft and driven by the same motor.

If there is an interruption to the supply to the diffuser, the pump may be stopped; this is a big advantage of electric drive over other forms of drive. Thus one can avoid flooding the diffuser and interrupting the percolation.

Types. There are 2 types of de Smet diffuser:

(a) The TS type, for bagasse diffusion: a short diffuser, generally with 9 hoppers, 35.8 m (118 ft.) in length

(b) The TN type, for cane diffusion: a long diffuser, generally with 15 juice hoppers, 48.3 m (158 ft.) in length.

In both types, the bagasse layer may be 1 -2 m (3 -7 ft.) in thickness, normally 1.80 m (6 ft.), on which thickness the nominal capacity of the diffuser is calculated. The mechanical limit for filling is 2.20 m (7½ ft.), but a thickness of 2 m (7 ft.) should not be exceeded.

At equal capacity, the width is the same for the two types. The width of the conveyor band is normally 25 mm (1 in.) per t.c.h. A TS type may be readily converted to a TN diffuser of the same capacity.

There is, on the other hand, an important difference in the two systems of circulation. In the TS type the total flow of juice is pumped from one tray up to the preceding tray, whereas in the TN type the juice is divided into two equal streams, one of which traverses the layer on the even-numbered trays and the other that on the odd-numbered trays. The latter circulation system would be particularly suitable for fine preparation, such as is obtained with a shredder. By fine preparation, de Smet means those giving an I.P. of more than 80.

Zones. Three zones are distinguished along the diffuser:

(a) The scalding or heating zone, of about 3-4 m (10 - 13 ft.)

(b) The recirculation zone, which follows, 3.5 (TS) or 1.5 m (TN) (11.5 or 5 ft.) in length

(c) The diffusion zone, which occupies 15 (TS) or 20 m (TN) (50 or 65 ft.).

Power. The installed power is of the order of:

(a) TS: 1.8 kW/t.c.h. (2.4 h.p./t.c.h.) for a diffuser of 100 t.c.h., descending to 1.6 (2.1) for one of 250 t.c.h.

(b) TN: 2 kW/t.c.h. (2.6 h.p./t.c.h.) for a diffuser of 100 t.c.h., reducing to 1.8 (2.4) for one of 250 t.c.h.

The power consumed is 10 - 15% lower.

Steam consumption. The steam used is at 105°C (223°F), or 0.2 kg/cm² (3 p.s.i.) gauge. Assuming the cane is at 30°C (85°F) and the imbibition water at 50°C (122°F), we may reckon:

<i>For heating</i>	<i>For treatment of recirculation water</i>	<i>total</i>
TS: 25 kg/t.c. (55 lb./t.c.)	20 kg/t.c. (44 lb./t.c.)	45 kg/t.c. (99 lb./t.c.)
77V: 74 kg/t.c. (163 lb./t.c.)	20 kg/t.c. (44 lb./t.c.)	94 kg/t.c. (208 lb./t.c.)

For the TS type, the steam consumption for the heating circuit is influenced by the first-mill extraction. For an extraction of 55%, this is approximately 30 kg/t.c. (66 lb./t.c.); for 65% extraction, 20 kg/t.c. (44 lb./t.c.).

Automation. Between the prepared-cane conveyor and the diffuser, for the TN type, or between the first mill and the diffuser for the TS, the prepared cane or the bagasse passes over a continuous weigher; the weight so registered controls the addition of water and the flow of the circulation pumps of the diffuser, so controlling the added water or draft.

The temperatures of heating and of diffusion are controlled thermostatically. A temperature of 70 - 75°C (160 - 170°F) is maintained for heating, and care is taken to maintain the temperature along the diffuser between 75 and 72°C. In diffusers furnished with agitation screws (see below), a temperature of 80°C (175°F) is sometimes maintained, with the object of decreasing the risk of infection at those positions. Heating of the recirculation water is then not employed, as this is sufficient to suppress infection apart from the screws. The diffusion juice is sent to manufacture at the same temperature (70 - 75°C).

The conveyor operates at variable speed, with hydraulic or mechanical drive. The movement is continuous. The speed v of the apron may be regulated from 0.24 to 0.90 m/min for the TS, generally 0.65 m/min (2.1 ft./min); and from 0.24 to 0.70 for the TN, generally 0.65 m/min.

Agitation screws. The surface of the bagasse bed tends to become compacted and so lose its permeability. With the object of avoiding this, some factories instal at one or two points on the length of the diffuser a transverse set of vertical conveyors in the form of Archimedian screws. The rotation of these screws has the object of agitating the bagasse in order to prevent compacting of the surface. This operation is tending to be more widely used.

Residence time. Typical time for passage of the bagasse through the diffuser is:

(a) TS: 30-45 min; for juice, 20 - 30 min

(b) TN: for bagasse, 55 - 65 min; for juice, 25 - 35 min.

Drainage time at the end of the diffusion is 7 min. The surface area of the megasse layer presented to the juice in zones (a) and (b) indicated above (p. 370) is 0.1 m²/t.c.h. (1 sq.ft./t.c.h.) for each of the two zones.

Bulk density. Bulk density, reckoned on the weight of cane only, is of the order of 550-600 kg/m³ (35 - 37 lb./cu.ft.). The density of the layer of cane and percolating juice together is of the order of 570 - 620 (36 - 39), generally 600 kg/m³ (37 lb./cu.ft.).

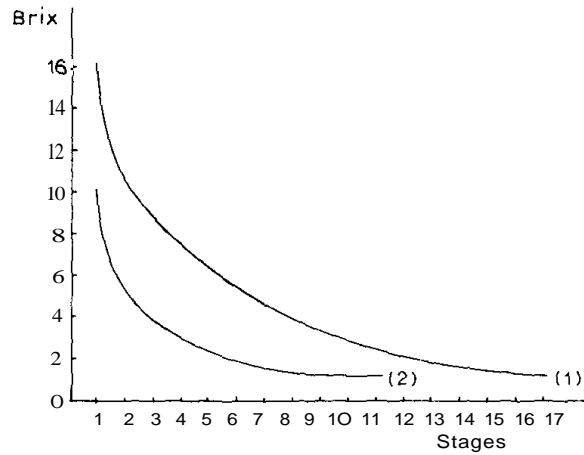


Fig. 23.6. De Smet diffuser. brix variation along the diffuser. (1) TNC diffuser; (2) TSC diffuser.

TABLE 23.1

DE SMET BAGASSE (TS-TYPE) DIFFUSERS

EXTRACTION DE SMET - BAGASSE DIFFUSERS																
TYPE	TS	20 c	25 c	30 c	35 c	40 c	45 C	50 c	55 c	60 c	65 c	70 c	75 c	80 c	85 c	90 c
CAPACITY	METRIC TONS/DAY	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000
PRINCIPAL	WIDTH M	2.800	3.300	4.200	4.700	5.200	5.700	6.200	6.700	7.200	7.700	8.200	8.700	9.200	9.700	10.900
	FT	9.2	10.7	13.8	15.4	17	18.7	19.6	22	23.6	25.3	26.9	28.6	30.2	31.8	35.7
OVERALL	LENGTH	35.8 M - 118 FT						39.5 M - 130 FT								
DIMENSIONS	HEIGHT	ABOUT 8 METRES - 25 FT														
INSTALLED POWER	AT FULL CAPACITY	APPROX 100 HP / 1000 TONS/DAY						APPROX 90 HP / 1000 TONS/DAY								
STEAM CONSUMPTION	KG/TON OF CANE	80-85														

DE SMET CANE (TN-TYPE) DIFFUSERS

EXTRACTION DE SMET - CANE DIFFUSERS																
TYPE	TN	20 c	25 c	30 c	35 c	40 C	45 C	50 C	55 C	60 c	65 c	70 c	75 c	80 c	85 c	90 c
CAPACITY	METRIC TONS/DAY	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000
PRINCIPAL	WIDTH M	2.800	3.300	4.200	4.700	5.200	5.700	6.200	6.700	7.200	7.700	8.200	8.700	9.200	9.700	10.900
	FT	9.2	10.7	13.8	15.4	17	18.7	19.6	22	23.6	25.3	26.9	28.6	30.2	31.8	35.7
OVERALL	LENGTH	48.3 M - 158 FT						52 M - 171 FT								
DIMENSIONS	HEIGHT	ABOUT 8 METRES - 25 FT														
INSTALLED POWER	AT FULL CAPACITY	APPROX 110 HP / 1000 TONS/DAY						APPROX 100 HP / 1000 TONS/DAY								
STEAM CONSUMPTION	KG/TON OF CANE	100-105														

Capacity. The de Smet diffusers are made in the dimensions given in Table 23.1, which indicates the corresponding capacities. A scheme of circulation is given in Fig. 23.6.

Cane preparation. For preparation of the cane to go to the diffuser, de Smet recommends:

(a) *TS*: For diffusion of bagasse: two sets of knives at 500–600 r.p.m., the second being at 20-mm ($\frac{3}{4}$ -in.) pitch. It is desirable that the extraction of the mill should be at least 60%, and there is no disadvantage if the extraction reaches 65% or better.

(b) *77V*: For diffusion of cane: as for *TN*, with the addition of a shredder provided with a means of adjusting the clearance between the hammers and anvil.

Water for heating. This may be reckoned as: for *TS*, 5–10 times weight of bagasse; for *TN*, 4–5 times weight of cane. Draft is generally of the order of 100–105% on cane. If it exceeds 110, addition of water is unnecessary.

Height of layer. The thickness of the bed of bagasse is an important element, as influencing the extraction. When the thickness varies from its minimum practical value of the order of 1.4 m (55 in.) up to 2 m (78 in.), the extraction for the various thicknesses remains practically constant; the pol lost in the bagasse % on cane will remain, for example, in the range 0.25–0.27%. At equal capacity, and for heights of layer from 1.4 to 2 m, the correspondingly longer time of contact compensates for the detrimental influence of the thickness of the bed. In practice, the equipment will normally be operated at about 1.8 m (70 in.), without exceeding 2 m (78 in.).

Imbibition of the megasse. The supply of imbibition water is controlled by the input of cane or bagasse by means of a continuous weigher installed on the conveyor to the diffuser.

The pumps taking the juice from below the diffuser return it to the trays above the diffuser and feed it to the preceding compartment (or the one ahead); the pipes end in a gutter notched at both sides in a saw-tooth pattern, which diverts the juice part to one tray and part to the next, giving a good distribution over the layer of megasse.

The rate of percolation of the juice through the bagasse bed depends largely on the state of preparation, and is limited in practice by the flow through the grille supporting the bagasse layer. The grille, of stainless-steel mesh, is supported on corrugated stainless-steel sheet with perforations along the troughs of the corrugations. This serves to reduce resistance to flow at this point. The perforations are square, 8 × 8 mm (0.3 in.). These grilles normally have 32% of open space, but one recent model has provided slots, giving a figure of 57%. The bagasse supported by the corrugated sheet thus provides zones of lower packing corresponding to the troughs of the corrugations, which serve the role of collectors to drain the juice towards the perforations. This type of grille has allowed an appreciable increase in the rate of percolation. The time of passage of the liquid through the bagasse bed is of the order of 2.5–3.5 min. This corresponds to a speed of percolation of 0.70 m/min (2.3 ft./min), thus of the same order as the speed *v* of movement of the conveyor. The rates of percolation obtained are:

TS: 14–25 m³ per m² of filtering surface per hour, averaging 17–18 (55–59 cu.ft./sq.ft./h)

77V: 7–15 m³/m²/h, or an average of 12 (39 cu.ft./sq.ft./h).

These grilles are made in widths of 900, 1,200, 1,350 or 1,500 mm (35, 47, 53 or 59 in.), according to the capacity of the diffuser. They are supported by frames of the same width, which divide the length of the conveyor into appropriate sections. They move forward on rollers extending the full length of the diffuser, which attains 10.9 m (35.7 ft.) in the largest models. The frames are of articulated construction, and the grilles are flexible.

Cleaning of the grilles is achieved by means of mobile jets of water situated just after the discharge position, some 30 cm (1 ft.) from it. They give a turbulent tangential jet, acting from below and delivering the bagasse particles towards the opposite side where they are readily picked up by the pump.

Treatment of juices from bagasse mill. The thin juices from the final mill are screened, and heated to 103°C (217.5°F). Their pH is about 5.5; after passage through a holding tank, they are limed to a pH of 7.5–8.2, settled, and cooled to 75°C (170°F) for return to the diffuser. It is essential to avoid pumping juices after adding lime before their entry to the subsider. This precaution is necessary to obtain good decantation, so that the clarified thin juices do not present problems of percolation in the diffuser. Addition of a flocculant before passing to the subsider is normally not necessary.

It is important to treat the expressed juice in the minimum of time, hence to have a settler of minimum volume. The settler should have a surface area of 0.20 m²/t.c.h. (2.2 sq.ft./t.c.h.).

It is possible to dispense with the clarification of these thin juices and to send them directly to the diffuser at 75°C (170°F) on condition that they are distributed over two or three neighbouring hoppers. This procedure, however, avoids the pasteurisation of the juices which is effected in the subsider, and so involves some risk of infection in the diffuser.

When the diffuser is fitted with vertical agitating screws, these thin juices are introduced at that point.

Addition of lime. Liming is effected by a light addition of milk of lime in the second compartment from the discharge end of the diffuser, and may be completed towards the middle of the diffuser: in the sixth or seventh compartment from the discharge for the TS type, or the tenth for the TN. It is important to carry out the diffusion at pH 6–6.5, and especially not to exceed 7. De Smet accepts the range of 5.8–6.5, which we think is going too low. We consider that the pH should not be allowed to go below 6. A value of 6–6.5 is obtained by addition of the thin juice at pH 7 after decantation and by the addition of milk of lime at the second compartment.

Advantages. The de Smet diffusion offers the following advantages:

- (a) Grille of the conveyor is very effective.
- (b) Good visibility of what is happening in the interior of the diffuser.
- (c) Opportunity to take samples of juice from each section along the length of the diffuser.

B.M.A. diffusion

B.M.A. diffusers, while similar to de Smet in general design (Fig. 23.7), possess some distinc-

tive features. They are employed for sugar extraction from both bagasse and cane. The bagasse diffuser was introduced into the cane sugar industry with the first generation of diffusers before 1977.

B.M.A. bagasse diffusers (Fig. 23.8) and cane diffusers (Fig. 23.9) consist of a horizontal, rectangular trough fitted with a bottom screen. A system of specially designed chains with variable-speed drive serves to convey the feed through the diffuser. Essential features of **B.M.A.** diffusers are the two rows of lifting screws in the front and middle extraction zones, which are designed to improve juice percolation, and a low-pressure pre-dewatering unit arranged in the dewatering zone.

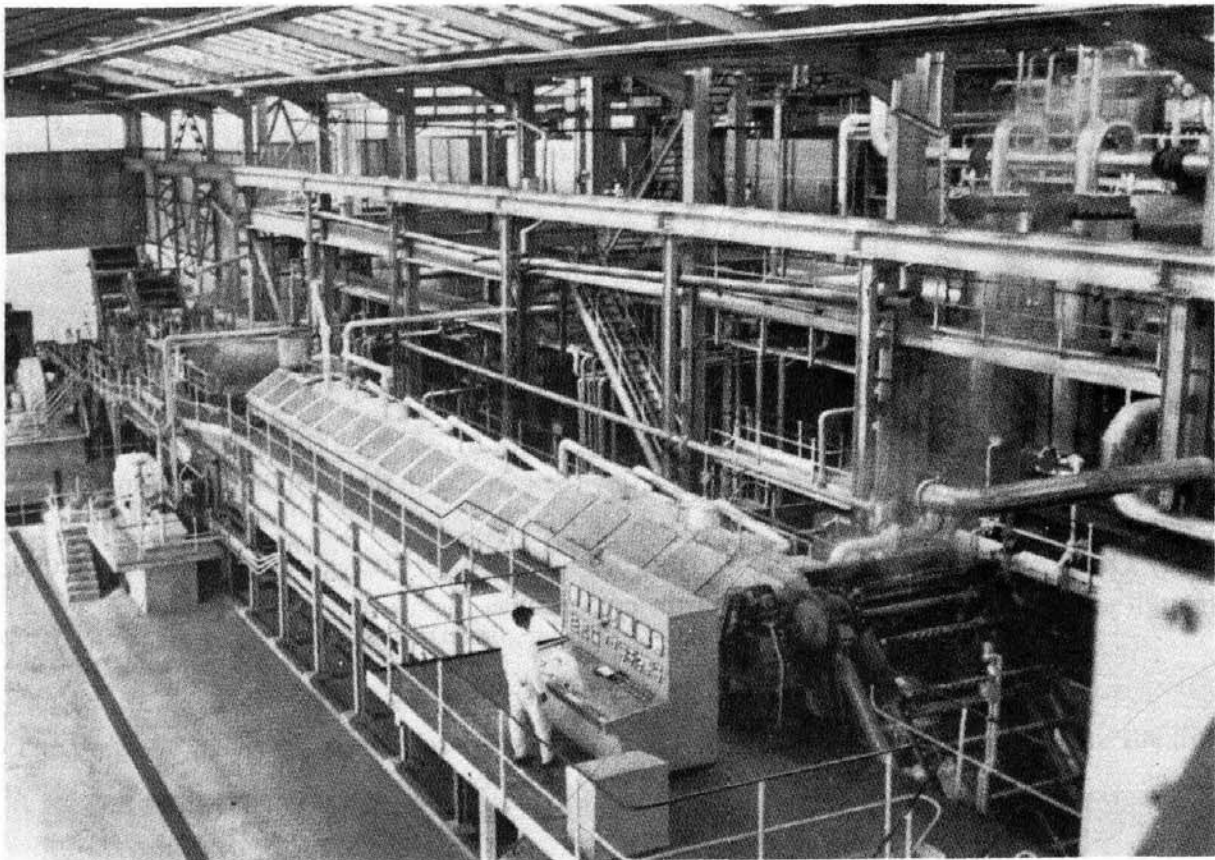


Fig. 23.7. B.M.A. diffuser.

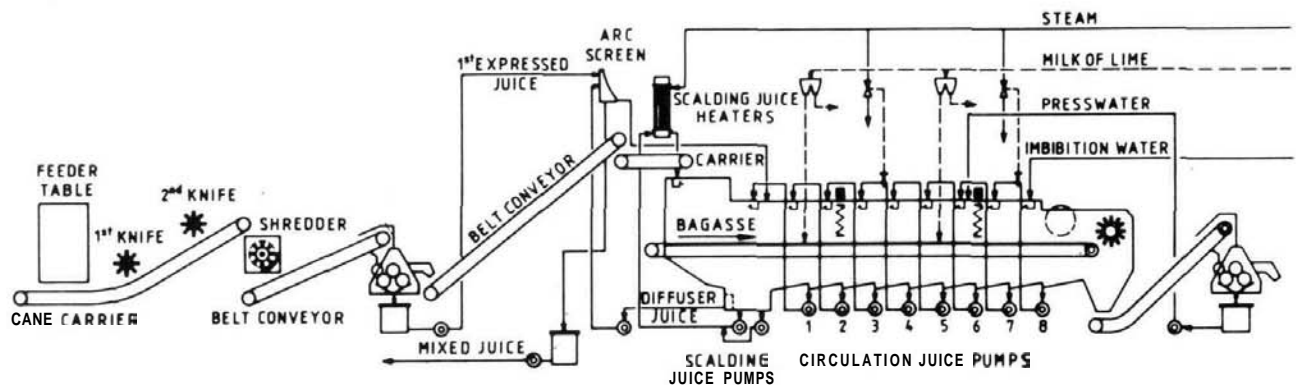


Fig. 23.8 Flow diagram of B.M.A. bagasse diffusion.

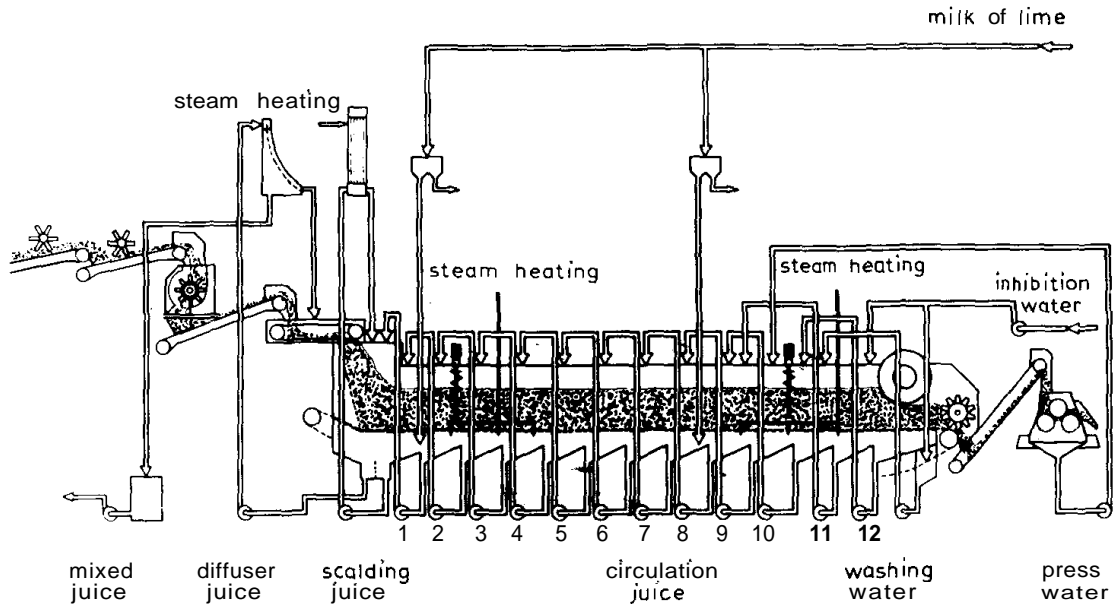


Fig. 23.9. Flow diagram of B.M.A. cane diffusion.

Dimensions. B.M.A. bagasse and cane diffusers are subject in their dimensions to the process parameters:

(1) The length of the trough, 47.25 m (155 ft.) and 61.25 m (200 ft.), and the number of percolation stages (8 and 12) are determined by the extraction times of 36 and 48 min, respectively.

(2) The megasse layer thickness of 1.35 m (4.5 ft.) and, consequently, the height of the trough are determined by the percolation conditions.

TABLE 23.2

CHARACTERISTICS OF B.M.A. DIFFUSERS

	<i>Bagasse processing</i>	<i>Cane processing</i>
Extraction time (min)	36	48
Percolation stages	8	12
Layer thickness (m)	1.35	1.35
(ft)	4.5	4.5
Linear speed (m/min)	1	1
(ft/min)	3.3	3.3
Extraction length (m)	36.0	48.0
(ft)	118	157
Trough length (m)	47.25	61.25
(ft)	155	200
Processing rate at 15% fibre (t/h)	70 /200 /400	70 /200 /400
Diffuser width (m)	2.0/ 6.0/12.0	2.0/ 6.0/12.0
(ft)	6.6/19.6/39	6.6/19.6/39
Driver power (kW)	10.5/28.0/56.0	13.0/35.5/70.0

(3) The width of the trough is determined by the processing capacity. **B.M.A.** diffusers can be supplied in 19 different widths between 2 (6.6 ft.) and 12 m (39 ft.), the rule of thumb being 25 mm trough width per tonne of cane per hour.

Details are shown in Table 23.2.

The megasse layer thickness can be varied between 1 and 1.5 m (3.3 - 5.0 ft.) and the linear speed between 0.5 and 1.5 m/min (1.5-5.0 ft./min).

Juice trays for draught juice, heating juice and recirculation juice are provided below the bottom screen (stainless-steel plate, 8 mm conical perforations) (see Table 23.3).

Operation. The feed enters the diffuser at ambient temperature and is heated to an extraction temperature of 75°C (167°F) by means of juice previously heated in heat exchangers to a temperature of 90 – 93°C (194 – 199°F) with vapour or exhaust steam.

The extraction temperature is maintained at a constant level by heating steam blown into the space below the bottom screen.

TABLE 2.13

JUICE QUANTITIES OF **B.M.A.** DIFFUSERS

	<i>Bagasse processing</i>	<i>Cane processing</i>
Draught juice (% on cane) ^a	45 ^a	105 - 112
Heating juice (% on cane)	300	360
Recirculation juice (% on cane)	106	112

^a 55% juice on cane has already been pressed from the feed by the upstream mill so that the quantity of mixed juice is approx. 100% on cane.

The megasse in the diffuser contains approx. 83% water. Prior to discharge from the dif-

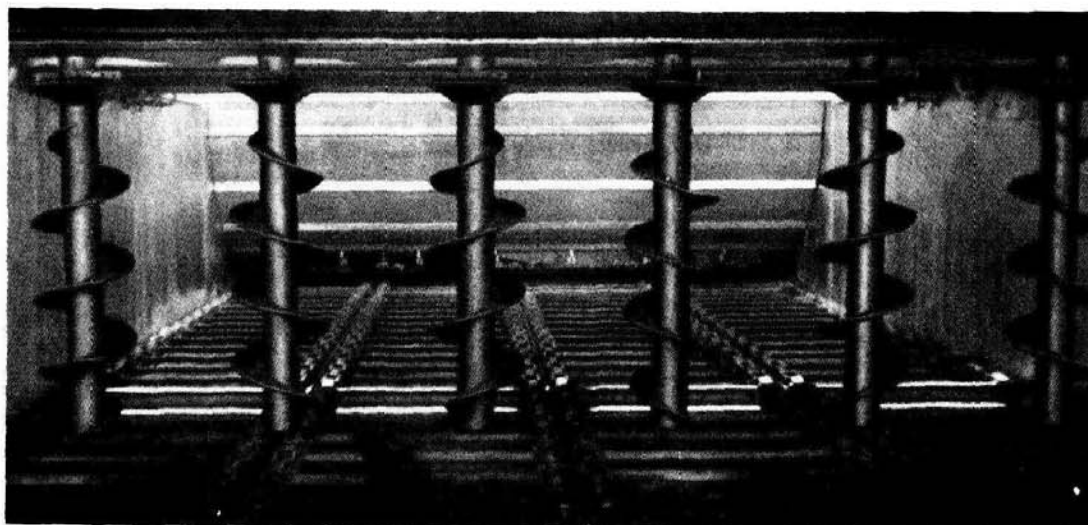


Fig 23.10. Interior view of a B.M.A. diffuser. Vertical stirring screws.

fuser by the rotating discharger, the water content is reduced to approx. 76% by the large water-filled low-pressure dewatering roller which is a characteristic feature of B.M.A. diffusers. Final bagasse dewatering to approx. 50% proceeds in conventional multi-roller mills.

The press water (40 - 50% on cane) is added to the megasse bed in front of the second row of lifting screws (Fig. 23.10 and 23.11) without having undergone any extra treatment.

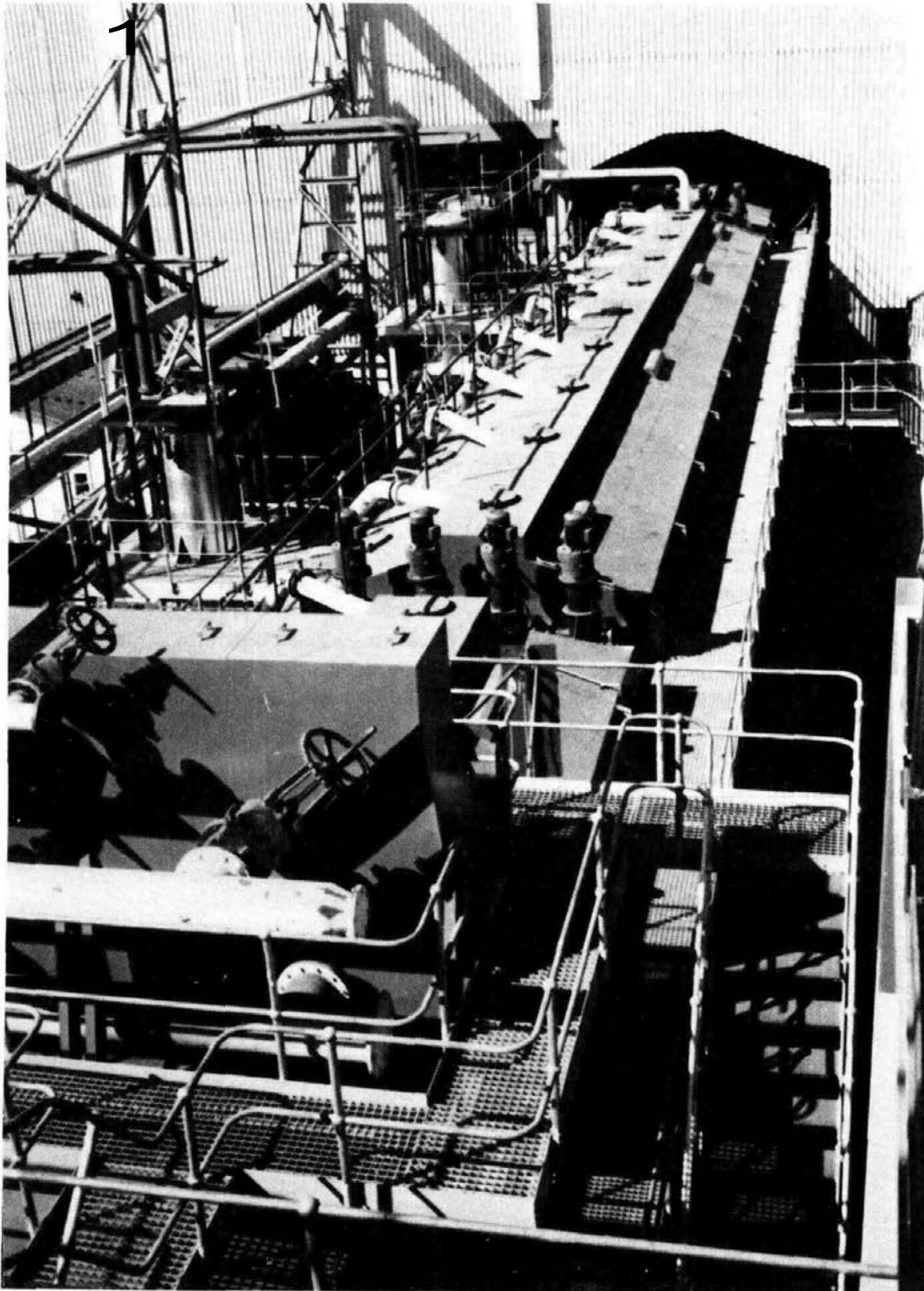


Fig. 23.11 B.M.A. diffuser View from above Screw motorheads

The pH is maintained at a mean value of 6.5 by adding milk of lime (controlled automatically) in the front and middle diffuser sections. Figures 23.12 and 23.13 show the variation of brix and pH along the diffuser, for bagasse and cane diffusers, respectively.

Apart from being cleaned mechanically, the chains in a B.M.A. diffuser are washed with part of the imbibition water which is subsequently added to the megasse bed together with the remaining water.

The total energy required to drive all the units of a B.M.A. diffuser installation (diffuser, screws, discharger, pumps, mill and intermediate carrier) is approx. 6.1 kW per tonne of cane per hour when processing bagasse, and approx. 4.4 kW per tonne of cane per hour when processing cane.

Comparison with mills. The energy required by a bagasse and cane diffuser installation is approx. 40 and 50%, respectively of the energy a mill train of comparable capacity would require.

The total steam consumption of factories using diffuser installations is 5–10% higher than that of factories with conventional mill trains; this is particularly due to a high juice draught when the highest possible extraction figures are sought.

B.M.A. (also a manufacturer of mills) estimate the capital cost for a diffuser installation

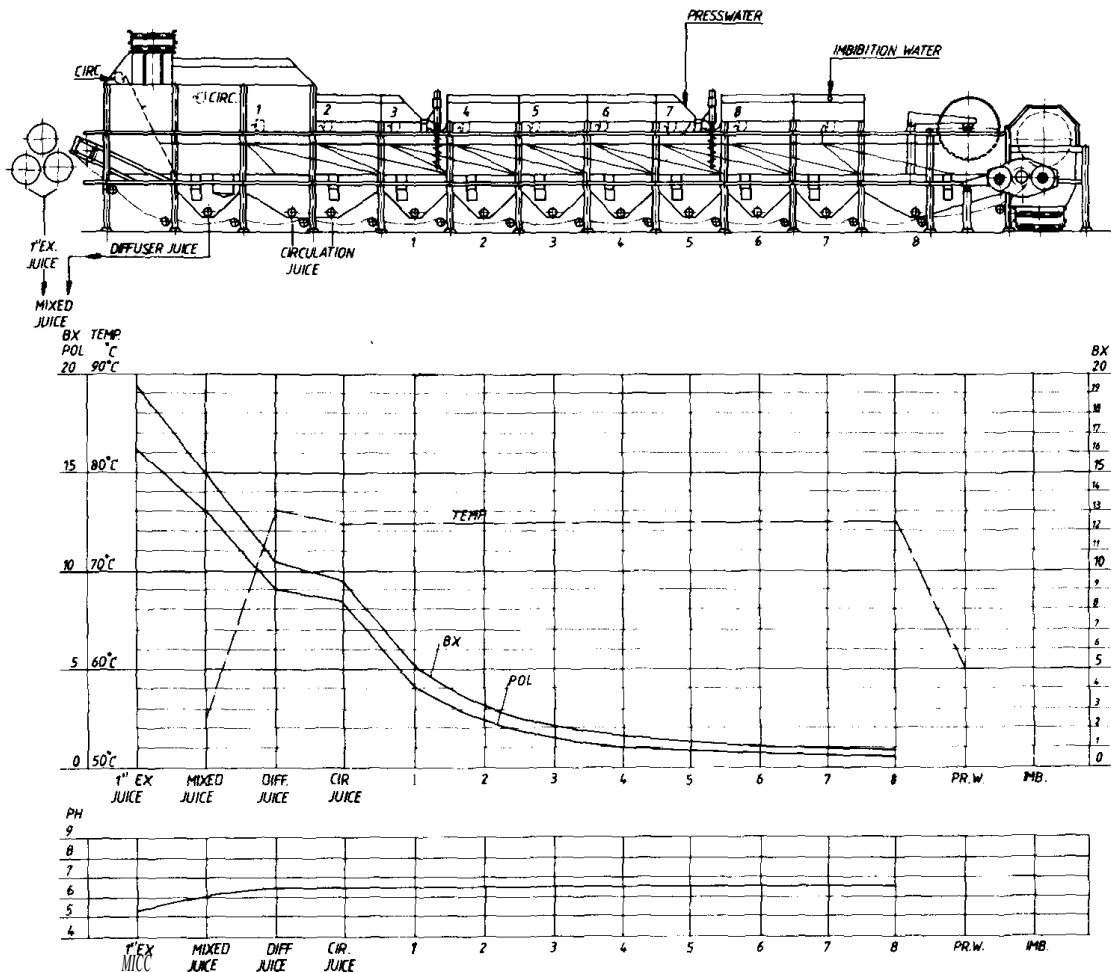


Fig. 23.12. B.M.A. bagasse diffuser. Variation of the brix along the diffuser.

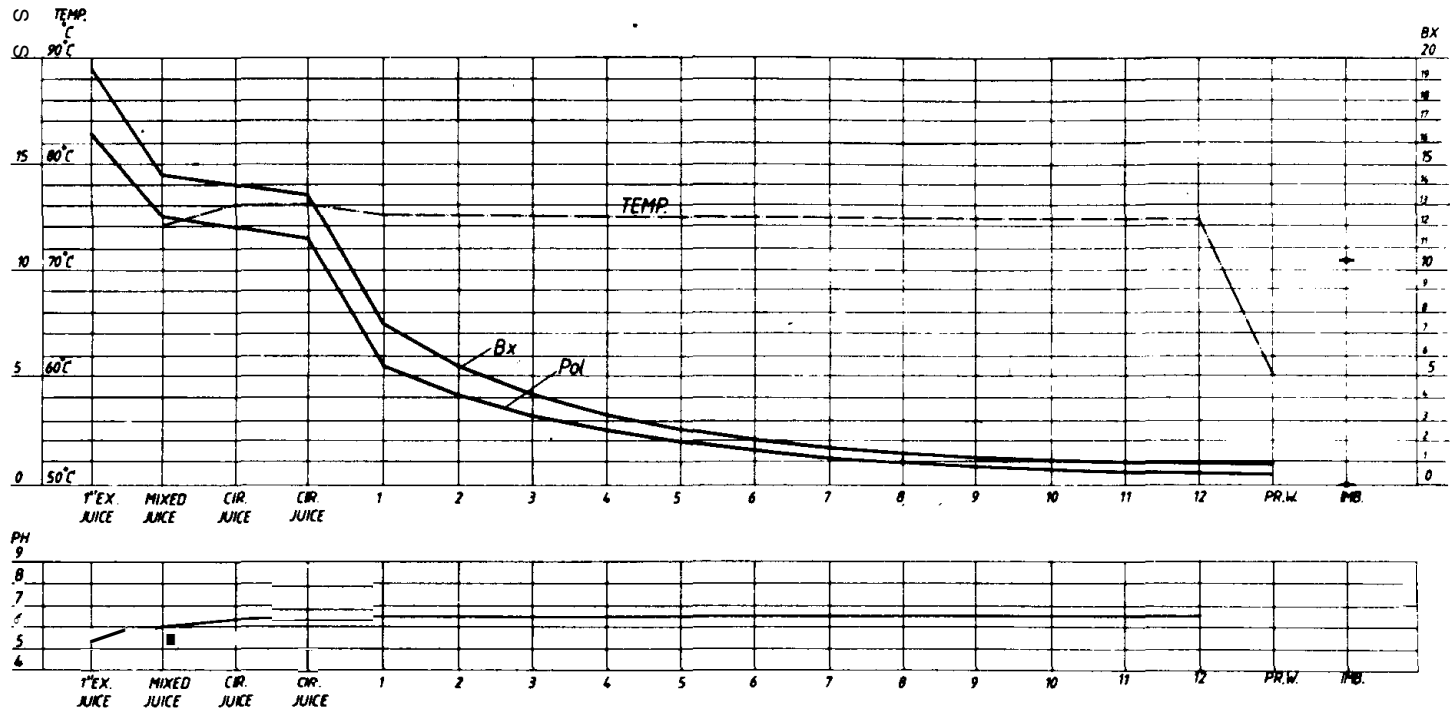
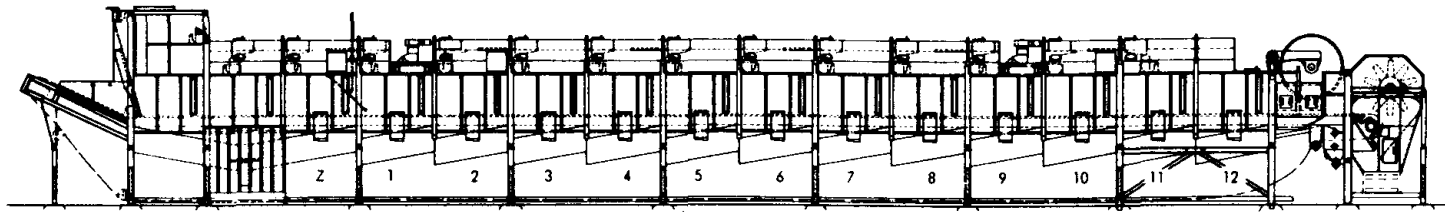


Fig. 23.13. B.M.A. cane diffuser. Variation of the brix along the diffuser.

at 65% of the cost for an installation with five mills, including conveyors and pumps. It should be noted that a major portion of the foundations required for the mills can be dispensed with. The same applies to the buildings, as diffusers are normally located outdoors. Maintenance costs may be reckoned as 40–78% of the costs for a comparable mill train.

B.M.A. point out that less wax and starch are extracted from the feed when applying the extraction process in question and that less filter cake is produced. This accounts for a slight extra gain in the general sugar balance. Reference is made to Dalton, South Africa, where 5% filter cake on cane produced with mill extraction was reduced to 2.5% on cane with a diffuser.

Silver diffusion

The Silver diffuser is also derived from the de Smet type. To sum up, the diffuser is a de Smet unit formed into a ring and thus cylindrical in shape instead of a parallelepiped. This diffusion process, however, displays its originality not so much in the diffuser as in the efforts made to break as completely as possible with the classical methods of extraction, namely, knives and mills. Though the items of equipment used are very recent, they have often attained their object and are very interesting. Their evolution merits some consideration, and certain of them are likely to compel attention in the near future.

The Silver diffusion is mainly designed for diffusion of cane.

Description. On account of the originality of the accessories of the diffuser, to which we have just referred, we shall describe a complete Silver diffusion process such as has been realised in the typical installation at Pioneer (Hawaii), although theoretically the diffuser could be installed without its accessories of preparation and dewatering. We shall distinguish then:

- (A) Preparation of the cane
- (B) Diffuser proper
- (C) The dewatering of the megasse.

A. Cane preparation. The cane passes first through a very rudimentary set of knives, carrying only nine blades in three rows of three, set 120° apart, thus leaving 23 cm (9 in.) between the 9 respective circles of cut, and set at 75–100 mm (3–4 in.) from the carrier apron. This device scarcely contributes anything to the preparation and has the sole object of facilitating feeding of the following equipment. The cane thus partially cut passes over a magnetic separator to protect the following preparators against large pieces of iron which may be present in the cane.

The cane then arrives at the "buster" (Fig. 23.14). This is a hammer mill, or shredder, rotating at 1,150 r.p.m. It is fed by two feeding rollers, rotating at about 120 r.p.m., and fitted with pins 20 mm ($\frac{3}{4}$ in.) in diameter, and 100 mm (4 in.) in length; these throw the cane into the buster. The hammers, placed in 4 rows, tear the cane and pass it through a grille formed of circumferential bars connected by horizontal anvil bars. This gives a cane coarsely shredded, in pieces of 7.5–15 cm (3–6 in.) in length.

It is then picked up by an elevator, and at this moment receives pulverised lime, conveyed and distributed by compressed air, or milk of lime, for the liming before diffusion, to which

we will return later. The elevator drops the prepared cane into the "fiberiser" (Fig. 23.15). This is a second hammer mill, resembling the classical shredders more closely, fed by a feeding roller similar to the upper roller of the cane buster, and rotating at the same speed, but smooth. The fiberiser hammers, similarly placed in 4 rows, achieve the shredding by scraping the pieces of cane against a toothed cylindrical plate. It rotates similarly at 1,150 r.p.m. and furnishes shredded cane, containing a large proportion of long detached fibres.

This material is picked up again by an elevator and a belt conveyor on which it is weighed by means of a gamma-ray cell, which controls the speed of the rollers feeding the cane buster. It is then conveyed to the diffuser and distributed along the first compartment of the diffuser.

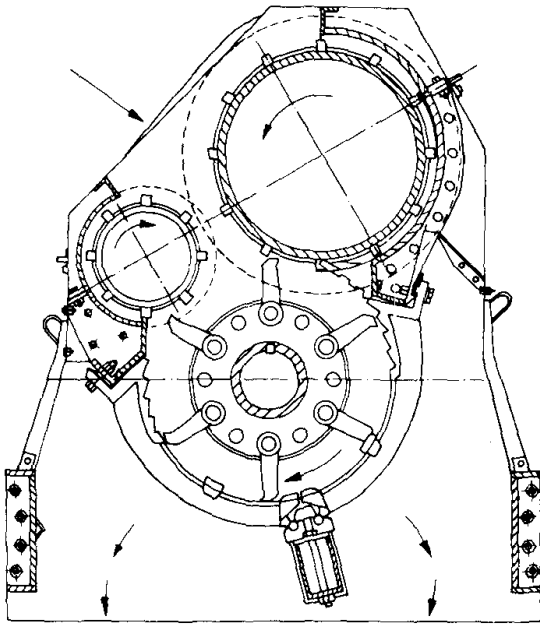


Fig. 23.14. Silver diffusion. Cane buster.

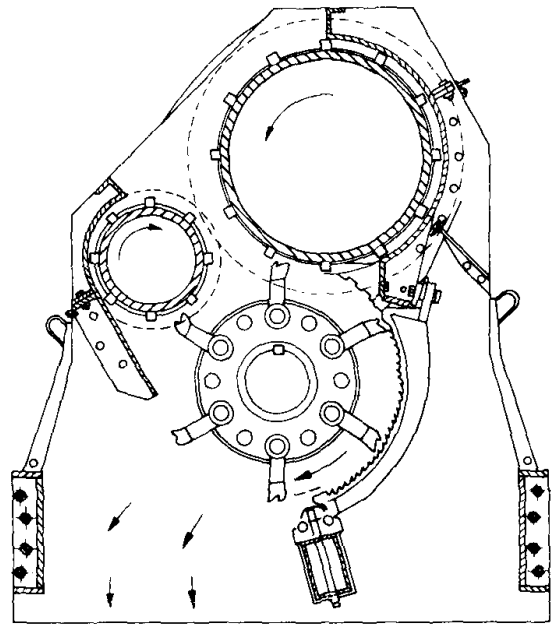


Fig. 23.15. Silver diffusion. Fiberiser.

B. The diffuser. The diffuser (Fig. 23.16) consists of a moving annular vessel of approximately 2.5 m (8 ft.) in height, consisting of two concentric cylinders, the inner one being of about 60% smaller diameter than the outer. This annular corridor is supported both on the inside and on the outside by wheels running on circular rails (Fig. 23.17); radial arms connect it with a central shaft; it is rotated by means of two hydraulic jacks acting successively on 36 arms rigidly attached to the outer wall at 10° intervals. The bottom consists of a plate with perforations in "honeycomb" pattern, 13 mm ($\frac{1}{2}$ in.) in diameter, giving very large drainage area.

Seen in plan, the diffuser shows inside this annular corridor:

- (1) The wall alongside which the prepared cane is deposited, and the device for distributing it.
- (2) 2 recirculation troughs, occupying together 15° .
- (3) 16 troughs for distribution of extraction juice, each occupying 15° of the circumference.
- (4) \ similar trough for receiving and distributing the thin juices.

(5) 1 similar trough for receiving and distributing the first extraction water.
 In all, 20 troughs occupying 285°.

(6) 3 pairs of vertical screw conveyors placed side by side, standing against the fixed wall which closes the end of the corridor; the screws of each pair rotate in opposite directions and the exhausted megasse meets the screws and is picked up by them, elevated above the megasse bed and distributed into a horizontal screw conveyor. This takes it out to a belt conveyor. A vertical plate, suitably curved, encloses the screws and prevents the megasse from falling back into the interior of the diffusion space. A scraper blade scrapes the bottom plate and guides the lowest layer of megasse into the lower turns of the screws.

(7) A fixed and empty compartment occupying 15°, separating the entry and exit walls, across the full width of the corridor, the purpose of this being to give a definite separation between the points of entry of the fresh shredded cane and the removal of the exhausted megasse.

Below the moving corridor (Fig. 23.17) there is a fixed annular tank of the same width,

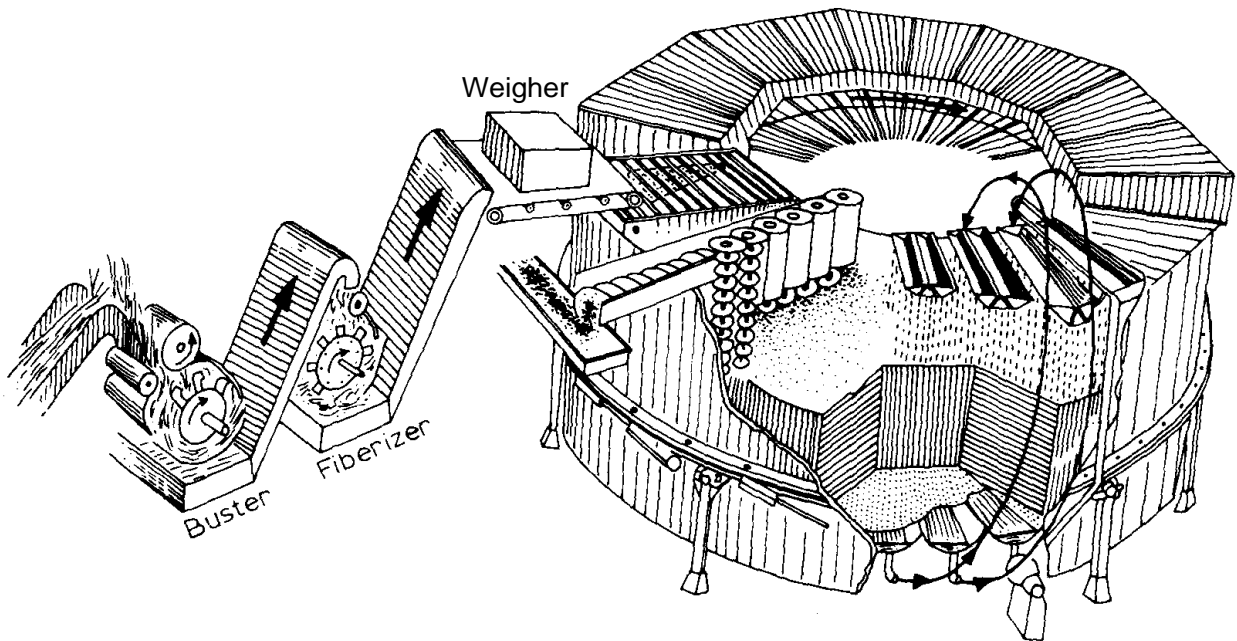


Fig. 23.16. Silver diffuser.

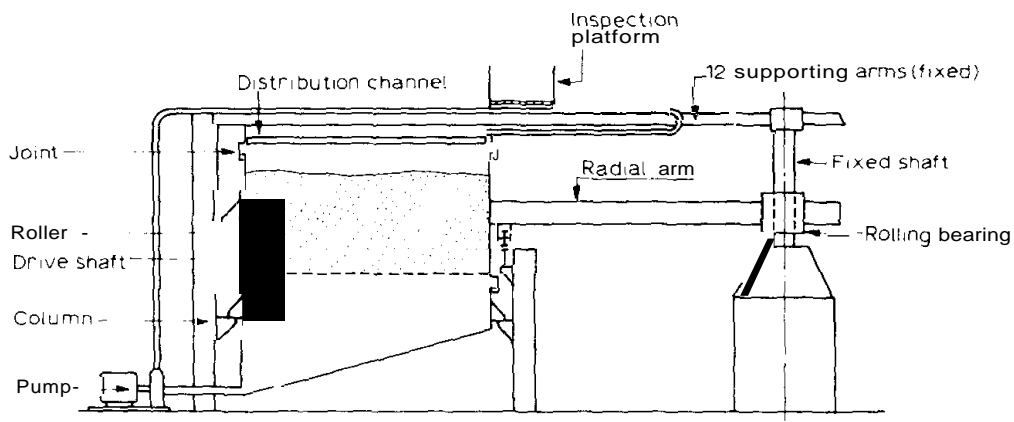


Fig. 23.17. Silver diffuser - vertical half-section.

divided into 19 trays for receiving the juice which has percolated through the megasse bed. The bottom of these is inclined towards the outside, and the juice which collects is removed by an equal number of pumps which return it to the distribution trough displaced by 30° or 45° towards the cane entry. In the Pioneer diffuser, the position of the distributors may be displaced by 32°. In later installations, there are only two positions possible: one corresponds to a displacement corresponding to exactly one distribution trough, the other to a complete recycling, through the tray, of its own juice. The latter position is used in the event of an interruption to cane supply².

Recirculation of juice immediately after the entry of cane to the diffuser is employed as with the de Smet system, this juice being re-heated to 80°C (180°F) by bled vapour. This recirculation has the object of:

- (a) Bringing the cane immediately to a suitable temperature
- (b) Assuring a good distribution of the lime, and making the mass homogeneous
- (c) Relieving any packing effect
- (d) Screening the diffusion juice before removing it.

Returning the various juices to a tray 45° ahead has three results:

- (a) The juice applied to the megasse bed and percolating through it is returned, not only into the next tray to that from which it is collected, but into the preceding one
- (b) Consequently, its course is partially divided into two and thus shortened; hence, a shorter time of the juice in the diffuser
- (c) The difference between the concentration of the megasse and the extraction juice is slightly increased.

On the other hand, this displacement of returned juice short-circuits the diffusion to a certain extent.

The moving annulus slides along the upper plates carrying the distribution troughs, and along the lower tank containing the juice trays with a circular joint, liquid-tight to vertical juice flow (Fig. 23.17).

C. Dewatering. The exhausted megasse contains about 86% water³. The original diffusion installation at Pioneer provided three different methods of dewatering this megasse:

(a) *Mills.* Part of the megasse was fed without difficulty into the last two mills of the old tandem.

(b) *Cone-press.* A second portion of the megasse was passed through a press designed and constructed by Silver (Fig. 23.18): it consisted of two cones of perforated metal plate 1 m (3ft.) in diameter with apical angle of 170°, with their axes horizontal and converging, and close together at one position, *i.e.* with two generatrices situated in the common axial plane, parallel to each other and close together. The megasse was introduced into the open space, and the cones in rotating compressed it into the narrow space between these two generatrices. The cones rotated at 3 r.p.m. and gave a bagasse of 45% moisture. The output corresponded to 15 t.c.h. and the power consumed was about 3 kW/t.c.h. (4 h.p./t.c.h.). It seems that this is the device which has the greatest future and which could lay claim to replacing the dewatering mills. It is the device recommended by Silver and Dorr–Oliver.

Silver manufactures cane presses in 3 sizes:

1,575 mm (62 in.) for 14.9 t.f.h.
 1,981 mm (78 in.) for 22.6 t.f.h.
 2,388 mm (94 in.) for 33.8 t.f.h.

It is commented that two 1,575-mm (62-in.) presses cost slightly less than one 2,388-mm (94-in.) press.

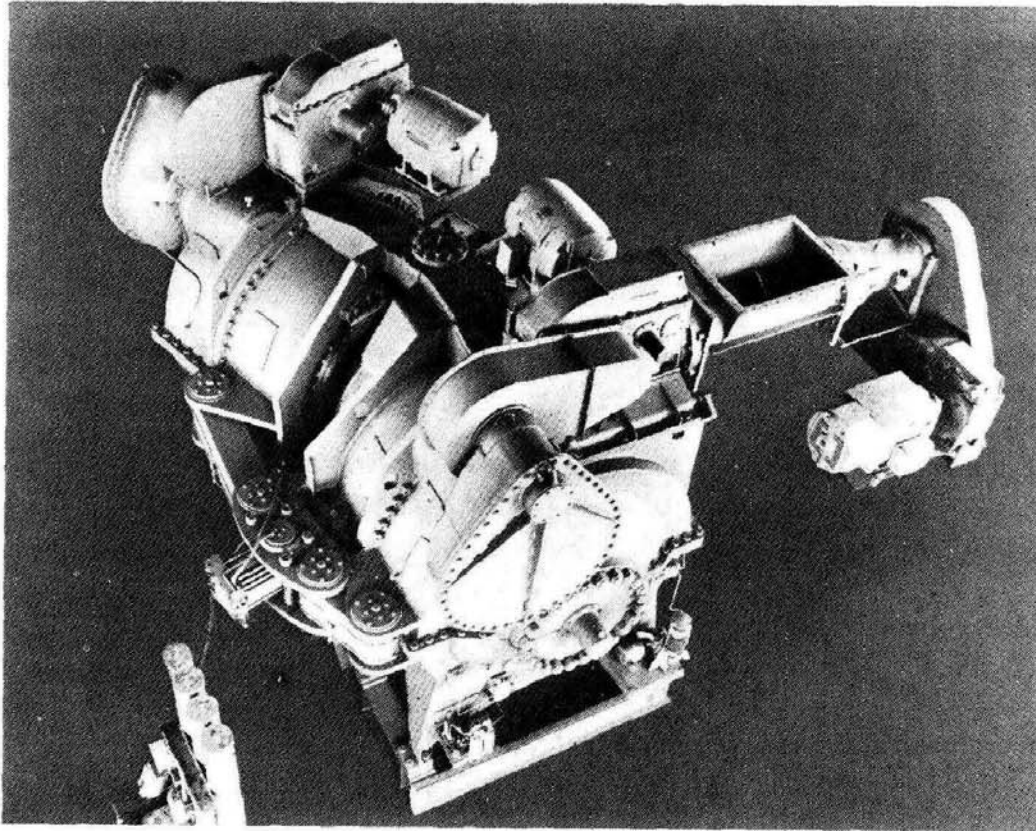


Fig. 23.18. Silver cane press.

(c) *French press.* Finally, the megasse leaving the diffuser could equally well be sent to a third type of press. This is a screw press forcing the megasse to pass through a decreasing area and reducing it to a moisture of about 48% (Fig. 23.19).

Materials balance. The materials balance of the Silver diffusion is completely similar to that of the de Smet TN (Fig. 23.20).

Thickness of bed. Silver recommends operation with a thick bed, of 1.2 - 1.8 m (4 - 6 ft.); in this he holds the opposite view to de Smet. The packing factor is of the order of 190 - 320 kg/m³ (12-20 lb./cu.ft.).

The thickness of the bed may be regulated by means of the speed of rotation of the dif-

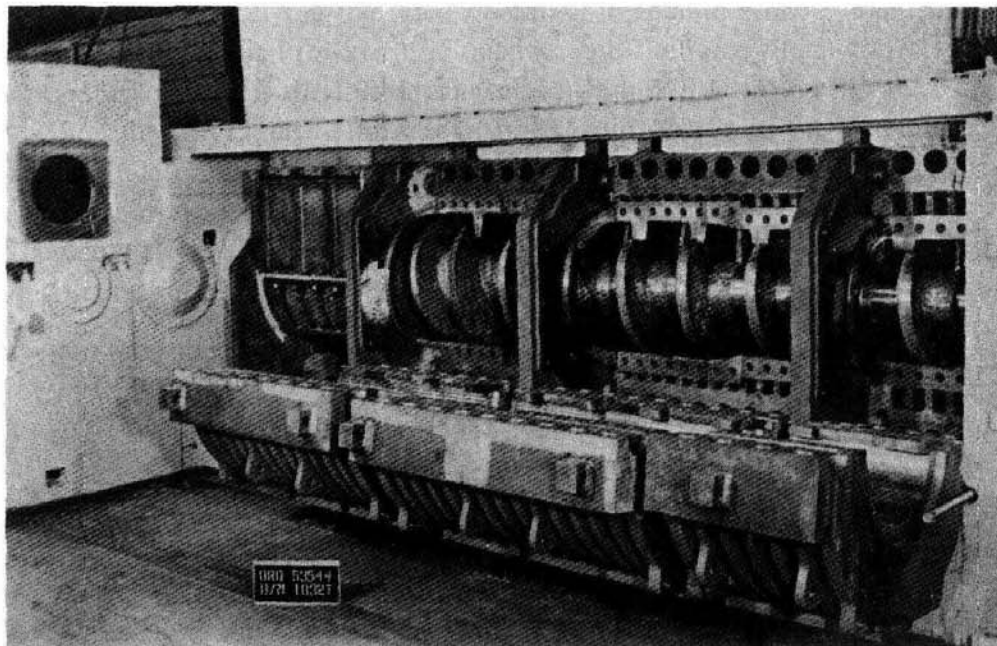


Fig. 23.19 French press (opened).

fuser. For a 1.5 m (5 ft.) depth of bed, the percolation rate is about $245 - 295 \text{ l/m}^2/\text{min}$ ($5 - 6 \text{ gal./sq.ft./min}$).⁴ The rate of percolation for the Suchem diffusion at Louisa in Louisiana⁵ is $400-410 \text{ l/m}^2/\text{min}$ ($10 \text{ U.S. gallons/sq.ft./min}$).

Residence time. This is given as about 60 min for the cane, hence 40 – 50 min for the diffusion process; the time for the juice is of the order of 20 min.

Temperatures. The recirculation heats the cane to 80°C (180°F) at its entry to the diffuser. The make-up heaters prevent the temperature from falling below 70° (160°) during diffusion, and maintain it between 70 and 73° (160 and 165°). The thin juices are raised to 80°C before introduction to the diffuser. The diffusion juice leaves at about 65°C (150°F). All these temperatures are thermostatically controlled and maintained.

The juices are heated by vapour bled from the multiple effects. Part of this heating is effected by direct injection of vapour, either in the extraction water, or in the course of the diffusion, by two injectors.

Liming. The delicate problem of liming in the diffusion has been resolved by Silver in the following way: the lime is added, either as powdered lime, or in the form of milk of lime, between the cane buster and the fiberiser, or before the cane buster, in the ratio of about 0.8 kg (1.8 lb.) CaO per tonne of cane.

At the entry to the diffuser, the recirculation assures a good mixture of the lime. The pH in the diffuser would be approximately 6.5, and the diffuser juice leaves at a pH in the neighbourhood of 7 (6.8 – 7.3). The thin juice from the press would then have a pH of the order of 6.5, the lime addition being regulated so that it does not fall below that limit.

Treatment of thin juices. The thin juices obtained, either with the French press, the cone press, or with dewatering mills, are screened successively by 2 DSM screens with bars spaced at 1.5 mm ($\frac{1}{16}$ in.), without any addition of lime or flocculant, and are then re-heated to 65–85°C (150–185°F) and sent to a subsider (Fig. 23.20). The clear juice leaving this subsider goes to the second last juice trough before the megasse removal screws, which is the last liquid addition except that of water. The mud from this subsider is granular and settles rapidly; the ordinary Oliver filtration does not handle it satisfactorily and it is necessary to feed the mud near the top of the filter drum. The filter cake would contain only 0.5% of sugar.

Trials made in the attempt to dispense with this treatment of the thin juices have shown that the materials which they contain in suspension are retained by the upper layer of the megasse bed and appreciably decrease the speed of juice percolation. The treatment is thus useful.

Power. Power consumption for the whole installation is given as follows:

	<i>kW/t.c.h.</i>	<i>(h.p./t.c.h.)</i>
Knives:	0.5	(0.7)
Cane buster:	5.8	(7.8)
Fiberiser:	4.2	(5.6)
Diffuser - Rotation:	0.03	(0.05)
Screws:	0.5	(0.7)
Horizontal bagasse conveyer:	0.03	(0.05)
French press:	4.5	(6.1)
	-----	-----
Total:	15.5	(21.0)

The cone press would consume about 3 kW/t.c.h. (4 h.p./t.c.h.), instead of 4.5 for the French press.

For comparison, extraction by mills would require, for knives, shredders and 5 mills, about 14–15 kW/t.c.h. (18–19 h.p./t.c.h.).

For the diffuser, however, if the cane buster and the fiberiser are driven by electric motors (they may of course be driven by steam turbine), the whole of the power requirement is electrical, whereas the greater part of the power demanded for the milling plant is generally provided by steam engines or turbines. It follows that an existing factory, in which the mill drives are not electrical, which is changing to a diffusion with accessories of the Silver type, should have available at its power house about 3–4 kW/t.c.h. (4–5 h.p./t.c.h.) more than previously. If the mills were electrically driven, the saving would on the other hand be equal to 10 or 12%.

Steam consumption. The steam consumption for the heaters and steam injectors is of the order of 4—5% on cane, but much of this requirement can be provided by bleeding from the multiple effects.

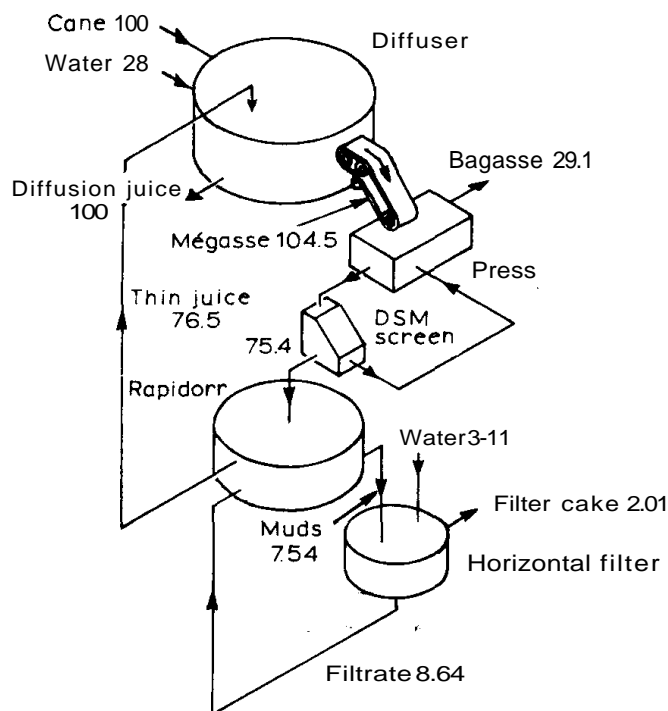


Fig. 23.20. Silver diffusion - material balance.

Maintenance. Estimates by manufacturers and users, of the maintenance costs for the complete diffusion equipment, with its special accessories, vary between 25 and 45% of the corresponding expenses for a milling tandem. If use and experience confirm a figure within this range, this would be an important argument in favour of this diffusion system and its accessories.

Accessories. We have drawn attention to the originality of the accessory equipment for cane preparation and particularly for dewatering of megasse. These are not indispensable, and it would be quite possible to adopt the more normal equipment which we have discussed with reference to de Smet and B.M.A. In particular, with clean cane, the cane buster could very well be replaced by a set of knives with close clearance. In the same way, a standard shredder would give very much the same result as the fiberiser, although the latter equipment appears to be a success and could compete advantageously with standard types of shredder.

Finally, the dewatering could be effected by one or two mills.

Moreover, the Eldorado factory in Mexico has converted to the Silver diffusion process while retaining precisely these standard devices for preparation and dewatering.

Clarification. The diffusion juice leaving the diffuser is sent directly to the evaporators. Elimination of the main clarification station is thus another point to be credited to Silver. The purity of the juice is slightly lower than that of the clarified juice obtained from the same cane by means of mills and a standard clarification, but it is slightly superior to that of the mixed juice leaving the mills in the latter case. In other words, part of the purity rise obtained by standard clarification methods is lost. However, it is quite possible that this loss, which

is small, may be partially or totally compensated by the fact that inversion losses taking place in the clarification are avoided, in particular those occurring during week-end stops. For a new factory, economy in space and investment is in any case attractive.

Material of construction. The metal of the Silver diffuser in contact with juices is stainless steel. This precaution would seem superfluous on account of the preliminary liming, and equipment in ordinary steel would be less expensive.

Sizes, The Silver diffuser is offered in 8 sizes, as given in Table 23.4.

Advantages and disadvantages

A. *Advantages.* The Silver diffusion process offers the following advantages:

- (a) Reduced maintenance costs if Silver or French presses are adopted
- (b) Supplementary saving in power in the same case
- (c) Possible elimination of the normal clarification.

B. *Disadvantages*

(a) The Silver diffusion is specially designed for diffusion of cane. If it were desired to use diffusion of bagasse, it would be hardly suitable and it would be desirable to re-design it for this purpose, with a smaller number of compartments.

(b) As with any cane diffusion process, it does not permit payment according to sugar content as determined by sampling of primary juice or first expressed juice. It would be necessary to convert to a system of direct analysis, and this may be contrary to existing regulations.

(c) Adoption of the recommended accessories presents no difficulty when building a new factory, but constitutes a complete rearrangement of the whole extraction plant for an existing factory. These accessories, however, may be ruled out and the normal equipment for cane preparation and bagasse dewatering retained.

(d) There is also the necessity to provide approx. 4 kW/t.c.h. (5 h.p./t.c.h.) extra from the power house, in order to suit the demands of the diffuser accessories, if the mills replaced are not electrically driven.

TABLE 23.4

SERIES OF SILVER DIFFUSERS

<i>Diameter</i> <i>(ft. and in.)</i>		<i>(m)</i>	<i>Capacity</i> <i>(t.c.h.)</i>	<i>Power requirement</i> <i>(diffuser only)</i>	
				<i>(kW)</i>	<i>(h.p.)</i>
29 ft.	2 in.	8.89	42	92	123
35 ft.		10.67	61	142	190
40 ft.	10 in.	12.45	84	188	251
46 ft.	8 in.	14.22	107	246	329
52 ft.	6 in.	16	138	276	369
58 ft.	4 in.	17.78	168	373	498
64 ft.	2 in.	19.56	206	415	554
70 ft.		21.34	244	535	714

(e) The circular form of the diffuser does not permit of adding further compartments and distributing troughs in order to allow for an eventual increase in capacity. However, the number provided (18) is generously designed and permits a certain flexibility.

(f) The circular arrangement theoretically has an inherent disadvantage: the distribution of juice from a trough should be proportional to the weight of bagasse which receives it, and hence the flow of juice from the trough should vary in proportion to the radius measured from the vertical axis of the diffuser. This is not readily arranged for all possible rates of flow. It follows that the extraction is liable to vary along the radius.

Similarly, it is difficult to distribute the prepared cane evenly in a uniform layer.

Conclusion. In spite of these disadvantages, the Silver diffusion is a success mechanically and seems to give good results, with extractions exceeding 97 and sometimes approaching 98. It would appear particularly that the system has shown how to avoid or reduce losses by fermentation and inversion. The results recorded seem to indicate that the gain in extraction is indeed found in the overall recovery, in the sugar leaving the factory.

Furthermore, certain accessories for cane preparation and bagasse dewatering, such as the fibriser and the cone press, once put into operation, could demonstrate certain advantages and become general.

Rotocell. In Venezuela a diffuser under the name of Rotocell⁶ has been installed; it resembles the Silver system but the vessel receiving the megasse is divided into compartments. It also gives a more convenient discharge of the exhausted material: at the end of its travel, the bottom of the compartment opens and drops the exhausted megasse.

Compaction and percolation. The three types of diffusers described are all of similar design, following the pattern of de Smet.

There are two factors which play an important part in this type of diffuser, which are inter-related. These are the degree of compaction of the material, expressed as weight of fibre per unit volume, and the speed of percolation; the latter factor characterises the efficiency of operation of the equipment and depends to a great extent on the degree of compaction. In this type of diffuser, the weight of fibre per m^3 varies generally between 60 and 80 kg/m^3 (4–5 lb./cu.ft.). Above 80 kg/m^3 , the speed of percolation falls sharply and thus produces flooding: the juice does not penetrate quickly enough and forms a pool on the surface.

The speed of percolation varies from 0.15 to 0.35 m/min (6–14 in./min); it is often held between 0.20 and 0.25 m/min (8 and 10 in./min). It varies in inverse ratio to the degree of compaction, but depends also on other factors, particularly the fineness of preparation. If fine preparation is carried too far, the rate of percolation falls and can reach the point of causing flooding. In the same way, the thickness of the bed plays a part. As it increases, the layers at the bottom undergo a greater degree of compaction on account of the weight of the upper layers.

Flooding occurs, for example, with a given degree of preparation, at the percolation rates as shown for different degrees of compaction:

<i>Compaction</i>		<i>Percolation</i>	
<i>(kg/m³)</i>	<i>(lb./cu.ft.)</i>	<i>(m/min)</i>	<i>(in./min)</i>
60	3.7	0.4	15
70	4.4	0.25	10
80	5.0	0.187	

Shredded cane compacts rather more readily than cane treated with knives.

With the slower percolation, and the longer time taken for the juice to traverse the bed, it will be seen that the flow of the juice follows a smaller angle with the horizontal as its rate of flow decreases. This angle also varies from the head of the diffuser to the discharge end. It may be, for example, 57° at the first tray, 38° at the second, and 14° when leaving the third tray. It follows that the rate of percolation determines the tray into which the juice is discharged into the canal below. The juice coming from tray *n* will not deliver into tray *n*

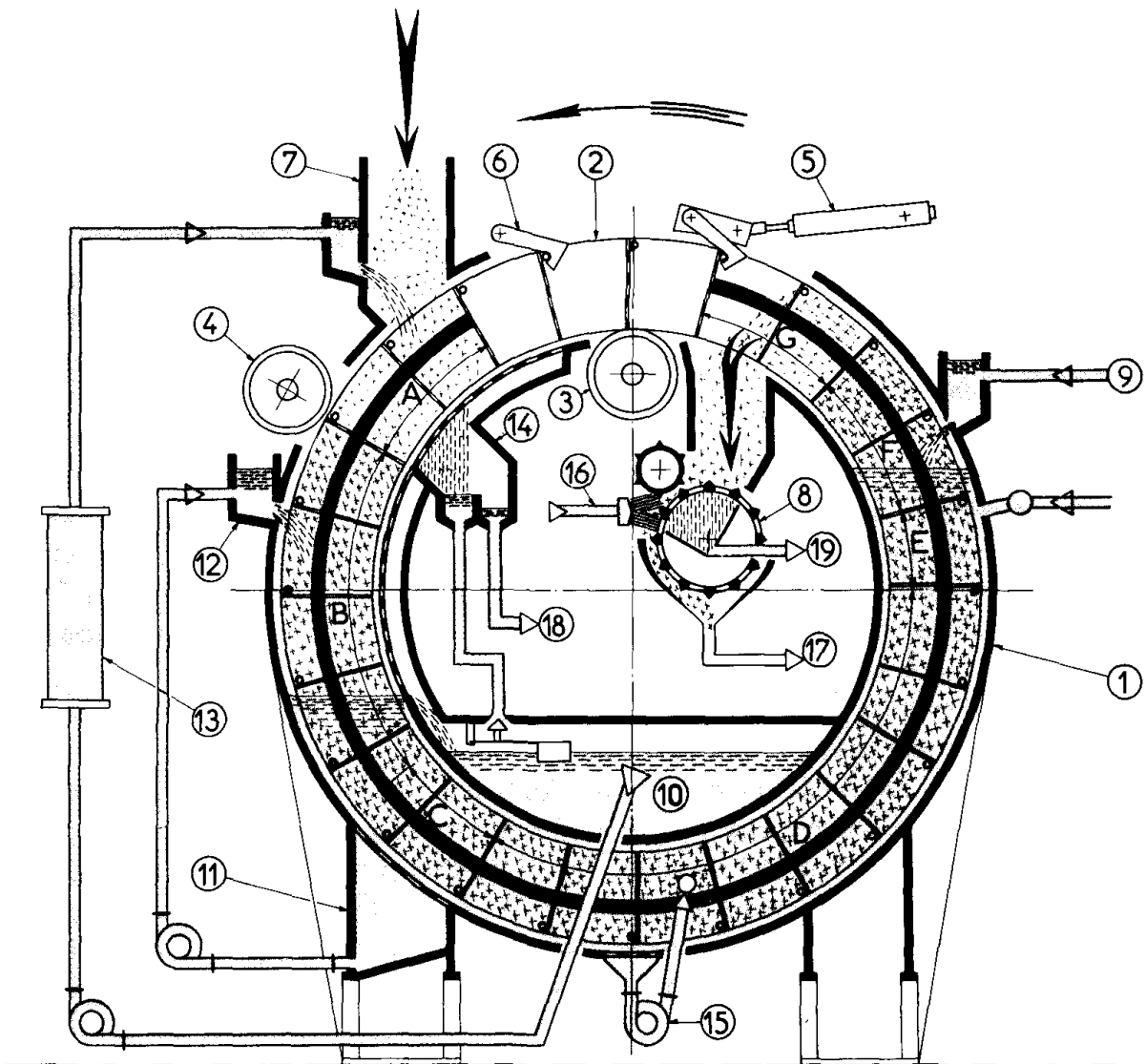


Fig. 23.21. Saturne diffuser.

- 1 except for a given rate of percolation and a given depth of layer. This is of interest in relation to the ability to advance or retard the corresponding collecting channel (*cf.* p. 384).

The moisture content of the megasse leaving the diffuser varies in the range 70–80% depending on whether or not a compression roller is provided.

Saturne diffusion

The Saturne diffuser was devised by Marcel Barre, and is built and sold by the Société Sucrière de l'Atlantique.

The diffuser (Fig. 23.21) has been termed the "Saturne". It consists of a vertical ring, of 12 m (39 ft.) external diameter and radial thickness about 1.50 m (5 ft.), rotating about a horizontal axis inside a fixed casing of rectangular section. It is divided into 18 compartments or "baskets" by grilles of perforated plate. Each basket is enclosed between the interior and outer walls of the casing (which are cylindrical in form) and its front and back grilles, which are flat; it is thus restricted in direction of movement by the grilles. When the basket has passed the top generatrix, which we shall take as the origin, or 0°, it receives the bagasse by an opening located slightly beyond; when it is about 45° further on, it is completely filled and receives at the same time the juice which has been recycled, limed and heated. Soon after it has passed the 90° position, *i.e.* the level of the shaft of the ring, the walls of the cylindrical casing become perforated plate, and the diffusion juice which fills the whole lower half of the ring escapes by these perforations, passes through a liming bath, is picked up by a pump and divided into two portions. One passes through a heater and is recycled and distributed on the bagasse to heat and saturate it; the other is withdrawn and taken to manufacture.

The basket at this stage enters the zone filled with diffusion juice and passes through it. This zone consists of the lower half of the ring and ends slightly beyond the 270° point, where the basket receives the juice from the dewatering equipment; at the 315° position it receives water. The basket then enters the discharging zone, and at the 335° position the interior casing terminates and allows the megasse to fall into a conveyor which takes it to the mills or other dewatering devices. As the outer casing is also open at this position, the basket is washed and is ready for the commencement of another cycle.

The juice thus circulates in a counter-current manner. The head necessary to maintain the flow through the bagasse moving in the opposite direction is approximately 4 m (13 ft.), and varies according to the degree of packing sought and the degree of preparation of the bagasse.

The speed of rotation of the ring is approximately 1 m/min (3 ft./min) at the periphery, giving one revolution in about 40 min. It may be varied between 0.5 and 2 m/min (1.5 and 7 ft./min), and is controlled at the delivery from the continuous scale which weighs the bagasse entering the diffuser. The rotational movement is furnished by a jack placed at the top of the ring, which acts on 18 bars connected to the 18 grilles; the movement lasts 1.5 min, the interval between movements is 0.5 min.

The ring rests and rotates on an upper roller, and is guided by two other rollers, one external and one internal. It consists essentially of a central girder of circular section, provided with numerous holes so that it does not form a dividing wall, and with the grilles bolted to it.

The power necessary to provide the movement is approximately 0.75 kW (1 h.p.) per 100

tonnes cane per day. The capacity is approximately 1,000 tonnes per day per metre of width of the ring.

The residence time for the bagasse is approximately 30 min, made up thus:

Scalding:	5 min
Maceration:	20 min
Discharge:	5 min

The residence time of the juice is about 20 min.

For recycling, about 5 kg of juice per kg of bagasse are used. A temperature of 75 – 80°C (170 – 175°F) is recommended, and a pH of 6.2.

There are two separate controls, one for the temperature and pH of the scalding juice, the other for the temperature and pH of the thin juice from the final pressing. The latter may be re-introduced directly, or treated; the treatment may be limited to screening, liming and heating.

The megasse leaving the ring contains about 85% water, and the excess juice is allowed to drain off. The material is also passed between two primary rollers provided with grooves, of which the lower, about 1.5 m (5 ft.) in diameter, is placed below and is hollow with a surface perforated with holes occupying some 30% of the surface area. The juice expressed drains into the interior of this roller and is withdrawn by inclined gutters and re-introduced into the diffuser. The pressure exerted by the small upper roller is adjustable. The moisture content of the megasse is thus reduced to about 70%.

The imbibition (Sucatlan recommend $\lambda = 2$) is applied on the megasse which is sent to a second pair of pressure rollers of similar design, but situated away from the main diffuser ring; these reduce the moisture to 65 - 70%. The megasse then goes to the final mill or mills.

The Saturne requires considerable height, but it is simple and requires little maintenance.

Fletcher - Stewart - van Hengel diffusion

This is a diffuser designed about 1968 by Van Hengel in South Africa, and built by Fletcher & Stewart.⁷ It consists of a series of units each having the form of an intermediate carrier, inclined at 30°, and to a great extent overlapping (Fig. 23.22). The conveyor proper, situated in the interior of the compartment, is furnished with scrapers and moves in such a way that the upper run of the conveyor descends and the lower run rises. The first cell receives the bagasse supplied from a mill following cane knives and a shredder, and discharges it, as do the following conveyors, mid-way on the upper run of the following conveyor. It descends on the upper run towards the bottom of the compartment where it is picked up again by the lower run and passes up to the upper end of the compartment, where it is again passed on to the following compartment. On these two conveyors it is sprayed copiously with water or mill juice in the first compartment, and with juice from the cell itself, or from the following one, in the other compartments. The imbibition may be of the order of 50%, for example, but the recirculation pumps can recycle 300-400% on weight of cane, of which, for example, 150% comes from the following cell and the remainder as recycled liquid from the compartment considered.

The megasse leaving the last compartment passes to two mills to reduce the water content

sufficiently for its use as fuel. The expressed juice thus obtained is reheated to 95°C (200°F), in order to maintain the compartments at a temperature of 80–90°C (175–195°F).

Compaction is of the order of $t = 80 \text{ kg fibre/m}^3$ of megasse (5 lb./cu.ft.). We may reckon on a movement of about 16 m (50 ft.) of bagasse per cell or compartment. The capacity of the equipment is given by:

$$1,000 Af = L \times 60v \times H \times t \times 0.8 \quad (2,240 Af = L \times 60v \times H \times t \times 0.8)$$

A = tonnes of cane per hour

f = fibre per unit of cane

L = effective length of the compartments, in m (ft.)

v = speed of movement of the bagasse in its travel, in m/min (ft./min)

H = thickness of the layer of bagasse on the conveyor, in m (ft.)

0.8 = coefficient allowing for the space between the scrapers, due to the inclination of 30° of the conveyors.

Assuming $t = 80 \text{ kg/m}^3$ (5 lb./cu.ft.), we have:

$$Af = 3.8 LvH \quad (Af = 0.107 LvH)$$

The driving wheels and the end of the intermediate carrier (which forms the mobile element of the unit) should be at least 1.16 m (46 in.) in diameter, the minimum adopted for the first installation. If they are driven at 1 r.p.m., the speed of the conveyor is $v = 3.66 \text{ m/min}$ (12 ft./min).

The residence time for the material in the diffuser should be about 25—30 min. The number of units comprising the diffuser can vary from a minimum of 5, with a recommended optimum in practice of 7. The thickness of the layer is 0.6 m (2 ft.), or about half that for diffusers of the de Smet or B.M.A. type. Thus, if we take the residence time as half an hour, we may determine the conditions of operation; if P is the distance of travel in one unit of the diffuser, and T the residence time, we have:

$$v = \frac{7 \times P}{60} = \frac{7 \times 16}{60 \times 0.5} = \frac{112}{30} = 3.73 \text{ m/min} \quad (12.25 \text{ ft./min})$$

Hence the rotational speed of the driving wheel is:

$$\frac{3.73}{3.66} = 1.02 \text{ r.p.m.}$$

and the width of the diffuser is:

$$L = \frac{Af}{3.8 vH} \quad \left(L = \frac{Af}{0.107 vH} \right)$$

This type of diffuser has several advantages: first, the low thickness of the layer facilitates

percolation, which is further improved by the large number of changes of direction (two per unit), which at each change presents a new surface, loosens and aerates the material, and so prevents the tendency to flooding, which is a frequent disadvantage where the moving layer is undisturbed. The cane preparation can thus be pushed to the maximum without fear of interference with the penetration of the bagasse layer, which would create an impermeable layer at the surface. The residence time of the material is low, thus minimising the retention of juice, and consequently the risk of inversion or microbial action; similarly, loss of time during stoppages when for a time the factory will proceed without bagasse and without juice at the start, without bagasse during the stoppage.

This diffuser especially permits of recycling a very high quantity of water or juice (Pongola has one of the highest figures for imbibition % fibre in the country): which permits making use of this important factor in the efficiency of the process.

On the other hand, the large number of intermediate carriers is complex and involves extra power for elevating the megasse in each unit, whereas straight diffusers operate at constant level; in compensation, however, fewer pumps are required, and the total power consumption of the diffuser is about 12 kW/t.f.h. (16 h.p./t.f.h.), lower than that consumed by other diffusers in South Africa.

At Pongola, the extraction at the preparatory mill was 71%, that at the diffuser 48% of the remaining pol content (or 85% total for first mill and diffuser), and total extraction 95–96%, including the two final mills. For the 1980-81 campaign, the pol in bagasse was 1.12%, overall extraction 97.55 imbibition 411% on fibre, Clerget purity of molasses 36.6, loss in molasses 9.23% of pol in cane, recovery on mixed juice 87.82, and overall recovery 85.67%.

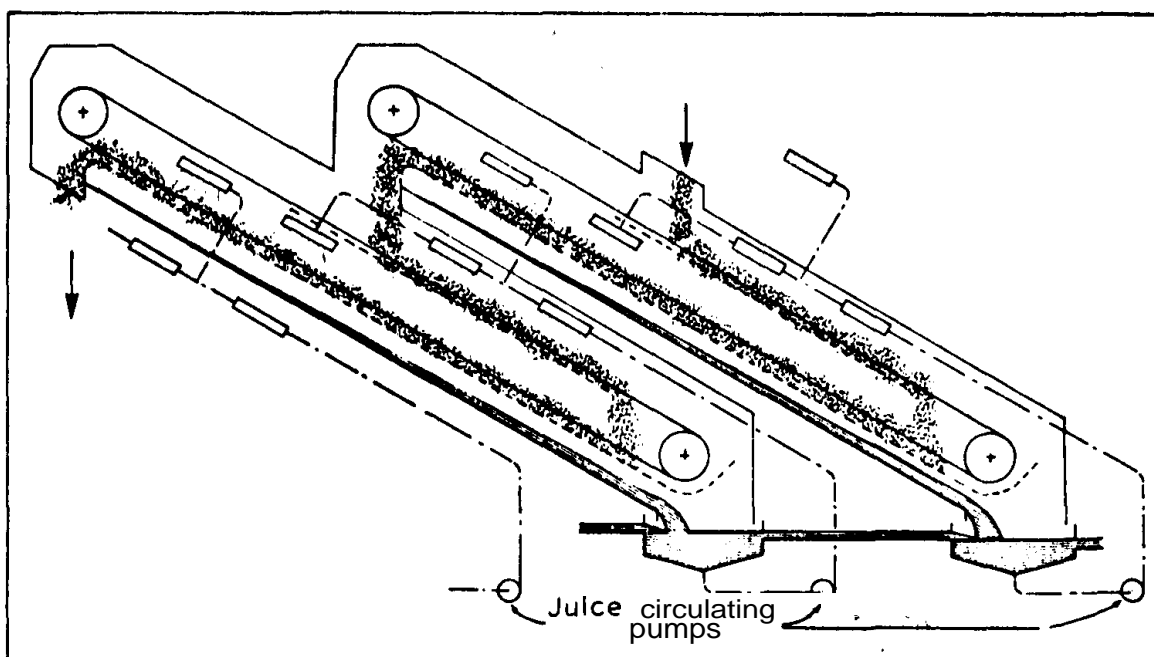


Fig. 23.22. Van Hengel diffuser (Fletcher and Stewart).

GENERAL CONSIDERATIONS ON DIFFUSION

We have described the four principal diffusion processes operating at present. Several others are in existence, some in operation, others in the form of pilot plants, many only existing on paper. Some of these extend horizontally, like the four described; others vertically, these being diffusion towers. Of these towers, one lifts the bagasse upwards by means of screw elevators, others are based on the attractive hypothesis that the bagasse will lift itself. These raise the much debated problem of the buoyancy of bagasse; if this buoyancy is real and sufficient, this type would give simple and economical processes.

In the competition between milling and diffusion, the mills have one advantage, that of extracting at each pressure the maximum quantity of juice, and then of replacing this more or less completely with water or imbibition juice, thus achieving a more marked effect. The diffuser has more difficulty in eliminating the juice enclosed in cells which have not been completely opened and in thick pieces of bagasse, but it bathes these more effectively and can repeat the washings more readily (8 – 18 washings, compared with 3 or 6 for mills).

The present trend of inventors and innovators is to attempt alternate pressure and diffusion, attempting thus to profit by the advantages of both systems. The idea is logical; in order to clean a floor mop, it is indeed easier to wet it and press it alternately than to run a continuous stream of water over it without pressure. Processes such as these, however, have not yet reached the industrial stage.

Effect of diffusion on overall recovery

Most manufacturers of diffusers have given great prominence to the gains in extraction realised, sometimes very substantial ones. They have seldom spoken of the final gain in sugar produced. Now, many pioneer users of diffusion have reported the gain in extraction, but have not found it in the sugar produced. Sometimes the extraction increased by 2-3 points, whereas the overall recovery fell by 1 or 2 points, disclosing a loss of sugar in the boiling house of 3 — 5 points. This loss was partly undetermined, partly a loss in molasses, either in purity or in the quantity of molasses produced.

The position is that diffusion is more critical to control than milling, on account of the risks of inversion and deterioration which are involved. The greatest danger lies in the contact of juice with bagasse, in a hot alkaline medium. Another risk lies in the separate circuit of impure juices, dilute and containing fine bagasse in suspension: the thin juices from the dewatering mills. For these two reasons, there is a risk of difficulties in crystallisation, particularly with B and C masseccutes, and in the exhaustion of the molasses.

In installing a diffuser, the following should therefore be especially studied: (a) the problem of liming and of the pH in the diffuser; (b) the juice circuits, where care should be taken to reduce the residence times, particularly of juice in contact with bagasse; (c) temperatures, taking care to maintain these between 70 and 80°C (158 and 176°F) (*cf.* p. 437). It is only on condition of having found good answers to these problems that the factory will be able to find in its overall recovery and its sugar production the gain in extraction anticipated from the fact of acquisition of a diffuser.

Recent experience seems to indicate that the Chief of Manufacture has little complaint as

to the quality of juice received from the Chief of the Juice Extraction Department. The purities of molasses are practically the same as in the case of milling plant. However, the quantity of molasses seems to be influenced by diffusion: South African figures⁸ for the 1979 - 80 season indicate the quantity of molasses of 85° brix % on cane as 3.87 for factories practising diffusion of cane, and 4.11 for bagasse diffusion factories, compared with 3.49 for factories using milling plant only.

For the 1980-81 season, the mean for diffusion factories showed 4.30% on cane, and for factories with mills, 4.05%, or a 6% disadvantage in the case of diffusion factories: in sucrose lost in molasses % pol in cane, 10.5 for the diffuser, and 10.0 for mills, or 5% greater with diffusion. This difference is on the debit side for the diffusion process.

On the credit side, there should be counted: (a) investment costs about 30% less for diffusion; (b) maintenance costs much lower; (c) economy in juice heating, due to the high temperature of diffusion juices; (d) economy in filtration, due to the lower proportion of muds, because of the filtering effect on the bed of megasse during the percolation (3.26% as against 5.26%); and (e) economy in power⁹ concerning the diffusers proper: there is a difference between cane diffusers and bagasse diffusers of 23 kW/t.f.h. (30 h.p./t.f.h.) in the former, as against 18 (24) for the latter (mean of 5 factories in each case).

As regards total power expended, Lamusse⁹ gives a mean for factories with mills of 190 kW/t.f.h. (255 h.p./t.f.h.), compared with 155 (205) for diffusion factories.

Capital and maintenance costs

It has been commented that the cost of diffusion plant is lower. The 30% reduction in cost relates to diffusion of cane. For the case of an existing factory desiring to change to diffusion of bagasse, it is appropriate to compare the normal tandem, with its cane preparators and its 5 or 6 mills, with the same tandem where only 3 mills and the preparatory plant are retained. To show a gain in the capital cost, it is necessary therefore that the cost of the diffuser should be less than the cost of 2 mills with their intermediate carrier, their drives and gearing; or, if an 18-roller tandem is being considered, the cost of 3 mills, with drives and gearing and 2 intermediate carriers. Now, the cost of the diffuser is close to that of 2 or 2½ mills. The initial cost is thus roughly equivalent, or lower for the diffusion plant, since it would require six mills to give an equivalent extraction.

Maintenance of the diffuser is also less costly. The difference here is considerable, on account of the cost of rollers, trashplates and scrapers, and parts for the steam turbines when these are used for mill drives. It is reported that for diffusion of cane, maintenance costs are only 50 - 80% of those for an equivalent tandem of mills.

The setting of final dewatering mills differs from that of ordinary mills. It is commented in South Africa¹⁰ that the ratio of feed opening to delivery opening is of the order of 3 : 1 or 4 : 1, as against 2 : 1 or 2.5 : 1 for ordinary mills.

Specific heat of prepared cane. This must be known for calculating the heat balance for diffusion¹¹. It has a value $c = 0.25 + 0.81h$ cal/g/°C, where h = water content of cane (approximately 72%). Hence: $c = 0.86$ approximately.

Screen area. In South Africa¹², the capacity of diffusers is related to their screen area per

unit of fibre treated. A mean figure of 11.1 m² (121 sq.ft.) per t.f.h. is reported, with values varying in the range 4.6 – 14.8; good results are noted with areas ranging from 5.8 to 10.6 m²/t.f.h. (62 – 114 sq.ft./t.f.h.).

CHOICE OF MILLS OR DIFFUSION

In spite of the years that have passed since the installation of the first diffusion equipment for cane, and the number of installations which have followed, technologists are not yet in agreement on the decision to take when the question arises of whether to instal mills or dif-fusing equipment. Each system has its supporters. We can, however, enunciate some general ideas.

The majority of factories, operating with a mill tandem, are inclined to remain with milling plant, to which they are accustomed, even when they have increased in capacity and are at a stage where they have to instal further mills or larger mills.

If the milling tandem is old, worn, obsolete or of low efficiency, there is a clearer case for deciding between the two solutions. The considerations in the preceding paragraphs will assist to make the right decision. It will be easier to see whether the new equipment can be installed without too much difficulty or concern regarding the equipment retained.

For a completely new factory, diffusion appears more attractive on account of the lower cost of installation and maintenance. The only argument against it consists of the loss of sugar in molasses, which seems to be definitely slightly higher. The difference is some 0.7 kg (1.5 lb.) of sucrose per tonne of cane, say 120.7 compared with 120 kg/tonne cane. The overall economy with diffusion seems generally higher.

Comment. Diffusion of cane presents one rather serious disadvantage; there is no primary juice. Analysis of primary juice normally gives the basis for determining the overall efficiency of the factory, as expressed by our coefficient k , the use of which is important and is strongly recommended (see pp. 342 – 350).

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24. *Defecation*

CLARIFICATION OF JUICES

Hundreds of materials have been tried for purification of juices. There are, however, only five which are of any industrial importance:

(7) Lime (CaO) which, since the beginnings of sugar manufacture, has remained the universal basic defecant; the treatment with lime is called "defecation"

(2) Sulphurous acid, from SO₂: "sulphitation"

(3) Phosphoric acid, from P₂O₅: "phosphatation"

(4) Carbonic acid, from CO₂: "carbonatation"

(J) Magnesia, MgO.

We shall study these various processes separately. We shall commence with defecation: this is the only one of the five methods mentioned which is always practised.

ACTION OF LIME ON THE JUICE

We shall not discuss the chemical effects resulting from the reaction of lime on the juice. Many of the organic acids are eliminated, since their lime salts are insoluble (oxalic, tartaric, etc.); albuminoid matter is coagulated. Part of the pectin and colouring matter is destroyed or rendered insoluble. But this elimination of impurities is relatively insignificant, the purity of the defecated juice being approximately the same as that of the juice before treatment. The gain is generally of the order of 0.5–2 points: 83–84, for example.

The purification is predominantly physical. A precipitate forms, readily demonstrated in a test-tube; the coagulated material entraps suspended impurities, and brings them down with it.

Lime

The quality of lime used is important. In many countries the only lime obtainable is very impure, approximately 60% CaO, containing much sand and unburnt stone.

It is recommended that lime containing more than 2% of MgO or of oxides of iron and aluminium should be avoided. These impurities would cause deposits in the multiple effects, and magnesia would give trouble in the defecation. We shall see, however (p. 429), that magnesia is sometimes used systematically with lime, precisely in order to avoid scale formation in the evaporators.

A good-quality lime should test 90–95% of CaO. Hydrated lime, Ca(OH)₂, is also employed, its activity being proportional to its CaO content.

Pure hydrated lime contains $\frac{56}{74} = 76\%$ of CaO.

Milk of lime

Some factories still use lime directly, adding it to the juice in the solid state.

The solubility of lime in juice increases with the sugar content and decreases with increasing temperature. At 80°C (176°F), 0.25–0.30% of CaO dissolves in a juice of 10–12% sucrose.

As solution is relatively slow and difficult, the use of lime in the pulverised state is not recommended; certain portions of the juice will be overlimed, other insufficiently limed. It is of advantage to make first a milk of lime, by mixing the pulverised lime or lump quicklime with water. It is this milk of lime which will then be mixed with the juice; its mixing will be much more rapid and its action much more uniform.

The milk of lime is made up in 2 tanks, one of which is emptied while the other is being filled. They are provided with a stirrer rotating at 8–10 r.p.m., according to the diameter. The capacity of each of these 2 tanks should correspond to 1½ or 2 hours' operation of the factory.

The bearings designed to work in the lime should be of cast iron and not of bronze.

TABLE 24.1

MILK OF LIME

<i>Baume</i>	<i>Density</i>	<i>g CaO per litre</i>	<i>% CaO by weight</i>	<i>kg water reqd. per kg CaO</i>
1	1.007	7.5	0.745	133
2	1.014	16.5	1.64	60
3	1.021	26	2.54	38
5	1.036	46	4.43	21.6
10	1.074	94	8.74	10.4
15	1.117	148	13.26	6.5
20	1.161	206	17.72	4.6

Density. Table 24.1 gives the relation between degrees Baume and lime content of the milk.

Milk of lime is generally made at 15° Baume. A density of 20° Be is not exceeded, as above this value the pumps and pipe lines block too frequently.

When an automatic pH regulator of the “Micromax” type is used, a milk of lime of 10° Be may be used for the fixed initial dose, but a value of 5° Be should not be exceeded for the variable addition regulated by the automatic controller, otherwise control would be defective. It is often kept at 2 or 3° Be.

Lime consumption. For the manufacture of raw sugar by ordinary defecation, one should allow for 0.5–0.8 kg (1.1–1.8 lb.) of CaO per t.c. The aim should be to use the minimum of lime which will give good clarification and good settling, with a pH of clarified juice close to 7.0, since any excessive lime is harmful and results in an increase in the content of lime in clarified juice. Honig¹ estimates that, in a well-conducted defecation factory, the lime content of clarified juice should not exceed 400 mg/l.

DEFECATION PROCEDURES

The defecation procedures most used are the following:

(1) Cold liming

The juice leaving the mills is generally at a pH in the neighbourhood of 5.5. Lime is added to a pH of 7.2–8.3, most often to about 7.8. It is then pumped to heaters, the heating surface of which should be sufficient to bring the juice to its boiling point, *i.e.* to 100.5°C (213°F) minimum. For safety, one always aims at 103–104.5°C (218–220°F). On leaving the heater, the juice passes into a vapour separator or "flash tank", a simple tank placed in the pipe line, in which it is released to atmospheric pressure and returns to its corresponding boiling point, say 100.5°C, with release of steam. It passes immediately afterwards to the subsidiers, where the precipitate formed by the lime is settled.

The defecated juice, or clarified juice, drops to a variable pH, of the order of 6.8–7.2.

The subsidation is complete at the end of:

1 hour approximately for juices of noble canes (which have practically disappeared)

1½–2 hours for ordinary canes

More than 2 hours for juice from refractory canes.

Cold liming is generally sufficient for the first two cases, when manufacturing raw sugar.

When the juices are difficult to treat, the lime is increased to raise the pH to 8.3. This system has given good results in Hawaii².

(2) Hot liming

Better clarification is often obtained by pumping the juice first to a heater heated by vapour bled from the second body of the multiple effects, so as to obtain a temperature of about 70°C (160°F), and then proceeding to add the lime to the same pH value (*e.g.* 7.8) and finally heating further up to 103°C (218°F).

In Java, the first heating is taken to 90°C (195°F) or even to boiling, and the second heating is then omitted.

Hot liming, according to Jenkins³, requires 15–20% less lime, for a better clarification.

The author at present uses this process, which gives a slight improvement when the juices are difficult and cold liming does not give satisfactory clarification.

(3) Fractional liming with double heating

This process has been recommended and put into operation at the College of Tropical Agriculture in Trinidad. It is intended especially for treatment of refractory juices, and generally gives a substantial improvement when the ordinary liming becomes insufficient. It consists of:

Liming the cold juice to a pH of 6.2–6.4

Heating to boiling

Re-liming to a pH of 7.6–8.2

Heating again to boiling

Leaving to settle.

The pH values indicated correspond to adding about one-third of the total quantity of lime

in the first addition, or pre-liming, and two-thirds at the second addition, or the liming proper.

The pre-liming may be carried as far as pH 6.6, but this point should not be exceeded. In the great majority of cases, a pH of 6.3 gives every satisfaction.

The first heating may be taken only to 93°C (200°F). This was the procedure employed at Central Guanica. Our own experience is that clarification is always less effective if the temperature falls below 98°C (208°F). For safety, it is preferable to aim at 104.5°C (220°F).

The second liming may be taken beyond 8.2, but the optimum value is generally in the neighbourhood of 7.8. If the juice is limed to 8.4, the defecated juice remains alkaline: 7.2-7.4.

It is imperative that the second heating should exceed the boiling point, and it will be preferable to insist on 104.5°C so as not to risk falling below 101°C.

The final pH of the clarified juice should be about 6.8–7.1.

Advantages. This procedure has the following advantages over cold liming:

- (1) Less scum is obtained
- (2) The clarified juice is much more brilliant
- (3) The muds filter better, giving dry and porous cakes
- (4) The nitrogenous colloids are separated to a much greater extent: about 80% instead of 50%.
- (5) Waxes are eliminated in a still greater proportion: 90% instead of 70%.

Rupture of flocs. The advocates of this procedure, especially Davies⁴ at the Trinidad College of Agriculture, have shown that, once the liming is completed, any violent agitation of the treated juice, such as passage through a plunger pump, or particularly a centrifugal pump, does considerable harm to the clarification.

It is recommended therefore that the installation should be designed in such a way that, once the pre-liming is completed, the juice should pass to the main secondary liming and to the second heating by gravity. This is obviously not easy to arrange, at least with ordinary heaters.

This rupture of the floc would be equally liable to occur in the case of ordinary cold liming.

These results have been confirmed in India⁵: the use of a pump rendered the juice more difficult to settle and increased the volume of muds as well as the percentage of colloids, reducing the rise in purity obtained by the treatment.

(4) Liming while boiling

Establishment of this fact has led to a modification of hot liming which avoids any rupture of the floc. This consists of liming the juice, not at 70°C (160°F), but after heating, in other words, after boiling at 100.5°C (213°F). The heated juice is pumped to a tank above the subsider, into which the milk of lime is also pumped in such a way as to obtain a complete and homogeneous mixture with the juice. From this tank, the juice flows by gravity, preferably by an inclined rather than a vertical pipe, into the subsider. No time of contact is provided, this being assured in the flocculation chamber of the clarifier.

It is necessary to cool the continuous sample of juice taken for the pH recorder-controller,

for example, by allowing it to cascade down the stepped bottom of a small trough, fitted with a double bottom with water circulating in the double bottom. For the pH instrument, special high-temperature electrodes are used, the glass electrode being of soda-free glass.

This procedure, in general use in Mauritius in 1966, gives a beautiful clarified juice, and has replaced all other methods in that country.

(5) Java method

The following is a method which, according to Honig⁶, promised in 1939 to become general in Java:

Pre-liming of the juice to between pH 6.0 and 6.6

Separation of the pre-limed juice into 2 portions:

(1) 40% limed cold to pH 9.5

(2) 60% heated

Mixing of the 2 portions, giving a pH of 7.6–7.8 and a temperature of 65°C (150°F).

(6) Lime addition with saccharate

This method of liming has been used in **Australia**, Mauritius and Reunion, and recently in **South Africa**⁷: it is now (1982) very general in Australia. It consists of mixing the milk of lime with clear juice or syrup, maintaining it in contact for up to 5 minutes to give time for the saccharate to form, and the lime addition to the juice is then made with this mixture. **Pongola** factory, which introduced the process in South Africa, has reported that the defecated juice was clearer and nearly one point higher in purity. A further advantage lies in the handling of the mixture, which avoids the trouble of blockages in the pipes, as experienced with the normal milk of lime mixture.

It is necessary that the lime should be in the monocalcium saccharate form, and for this the temperature must be lower than 58°C (136°F) and the ratio of sucrose to CaO should be above 6.1 : 1. A ratio of 7 : 1 is adopted and the lime prepared at 15° Be, say 148 g CaO per litre. Pongola uses 700 g CaO per t.c.

The quantity of milk of lime per 100 t.c.h. is then:

$$\frac{100 \times 700}{148} = 473 \text{ l/h (105 gal./h)}$$

and the quantity of juice at brix B , purity P and density d :

$$\frac{100 \times 0.7 \times 7 \times 100}{BPd} = q \text{ l/h}$$

The mixing tank should have a working volume of:

$$V = \frac{5}{60} (473 + q)$$

Juice for the mixture may be the cloudy filtrate from the vacuum filters; this may resolve, at least partially, the problem of disposal of that filtrate.

REACTION OF DEFECATED JUICE

The final acidity or alkalinity of the defecated juice going to the evaporators is not unimportant.

When white sugar is required, an acid juice is desirable, and the pH should be maintained between 6.4 and 6.6.

If only a good-quality raw sugar is required, the juice may be left slightly alkaline, with a pH between 7.1 and 7.5. This avoids all risk of loss by inversion, which is possible in the case of juices at pH 6.9.

On the other hand, alkaline juice slows down the work at the pans, and crystallisation is less easy. For a pH of 7.5, boiling times 20% longer may be expected compared with a pH of 7.0.

PH CONTROL

Control of juice pH is effected by automatic pH controllers. These instruments, such as the Leeds and Northrup "Micromax" or Kent "Mullelec", record on a chart the indications of a galvanometer; they demand careful maintenance and adjustment, but they give precise values. They are far more valuable if used with automatic lime regulators of the same make; the combination functions very effectively and gives much improved clarification.

This type of equipment is made by numerous firms: Kent, Foxborough, Beckman, Honeywell, etc. The measuring electrodes become rapidly dirty, and it is recommended that they be cleaned twice per week with hydrochloric acid.

It is also desirable to check the pH of the settled mud. If it is lower than that of the clear juice by more than 0.2, it is liable to cause some inversion. It is then necessary to reduce the level of mud in the clarifier.

ELIMINATION OF STARCH

Starch is sometimes found in the juices, depending on the soil and the cane variety. The normal content is of the order of 200–300 p.p.m. on mixed juice. Starch is a very detrimental material, since it is not eliminated in normal defecation and eventually remains in the raw sugar and reduces its filterability.

In South Africa, Boyes⁸ indicates a simple method of removing 50-60% of the starch from the juice. It consists of passing the filtrate from the rotary filter into the mixed-juice tank, which reduces the acidity of the mixed juice or even renders it alkaline, and enables a reaction to take place between the diastase contained in the juice and the gelatinised starch in the filtrate. The juice is then heated to about 70–80°C (160- 175°F) and is allowed to stand for 8–12 min. The lime is added only after this.

A method developed in the same country and which, from the aspect of starch elimination, would have an efficiency of 90%, is the Rabe process. As employed at Umzimkulu, it consists of an additional step in the usual process of defecation with lime, phosphoric acid and

Separan; the juice, after liming, phosphating and heating to 65°C (149°F), is pumped to a mixer where it receives the flocculant in a small subsider, of approximately 90 l/t.c.h. (20 gal./t.c.h.); it is held here for approximately 6 min only, and vacuum is applied. The air saturating the juice is thus liberated and rises to the surface as bubbles, entraining the precipitates which are thus removed in the form of scums, while the clear juice is withdrawn from below. It is then heated to boiling and goes to a normal subsider.

This process depends on the principle that starch is brought into solution only at 67°C (152°F); it is thus a case of getting rid of it before the juice is brought to that temperature.

This process has been found too expensive due to the consumption of phosphoric acid and has been abandoned during the 1970's, the same improvement in filterability having been obtained by the addition of amylase to the juice.

OTHER REAGENTS USED

Apart from lime, which we have just studied, and SO_2 , P_2O_5 , CO_2 and MgO , which are discussed in the following chapters, the following reagents are also used in various parts of the world, to an extent sufficiently important to warrant discussion:

(1) **Soda ash.** It may be advantageous to replace part of the lime by soda ash (sodium carbonate) when it is necessary to treat canes where the juice has been affected by frost or by a long delay in the field after cutting, or burnt cane which has been delayed in transport, or generally cane with abnormal juice. It is preferable, however, to avoid use of soda ash as a regular practice, since it is expensive, causes some coloration of the juice and consequently of the sugar, and tends to decrease the filterability and consequently the refining qualities of the sugar produced.

(2) **Bentonite.** This is an aluminium magnesium silicate, which has similarly been added in clarification in Puerto Rico, where certain factories have expressed complete satisfaction with it.

(3) **Separan AP 30.** This is a coagulant added to the juice before clarifying or to the mud before filtration. It is expensive, but appears to improve the clarity of the juice and the subsidence. It forms an integral part of the **Rapi-Floc** process (*cf.* p. 485). In present practice, it is used in the proportion of 1-3 p.p.m.; the following dosages are recommended: ⁹

(a) Normal juice, 0.1-0.2 p.p.m. (to improve capacity)

(b) Refractory juices, 1-1.5 p.p.m.

(c) Maximal usage, 3 p.p.m.

Many other products achieve exactly the same result. We may mention Talosep A3 or A5, Magnafloc LT 26 or 28, Cyanamid A 110 or 130, and Fabcon Zuchar.

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25. Sulphitation

Defecation is a universal and indispensable process. With sulphitation it is different; most of the world's sugar factories do without it. But it is the most widespread process auxiliary to defecation.

Action of sulphurous acid on juice

The action of sulphur dioxide, SO_2 , on juice is as follows:

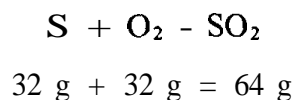
- (1) It eliminates colouring matters (a property common to all acids)
- (2) It reduces to colourless compounds the ferric salts which have been formed by contact with mills, tanks and pipes.

Sulphur

Sulphur dioxide is prepared from sulphur, which is delivered in the form of small sticks or cylinders. Sicilian or American sulphur is superior to that from India and Java. Sicilian sulphur in fact is in the form of octahedral or α -type sulphur, the melting point of which (114°C) is well below that of β or prismatic sulphur and of the precipitated or γ -form (200°C).

Combustion of sulphur

Sulphur dioxide is a gas resulting from the combustion of sulphur:



Hence, 1 kg of sulphur requires 1 kg of oxygen; the reaction releases 2,217 kcal per kg of sulphur (3,990 B.T.U./lb.).

Combustion takes place at a temperature of 363°C (685°F).

If the combustion were to take place with the theoretical quantity of air, the weight of air required would be:

$$\frac{100}{23.15} = 4.3 \text{ times the weight of sulphur}$$

(air contains 23.15% of oxygen by weight).

Similarly, the maximum volume of sulphur dioxide in the combustion gases is 21% (air contains 20.84% of oxygen, by volume).

However, it would not be possible to obtain complete combustion with this amount of ox-

ygen, and it is necessary to allow an excess of air above theoretical. If it proved necessary to use 100% excess air, a weight of air equal to eight or nine times the weight of sulphur would have to be introduced, and one would obtain only 10% of SO₂ in the gas. Generally, 12- 16% is obtained, or an average of 14%.

Precaution. In the presence of water, sulphur burns to give sulphuric anhydride, SO₃, instead of sulphurous anhydride, SO₂. On reacting with water, SO₃ immediately forms sulphuric acid. It is therefore necessary to exercise the greatest possible care to avoid the introduction of water into the sulphur furnace.

Accordingly, the air for combustion of the sulphur should always be dried. For this purpose, the air is passed over quicklime, which has a great affinity for water and absorbs the moisture from the air.

It is advisable to renew the quicklime before it has become completely saturated. This may be done for example at each change of shift, that is, every 8 h. This is a detail which does not always receive sufficient attention in certain sulphitation factories, and it proves very expensive in loss of sugar, and in corrosion of plates and of pipes.

Sulphur furnaces

In Fig. 25.1 is shown a sulphur furnace of the type most commonly used, being that furnished with the Quarez sulphitation apparatus.

It is composed of trays in the form of drawers sliding in suitable guides. On the lower tray, by which the combustion air enters, is placed the quicklime. The air then passes to the upper plate, on which is spread the sulphur. Baffles deflect the air and force it to pass in contact with the quicklime and the sulphur. In the absence of these baffles the output of the furnace *would* fall by half.

At the outlet of the sulphur compartment the gas, which now contains 12 – 16% of SO₂, rises to the aspirator of the sulphitation apparatus, by way of a sublimation chamber and a jacketed water-cooled flue.

The water which has served to cool this flue then passes over the furnace, which it cools in turn.

These precautions have the object of:

(1) Protecting the metal of the furnaces, which would corrode very quickly if it were taken to red heat.

(2) Condensing the portion of the sulphur which has sublimed. In the furnace, the greater part of the sulphur burns to form SO₂, but there is always some which sublimes and rises in the form of vapour, especially if insufficient air is admitted.

(3) Cooling the sulphurous gas before allowing it to come into contact with the juice.

Temperature. At 1,200°C (2,200°F), sulphur dioxide decomposes:



and the nascent oxygen immediately reacts with the ambient SO₂ to produce sulphur trioxide, or sulphuric anhydride:



This reaction begins to be noticeable above 900°C (1,650°F). This is a further important reason against allowing any part of the furnace to reach red heat, or forcing the combustion to a local high rate (this is possible with certain furnaces using compressed air supplied through nozzles).

To remove any sulphuric acid which may have formed, the gas is generally washed by bubbling through water. The Quarez furnaces, in which combustion is well controlled, do not provide washers, and the gas which they produce is practically free from sulphuric acid, if the furnace is well operated.

The furnace temperature is 320 – 350°C (610 – 660°F), with 260 – 290°C (500 – 550°F) at the outlet. A temperature of 300°C (570°F) should not be exceeded at this point, as this would cause sublimation.

After the cooler, the gas temperature should drop to between 100 and 205°C (212 – 400°F).

Capacity. For optimum performance, the tray area for a sulphur furnace should provide for a capacity of 20 – 25 kg sulphur/m²/h (4 – 5 lb./sq.ft./h). It is sometimes possible to work at 35 kg/m²/h (7 lb./sq.ft./h) (Honig¹).

Where compressed air is used, air requirements may be estimated at 6 m³ (assumed at 0°C and 760 mm pressure) per kg of sulphur (100 cu.ft./lb.). This quantity corresponds to a gas of about 12% SO₂.

Lime consumption. In practice, this may be estimated at 0.3 kg/m³ (0.02 lb./cu.ft.) of air introduced into the furnace.

This corresponds to about 1.75 kg CaO/kg sulphur. The consumption of quicklime therefore should be about twice that of sulphur.

Sulphur consumption. The quantity of sulphur consumed depends on the clarification procedure adopted, on the final acidity sought, and on the quantity of lime added in the defecation.

For a given final pH of juice, it may be reckoned that each kg of sulphur used corresponds to an additional consumption of 2.2 lb of lime².

In general, when sulphitation of juices is practised, sulphur consumption is 0.25 — 0.50 parts per 1,000 of cane, or 0.3 – 0.45 kg per tonne cane (0.6 – 1.0 lb./ton).

If both juice and syrup are sulphited, about 0.9 kg (2 lb.) of sulphur is used per tonne of cane.

Sulphitation apparatus

Formerly, the sulphitation apparatus almost universally adopted in French sugar factories was the Quarez, so called after the name of its inventor (Fig. 25.1).

It consists of a rectangular tank divided into 2 unequal and connected compartments. The juice to be sulphited is fed into the smaller compartment. It is picked up by a pump which takes it to the SO₂ aspirator. This is designed in the form of a nozzle, on the usual principle of an ejector. Thus it produces an aspiration of the sulphurous gas, and the sulphitation takes

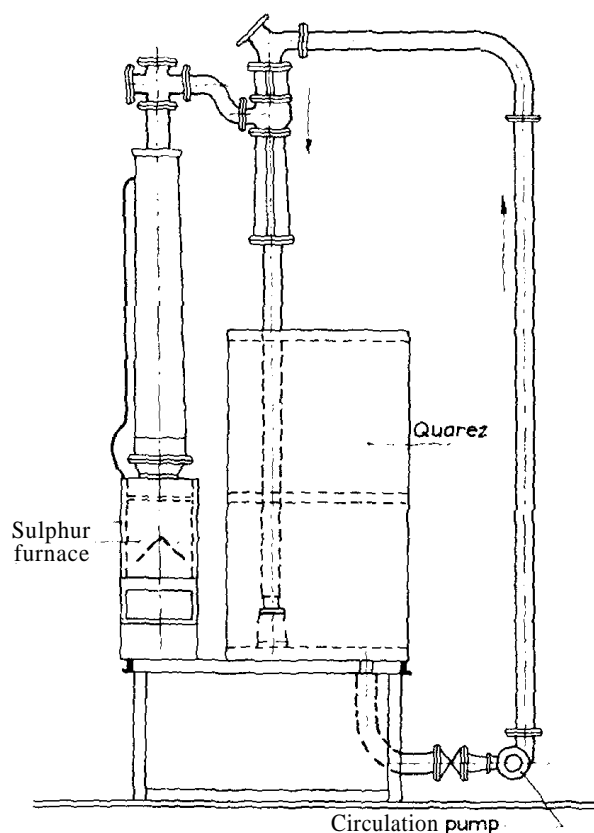


Fig. 25.1. Quarez and sulphur furnace.

place by contact and mixing in the vertical descending column returning the juice to the tank.

The nitrogen and oxygen accompanying the SO_2 in the furnace gases are released as bubbles at the bottom of this column and are rejected to the atmosphere by a small vent pipe.

The pump is designated by the name "Quarez circulation pump". Its output regulates the degree of sulphitation obtained. The SO_2 content of the juice will therefore be varied by regulating the speed of this pump.

In general, it should be capable of an output at least equal to 1.5 times the volume of juice to be treated.

The Quarez is now being replaced by a sulphitation column, which eliminates the circulation pump, and the standard sulphur furnace with trays is being replaced by a rotary furnace, which is more efficient (Fig. 25.2).

Sulphitation procedures

The standard sulphitation process was the following:

- (a) **Cold sulphitation.** Starting with the mixed juice:
Sulphite to pH of 3.8 (control by methyl orange or bromphenol blue)
Lime to pH of 7.0 (control by bromcresol purple or phenol red)
Heat to boiling and pump to the clarifier.

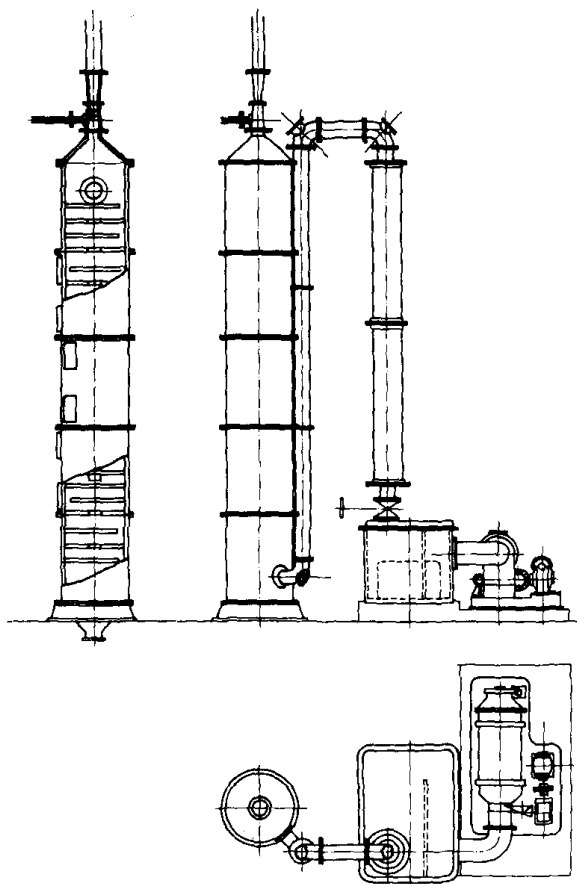


Fig. 25.2. Rotary sulphur furnace and sulphitation column (Fives Cail- Babcock).

The sulphitation may be stopped at a pH of 4.0 or 4.5 according to the results obtained. The final pH sought is 6.9–7.0. In other words, the juice passing to the heaters should be neutral, or slightly acid.

Before the advent of pH control, this point was obtained by checking that the juice did not turn phenolphthalein paper pink, nor turn litmus paper blue.

(b) Hot sulphitation. Sulphitation involves the production of calcium sulphite. This salt is more soluble cold than hot, the minimum solubility occurring in the neighbourhood of 75°C (165°F); it is thus deposited in the tubes of the heaters, in cold sulphitation.

To avoid this drawback, the standard sulphitation process is modified as follows. Starting from the mixed juice, it undergoes:

Heating to 70–80°C (160–175°F)

Sulphitation

Liming

Heating to boiling

Pumping to clarifier.

In 1945, the sulphitation procedure in use in Java was the following³:

Heating to 50°C (122°F)

Liming

Sulphitation to pH of 7.2

Reheating to boiling

Settling.

From the point of view of colour and colloid elimination, the Java procedure, investigated in the laboratory in India, was judged to be the best sulphitation process.

Hot sulphitation reduces appreciably the quantities of lime and SO_2 required.

(c) Fractional liming and sulphitation. Hot sulphitation is sometimes used with success in the clarification of refractory juices. When it proves insufficient, better results are sometimes obtained by modifications similar to the following procedure, for example, which was developed in India⁴:

Pre-liming to pH of 8.0

Heating to 50 or 70°C (122 or 158°F)

Sulphitation to pH of 5.1 – 5.3

Second liming to pH of 7.0 – 7.2

Reheating to boiling.

In Mauritius, where the juices are easier to treat, the following has been devised:

Heating to 55 or 60°C (130° or 140°F)

Pre-liming to 0.4 g CaO per litre

Sulphitation to 0.4 g SO_2 per litre

Final liming to pH of 7.4

Reheating to boiling.

Should sulphitation be done before or after liming?

Sulphitation may equally well be done before liming, or liming before sulphitation. The important point is that the same final pH should be reached in the juice going to the subsidiers.

However, it is not entirely unimportant which method is followed. The rapidity of settling and the volume of muds depend on the reaction of the medium in which the precipitate is formed:

(a) If it is formed in an alkaline medium, it will be hydrated, hence an increase in the volume of muds

(b) If formed in an acid medium, the speed of settling will be greater and the muds more compact.

In practice, the following differences are acknowledged:

(1) Liming before sulphitation. Settling is slower; volume of muds is greater, requiring a greater capacity in filter presses.

Hot pre-liming decreases these disadvantages.

(2) Sulphitation before liming. It follows that, in general, the second process is preferable. It is particularly indicated in the case of treating immature cane, as is the general practice in Louisiana.

A good coagulation of impurities requires that before the addition of lime, the sulphitation should be made:

(a) At high acidity (pH 3.8 – 4.0) and low temperature (30 – 40°C; 86- 104°F)

(b) At low acidity (pH 5.1 – 5.3) and high temperature (70 – 75°C; 160- 165°F).

From the point of view of impurity elimination, which is measured by rise in purity between raw and clarified juice, reports are conflicting. For example, we have noted two contradictory points of view coming from the same country (India^{5,6}). It is probable, then, that results alter according to circumstances, and it is best to try both methods.

Indian technologists moreover insist that the two operations, liming and sulphitation, should be carried out *simultaneously*, as nearly as possible. They are carried out in the one vessel, and this ensures that the results are superior to those obtained either with sulphitation first or with liming first.

It is important to ensure above all that the juice is not held at high alkalinity and high temperature at the same time, as this would destroy reducing sugars and would increase formation of colour.

Advantages and disadvantages of sulphitation

On comparing the practical results of sulphitation with those of ordinary defecation, the following advantages and disadvantages may be noted:

Advantages:

- (1) The juice settles more rapidly (hence improved capacity of subsidors)
- (2) The massecuites are less viscous and boil faster
- (3) Better crystallisation in consequence
- (4) Marked improvement in colour of the sugar
- (5) Savings in time in clarification and boiling
- (6) Slight gain in capacity of centrifugals
- (7) Better elimination of phosphates and waxes (Douwes Dekker⁷), leading to better refining quality and filterability of the sugar produced.

Disadvantages:

- (1) Much heavier deposits in heaters; these may be avoided by sulphiting hot, but this involves an increase in the heating surface required
- (2) Higher ash content of the sugar obtained⁸
- (3) Greater expense (Quarez, circulation pump, sulphur, corrosion of vessels and pipes, etc.).

Sulphitation of syrup

One of the best methods of obtaining white sugar is to proceed to a second sulphitation, after evaporation.

Lime is added, and sulphur dioxide. The operation is generally carried out on the syrup, after the multiple effects, but in Hawaii sulphitation of juice from the 3rd effect has been practised, removing this juice from the 3rd effect and returning it continuously.

Lime and SO₂ were added simultaneously, maintaining the pH in the neighbourhood of 7.0. The quantity of lime varied from 1.6 to 2.0 kg/tonne cane (3.5 – 4.5 lb./ton), and that of sulphur from 1.4 to 1.8 kg/tonne (3 – 4 lb./ton)⁹. The juice treated was filtered on a Vallez

filter, and the cakes obtained sent to the Oliver filter used in clarification. On entering the 4th effect, the pH was 7.1.

The processes vary somewhat, but generally consist of heating, sulphiting, and filtering. Deerr¹⁰ recommends adding milk of lime to the extent of 2–2.5% on weight of syrup, or 0.3–0.4% of quicklime. Sulphitation to neutrality follows immediately; the dense precipitate formed is filtered, and the juice is re-sulphited to a slight acidity.

In India¹¹ the syrup is maintained between 6.0 and 5.5 pH, and consumes only 0.1–0.2 kg of sulphur/tonne cane (0.25–0.4 lb./ton).

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26. Clarification with Phosphoric Acid

Role of phosphoric acid in the juice

Phosphoric acid occurs in the cane in two principal forms:

- (a) Soluble phosphates of the juice
- (b) In combination as protein in the cell material.

The latter compounds being insoluble, only the soluble phosphates take part in the defecation. They react with the lime and form a precipitate which constitutes an important part of the floc produced in the juice by the lime.

It has been shown that the greater the amount of phosphoric acid in the juice, the easier is the clarification. The mean phosphate content of normal juice is approximately 250 mg P_2O_5 per litre.

Unfortunately, certain varieties of cane give a juice poor in phosphoric acid, and which is difficult to treat. These are the canes termed "refractory", typical of which is POJ 2878.

These juices are improved by adding to them artificially, before defecation, the quantity of phosphoric acid by which they are deficient.

Quantity of phosphoric acid necessary

This quantity will therefore vary, depending on the variety of cane and the circumstances. It is considered in general that at least 300 mg of P_2O_5 are required per litre of juice. Now, certain canes, such as POJ 2878 and Co 290, have generally only 200 mg, and sometimes as little as 50 mg. It is therefore necessary to add the balance.

Action of phosphoric acid on the juice

The phosphoric acid added to the juice precipitates part of the colloids and colouring matter which it contains.

The precipitate formed with lime is mainly tricalcium phosphate. This unfortunately is a gelatinous precipitate and difficult to filter.

Forms in which phosphoric acid is used

Phosphoric acid is available commercially for use in the sugar factory in powder or paste form: Packard superphosphate, asbuslite, sumaphos, etc. Failing these, ordinary superphosphate or dicalcium phosphate may be used.

These materials contain 30 – 60% P_2O_5 ; sumaphos, 42%; dicalcium phosphate, 38 – 42%.

They are diluted to form a solution of 12° or 15° Baumé.

Procedures

The phosphoric acid will be added before liming. Then follows the clarification process which has been adopted. To obtain the same pH after liming, it will evidently be necessary

to add more lime, in order to neutralise the added acid; this will have reduced the original pH of the juice, generally about 5.5, to 4.5 or 4.6 for an average dose. However, certain persons consider that, in practice, the use of phosphoric acid leads to a saving in lime.

Phosphoric-acid clarification may without difficulty be combined with sulphitation. For example, the following is a procedure employed on refractory juices in India¹:

Addition of phosphoric acid to the mixed juice

Liming to pH ≥ 8

Heating to 70 or 75°C (160 or 165°F)

Sulphitation to pH of 6.8 – 7.0.

Advantages

The use of phosphoric acid improves the clarification of refractory juices.

It often permits of eliminating the sulphitation. In other words, it is possible, to a certain extent, to replace sulphurous by phosphoric acid.

Considered from this angle, phosphoric-acid clarification is much less expensive than sulphitation, does not require special equipment, avoids corrosion of plates and pipes, and reduces scaling at heaters and multiple effects.

However, phosphoric acid should rather be regarded, not as a replacement, but as an adjunct to the clarification, which should be considered when exceptional difficulties are encountered in subsidence or filtration.

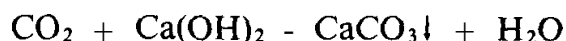
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27. Carbonatation

Action of carbon dioxide on juice

Carbonatation was originated in the beet sugar factory by Perier and Possoz, in 1859. They reported that, if a precipitate of calcium carbonate be formed in the juice, it entraps colouring matter and gums, if the reaction remains alkaline, and thus furnishes a marked additional degree of clarification. The precipitate so formed is granular and filters easily:



The same reaction has been utilised in the cane sugar factory. However, though carbonatation is general practice with beet, it remains the exception with cane.

Procedures

There are various ways of operating, the most important of which are:

Single Carbonatation

Double Carbonatation

De Haan Carbonatation

Middle-juice Carbonatation

Double carbonatation, which is more effective than single, comprises in general:

Heavy liming

Carbonatation leaving a very high alkalinity

Filtration

Second carbonatation until the alkalinity falls to a very low value

Boiling

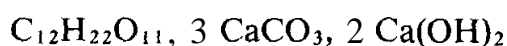
Second filtration.

Carbonatation should never be carried out at a temperature above 55°C (131°F), the maximum permissible if destruction of reducing sugars is to be avoided. Reducing sugars, apart from their value in the distillery, serve to prevent the loss of a corresponding quantity of sucrose in the molasses.

On the other hand, below 45°C (113°F), the reaction is very slow and incomplete. One must be careful, therefore, to approach 55°C (131°F), and to conserve a margin of safety in both directions, a temperature of 50°C (122°F) is generally maintained.

This temperature is obtained either by passing the juice through a heater, or by means of steam pipes in the bottom of the carbonating tanks.

In the course of the first carbonatation, there is formed by the lime, the CO₂ and the juice, a sucro-carbonate of lime:



which is gelatinous and insoluble, and would hold a large quantity of sugar in combination. Fortunately, it is decomposed after a certain time.

This viscous compound also promotes the formation of abundant froth.

The juice will therefore be held in the carbonating tank for a time of 25 min for the first carbonatation, say 30 min including filling and emptying, and a total of 15 min for the second. The capacity of the 2nd carbonatation tanks should therefore be half that of the first,

Double carbonatation. The considerations which have given rise to double carbonatation are the following:

(1) If the action of CO_2 were pushed to the limit, the insoluble lime salts formed would be decomposed, and would thus be re-dissolved in the juice. Hence the carbonatation is stopped at the degree of alkalinity desired to prevent this decomposition. After filtration, which removes these salts, carbonatation may be continued, or rather a second carbonatation commenced, which will then be carried practically to neutrality.

(2) Large pieces of poorly mixed lime would continue to dissolve slowly after a first carbonatation, and would gradually render the reaction alkaline. When the juice passes to the heaters, the reducing sugars would suffer decomposition (since such decomposition occurs at high pH and high temperatures), and the juices would again develop colour. The first filtration and the second carbonatation eliminate these pieces of lime and so stabilise the pH.

In beet sugar work, sometimes even 3 successive carbonatations are carried out.

Sulphitation. With the same object of removing all alkalinity, sulphitation may be combined with carbonatation by sulphiting after the first filtration, to neutrality or slight acidity.

This is the process which would give the best white sugar.

It renders the second carbonatation superfluous,

In Formosa¹, double carbonatation is operated with double sulphitation. The juice is heated to 55°C (131°F), carbonated batchwise for 10-15 min in a vertical tower, while the milk of lime at 15° Be is introduced at the top of the tower, in the ratio of about 2% on weight of juice; the juice thus reaches a pH of 10.5, which gives a copious precipitate of lime salts entraining colloids. The further steps are as follows: 1st filtration (using 0.7-0.8 $\text{m}^2/\text{t.c.h.}$ (75-85 sq.ft./t.c.h.)); heating to 85°C (185°F); second (continuous) carbonatation, to remove the excess lime salts, to a pH of 8.5; second filtration (2.8-3.2 mVt.c.h. (30-35 sq.ft./t.c.h.) of filter-press area); 1st sulphitation to neutrality; evaporation to 60° brix; 2nd sulphitation and 3rd filtration. This gives a clear and colourless syrup which goes to the vacuum pans. The process gives a sugar of 99.6-99.9% pol, 0.04% moisture, 0.04% reducing sugars and 0.06% ash.

De Haan carbonatation. De Haan has conceived the idea of adding the lime and CO_2 simultaneously, so that the alkalinity remains relatively low.

This procedure has the great advantages of:

- (1) Suppressing the frothing
- (2) Avoiding the formation of the gelatinous sucro-carbonate.

The juice is heated to 55°C (131°F) in the heaters and sent to the carbonating tanks. Gas is introduced at the same time as the milk of lime is run in, so maintaining the pH at a cons-

tant value, which is checked with the aid of a special phenolphthalein paper treated with oxalic acid, and which changes colour at a given alkalinity. When the desired quantity of lime has been introduced, the carbonatation is continued to neutrality to phenolphthalein, and is finished as for single carbonatation.

This process permits of a substantial economy in space, while increasing the capacity of the heaters, and giving a marked economy in lime consumption, with a slight rise in purity, and gives a better quality sugar. It is, according to Douwes Dekker², the process which assures the best elimination of non-sugars.

Middle-juice carbonatation. The term "middle juice" is used to designate juice which has been partially concentrated, and withdrawn from the evaporators at a point where it has reached 35 – 42° brix.

Middle-juice carbonatation commenced about 1938 in Java, but is today confined to Formosa, where it is gradually extending (Chou³). It consists of the following stages:

- (1) Heat the juice to 101.5°C (215°F)
- (2) Add lime to pH 7.0 – 7.2
- (3) Send the juice thus limed to the evaporators, where it is passed through vessels 1, 2 and 4;

It leaves at about 35 – 42° brix and 55°C; the choice of the latter vessel is made because this temperature is the optimal for carbonatation

- (4) Lime and carbonate to pH of 9.8 - 10.3
- (5) Filter
- (6) Second carbonatation to pH of 8.2 or 8.5
- (7) Heat to 77 – 80°C (170 – 175°F)
- (8) Second filtration
- (9) Sulphitation to pH of 7.0 or 7.3
- (10) Return to the evaporators, to 3rd vessel
- (11) Second sulphitation to pH of 6.0 – 6.2.

This process would offer the advantage of reducing substantially the consumption of lime, by about 40%; of giving a better purification and a greater elimination of non-sugars; of giving a better-quality sugar; of considerably decreasing scale formation in the multiple effects, with consequent economy in soda ash for cleaning of the vessels.

On the other hand, it is very sensitive to operate and necessitates close control.

Quantities of lime necessary

Table 27.1, after Prinsen Geerligs⁴, gives the quantities, in kg/tonne cane, of lime required for the principal clarification procedures, also for the corresponding quantities of limestone and other primary materials.

We have completed this table by adding the last line.

Honig⁵ estimates 25 – 30 kg (55 – 65 lb.) of limestone per t.c. (probably for de Haan carbonatation); 15 kg/t.c. (33 lb./t.c.) in the case of middle-juice carbonatation.

Rault⁶ states that the more lime used, the better is the elimination of gums.

TABLE 27.1

MATERIALS USED FOR THE VARIOUS CLARIFICATION METHODS (kg/t.c., X 2.2 for lb./t.c.)

	<i>CaO</i>	<i>Limestone</i>	<i>Coke</i>	<i>Sulphur</i>	<i>P₂O₅</i>
Sulphitation	0.8-1.5	—	—	0.5	
Sulphitation and phosphoric acid	1	—	—	0.1	0.1
Single carbonatation	18	35	4	0.2	
Continuous double carbonatation	18	35	4	0.2	
Ordinary double carbonatation	20-23	40-45	4.4	0.2	
De Haan carbonatation	12	23	2.5	—	
Middle-juice carbonatation	9	20	2	—	

Lime kiln

The CO₂ necessary for the carbonatation is produced at the same time as the lime, in a lime kiln adjacent to the factory.

As lime is a solid material and comparatively easy to store, defecation factories can buy it from outside suppliers. Carbon dioxide being a gas, it has so far not been possible to procure it other than by preparing it on the site, using as raw material limestone or coral, which consists of more or less pure calcium carbonate; quicklime is obtained at the same time:



Fuel. In order to promote this dissociation, it is necessary to supply in practice about 700 kcal/kg (1,260 B.T.U./lb.) of limestone. This heat is obtained by burning coke or other fuel in the kiln. To ensure the proper distribution of the heat in the mass of limestone, layers of coke and of limestone are fed into the kiln alternately.

Generally, 8-10 kg of coke are used per 100 kg of limestone. In exceptional cases, this is reduced to 7 kg.

The coke contains 75-90% carbon, averaging 85%. Its calorific value is about 7,000 kcal/kg (12,600 B.T.U./lb.).

In Java, a mixture of coke and charcoal is used as fuel, in the proportion of 1 of coke to 10 of charcoal. On starting the kiln, the first two layers are prepared using 1 of coke to 5 of charcoal.

Tromp⁷ reports that bagasse has been used successfully as fuel in lime kilns, with excellent results.

Limestone. The limestone is obtained either from limestone quarries or from coral deposits.

Limestone generally contains 85-98% of CaCO₃, averaging 90%. Coral deposits, such as those normally used in lime kilns in Mauritius and Reunion, contain 87-94% of CaCO₃. Again, 90% may be taken as an average figure.

Coral limestone, if recently recovered from the sea, should be washed or left in the rain, to remove the salt which it contains. It is recommended that coral lime with a high proportion of magnesia should be avoided. This recommendation is rather inconsistent with the procedures described in Chapter 28.

It is desirable to use stones of approximately the same size, otherwise the large pieces will be insufficiently burnt ("underburnt"), and the smaller ones overburnt. The pieces of coke should be about half the size of the pieces of limestone (0.5 – 0.6).

Operation of the kiln. The temperature in the kiln should be maintained between certain limits: if it exceeds $1,350^{\circ}\text{C}$ ($2,460^{\circ}\text{F}$) it gives what is known as "dead-burnt" lime, *i.e.* an inert and unsuitable lime.

The dissociation of limestone begins at about 600°C ($1,110^{\circ}\text{F}$). The temperature of a good lime kiln is maintained between $1,100$ and $1,300^{\circ}\text{C}$ ($2,000$ and $2,400^{\circ}\text{F}$).

Kiln. We give in Fig. 27.1 a view of a Khern lime kiln, which is the type most used. Charging (by skip) is effected at the top, also removal of the CO_2 ; ignition of the fuel, and removal of the lime, from the bottom.

The gas leaves at a temperature of $70 - 120^{\circ}\text{C}$ ($160 - 250^{\circ}\text{F}$), and passes through a washer-cleaner and dust remover which reduces it to about 60°C (140°F). At this temperature, 1 kg of CO_2 occupies 617 litres under atmospheric pressure (10 cu.ft./lb.).

Capacity of lime kilns. This capacity is very variable. Tromp⁸ estimates a production of

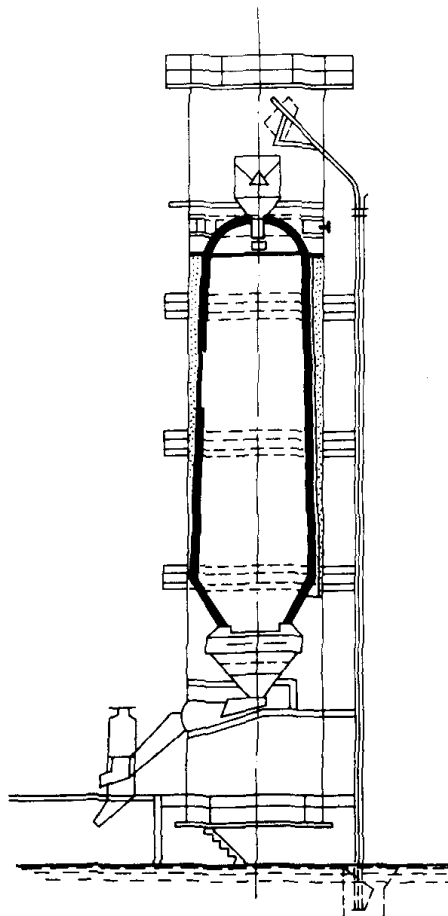


Fig. 27.1. Lime kiln (sectional view).

370–1,235 kg CaO/24 h/m³ (23-77 lb. CaO/24 h/cu.ft.) of volume. A figure of 480 kg/m³/24 h (30 lb./cu.ft./24 h) may be taken as an average.

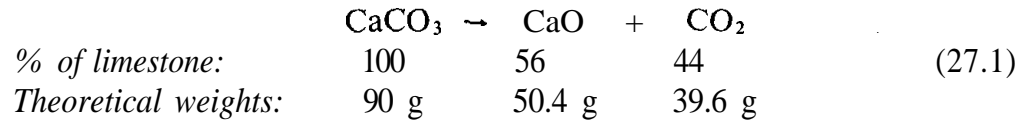
Composition of gas. Let us assume the following conditions:

Limestone at 90% CaCO₃

Coke at 85% carbon

Proportion of coke to stone = 8%.

The dissociation gives:

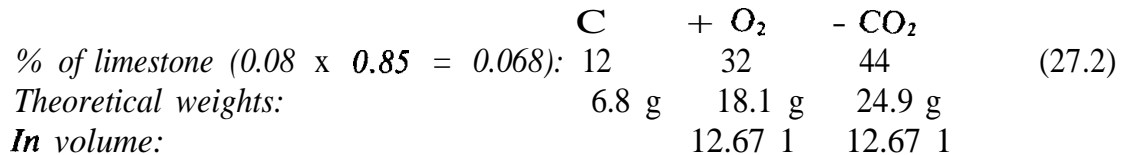


This decomposition is endothermic and absorbs 42.5 kcal per mol, or say 425 kcal/kg CaCO₃ (765 B.T.U./lb.).

This 39.6 g CO₂ will occupy a volume of:

$$22.4 \times \frac{39.6}{44} = 20.16$$

The combustion of the 8% of coke used to furnish the necessary heat for the dissociation will give:



since:

$$22.4 \times \frac{18.1}{32} = 22.4 \times \frac{12.67}{44} = 12.67 \text{ l}$$

But the 12.67 l of oxygen bring with them:

$$12.67 \times \frac{79.16}{20.84} = 48.13 \text{ l of nitrogen}$$

The maximum proportion of CO₂ therefore is:

$$\frac{(20.16 + 12.67) 100}{20.16 + 12.67 + 48.13} = \frac{32.83 \times 100}{80.96} = 40\%$$

In practice, it is not possible to operate with the exact theoretical quantity of air, and it

is necessary to tolerate a certain excess of air. It follows that in practice the proportion of CO₂ by volume in the kiln gases varies from 25 to 33%, averaging 30%.

The uncombined oxygen in the gas varies from 0.5 to 8%.

An exceptional performance is that of Tirmont Refinery⁹, in Belgium, giving 38% CO₂ in a very modern and improved kiln, and reducing the coke to 7%.

We may comment that, for the conditions adopted above, we obtain:

$$39.6 + 24.9 = 64.5 \text{ g}$$

of CO₂ for 50.4 g of CaO, or about 128 g CO₂/100 g CaO. Even allowing for the low efficiency of the carbonating tanks, this quantity is amply sufficient to supply the carbonation.

CO₂ pump

The CO₂ gas leaving the washer is at a temperature of about 60°C (140°F). Its pressure, at the suction of the pump, varies from -5 to -12 cm (-2 to -5 in.) of mercury, most often -7.5 to -10 cm (-3 to ~4 in.).

The delivery pressure varies from 0.3 to 0.7 kg/cm² (4 - (0 p.s.i.). As an average, we may take 0.4 kg/cm² (6 p.s.i.).

Pumps for CO₂ are fully analogous to air pumps, which we shall discuss later (Chapter 40). However, they do not require water cooling, thanks to the small difference between suction and delivery pressures.

Specification of a CO₂ pump. To fix our ideas, let us take a concrete example. Given:

Crushing rate:	100 t.c.h.
Consumption of CaO:	18 kg/t.c. (40 lb./t.c.)
Proportion of CaCO ₃ in limestone:	90%

Limestone. Basing our calculations on the CaO required, we see from the reaction (27.1) that we shall require per tonne cane:

$$18 \times \frac{100}{56} \times \frac{1}{\dots} \approx 35.7 \text{ kg (79 lb.) limestone}$$

To allow for losses and unburnt, we shall allow for

$$35.7 \times 1.05 = 37.5 \text{ kg/t.c. (83 lb./t.c.)}$$

Density of CO₂. Let us assume that the withdrawal of gas from the lime kiln is carried out at an absolute pressure of 68 cm (27 in.) of mercury (8 cm vacuum). The density of CO₂ relative to air is 1.529. At 60°C (140°F) and 68 cm absolute, the density of the gas will be:

$$1.293 \times 1.529 \times \frac{1}{76} \times \frac{1}{1 + (0.00367 \times 60)} \approx 1.450 \text{ kg/m}^3 \text{ (0.091 lb./cu.ft.)}$$

Volume of CO₂. Our calculation for the composition of gas has shown that 100 kg of limestone of 90% CaCO₃, with coke of 85% C, would produce about 64.5 kg of CO₂; or, per t.c.:

$$37.5 \times \frac{64.5}{100} = 24.2 \text{ kg (53.5 lb.) CO}_2$$

occupying a volume of

$$\frac{24.2}{1.450} = 16.68 \text{ m}^3 \text{ (588 cu.ft.)}$$

Volume of gas. We have seen that the gas contains on an average only 25 - 33% CO₂. Basing our figures on 30%, we shall have a gas volume of:

$$\frac{16.68}{0.30} = 55.6 \text{ m}^3/\text{t.c. (1,960 cu.ft./t.c.)}$$

Total volume. For 100 t.c.h., we shall have:

$$V_1 = 5,560 \text{ mVh (196,000 cu.ft./h)}$$

Increasing this by 10% for safety:

$$V_2 = 5,560 \times 1.10 = 6,116 \text{ mVh (215,600 cu.ft./h)}$$

The volumetric efficiency of CO₂ pumps varies from 0.60 to 0.75. Taking 0.70, we shall require:

$$V_3 = \frac{6,116}{0.7} = 8,737 \text{ mVh (308,000 cu.ft./h)}$$

If the pump is double-acting, we should have:

$$60 \times 2 \times L \times S \times n = V_3 \quad (27.3)$$

L = stroke of the pump, in m (ft.)

S = effective area of the plunger, in m² (sq.ft.) = $(\pi D^2)/4$ or $[\pi (D^2 - d^2)]/4$

D = diameter of the pump, in m (ft.)

d = diameter of the piston-rod and the tail-rod, in m (ft.)

n = speed of rotation of the pump, in r.p.m.

From (27.3), we calculate the product, $nL (D^2 - d^2)$, and hence the specifications for the pump, by choosing:

(1) A convenient proportion L/D

(2) A reasonable speed.

This speed should be maintained constant, otherwise the performance of the kiln will become erratic and difficult to control.

Power for the pump. On account of the small difference in pressure between suction and delivery pressures, the CO_2 piston, in steam pumps, is appreciably greater than the steam piston. In general, its diameter is double.

Power required is estimated at $1.1 - 1.5 \text{ kW/m}^3/\text{h}$ ($0.04 - 0.06 \text{ h.p./cu.ft./h}$).

Carbonating tanks

The carbonatation is generally carried out batchwise in a set of carbonating tanks or "boilers". These are vertical tanks, of rectangular or circular cross-section. They hold juice to a depth of 2–3 m (7–10 ft.), and the tanks extend to a height of 4–7 m (13–23 ft.), the difference serving to allow for the copious frothing of the first carbonatation.

The tanks for the second carbonatation are identical except for the height, which is less, as the frothing is practically negligible.

Rectangular tanks take up less space than cylindrical ones, but should be provided with stay-rods to avoid bulging of the walls under the pressure of the juice. Moreover, the juice near the corners of the tanks is less effectively carbonated.

The carbonating tanks are sometimes provided with a cover which leads into a flue serving to evacuate the gas after it has passed through the juice in the tank. Often, they are left open at the top. They have a coil for heating with steam, a pipe for admission of the CO_2 which terminates in a star-shaped distributor, with serrated edges (Fig. 27.2), a juice inlet pipe, an overflow pipe, and, generally, a device for breaking the foam, made from a perforated pipe blowing small jets of steam in a horizontal plane above the surface of the juice.

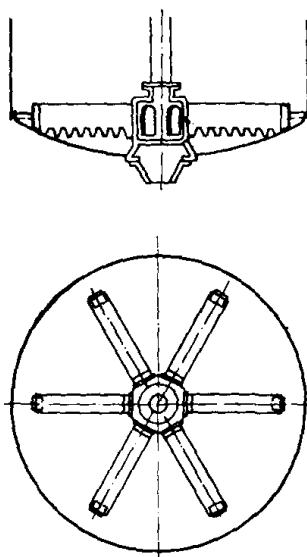


Fig. 27.2. Star distributor.

Capacity of carbonating tanks. A carbonatation installation will be planned on a basis of 3–6 vessels, preferably 4, for each of the two carbonatations. It will be assumed that one is emptying, one filling, allowing 5 min for each of these operations; and the capacity of one

vessel will be deduced from the height of juice chosen (2–3 m) and from the necessity of allowing the juice to remain for at least 20 min in each vessel during the saturation with CO_2 , or about half an hour total, for single carbonatation or first carbonatation, and 15 min for the second carbonatation.

Honig¹⁰ specifies:

(a) A volume of heaters for 1st carbonatation of 1.4–1.7 m³/t.c.h. (50–60 cu.ft./t.c.h.), for a juice-holding period of 8–12 min per batch, with a juice depth of 2–2.5 m (7–8 ft.); milk of lime of 15–20° Be.

(b) A heater volume for second carbonatation of 0.11–0.15 mVt.c.h. (4.5–6 cu.ft./t.c.h.), corresponding to precipitation of 200–400 mg of CaO per litre of juice filtered and heated to 70°C (160°F).

Continuous carbonatation

Plant for continuous carbonatation has also been used, which reduces the labour requirements and simplifies operation. These are similar to the batch carbonators, but are designed to facilitate and prolong as much as possible the contact between CO_2 and juice.

Control of the proper degree of saturation is more delicate, and is effected by regulation of the juice flow.

Continuous carbonatation is very commonly met with in the second carbonatation, less often at the first.

When the second carbonatation is carried out continuously, the 2nd-carbonatation tanks are generally given a juice capacity of 150 l/t.c.h. (33 gal./t.c.h.)¹¹, although some factories are content with 60 l/t.c.h. (13 gal./t.c.h.).

The juice from the 2nd carbonatation should be filtered at 70°C (158°F). With filter-presses, 3 m² (33 sq.ft.) of filtering surface should be provided per t.c.h.; with Sweetland filters, 1.7 mVt.c.h. (18 sq.ft./t.c.h.) should be adequate.

The heaters for bringing the juice to 70°C will preferably be heated with vapour from the 2nd or 3rd effect. The necessary heating surface may be reckoned as 16–20 mVt.c.h. (175–220 sq.ft./t.c.h.) using vapour from the 2nd effect, or 20–22 m² (220–260 sq.ft.) with vapour from the 3rd effect.

Efficiency of utilisation of CO_2

The completeness of utilisation of the CO_2 depends mainly on the surface of contact between juice and gas, and on the duration of this contact.

Consequently, there are two principal factors influencing this utilisation:

(a) The system of gas distribution in the mass of juice. We have referred to the distributor in star form with serrated branches. The smaller and more numerous the bubbles formed, the greater will be the area of contact.

(b) The length of the gas path from the distributor pipe to the surface of the juice, *i.e.* for ordinary carbonating tanks, the depth of juice.

The efficiency is never perfect. The efficiency of utilisation of the CO_2 is of the order of:

30–75% according to Quillard¹²

40–60% according to Tromp¹³

It sometimes reaches 80–90% in the most favourable circumstances¹⁴. Even in this case, the gases discharged by the flue still contain about 12–15% of CO₂.

There is indeed no need to seek perfection in utilisation of CO₂, since the quantities furnished by the lime kiln and corresponding to the normal quantities of lime used are always amply sufficient for the needs of the carbonatation.

Favourable conditions for use of carbonatation

Carbonatation, and especially double carbonatation and the de Haan process, is the standard method for white sugar manufacture.

It is, however, expensive, both on account of the extra equipment required, and on account of the extra raw materials which it demands.

Its use will logically be restricted, therefore, to cases where both the two following conditions apply:

(1) The factory plans the systematic and continuous manufacture of white sugar. If it were desired to make white sugar only in an incidental or accessory fashion, sulphitation would be indicated, since it requires only an insignificant amount of extra equipment; putting it into operation or shutting it down does not involve a great upset to the operation of the factory.

(2) The factory can procure readily and cheaply the necessary limestone or coral.

A carbonatation factory has an efficiency generally superior to that of factories working on simple defecation or on sulphitation. This can generally be reckoned at 1–2% higher yield of sugar. Honig¹⁰ estimates 1.5–2.5%.

Steam consumption

It is estimated by Podder¹⁵ in India that a carbonatation factory consumes 10–12% more steam (% on cane) than a defecation or sulphitation factory. Mean figures for that country would be:

Sulphitation	55% steam on cane
Carbonatation	66% steam on cane

In Taiwan¹⁶ it is also agreed that carbonatation involves a 20% increase in consumption relative to that for defecation.

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28. *Use of Magnesia in Clarification*

Action of magnesia on the juice

The action of magnesia is similar to that of lime, but it has often been stated that magnesia was undesirable and that it was important to avoid the use in defecation of lime containing more than a certain proportion of magnesia.

Magox

“Magox” is a very finely powdered magnesium oxide which may generally be partially, or in exceptional cases completely, substituted for lime in the defecation process. It is never employed only to replace the lime, since its use is more expensive, but rather with the object of avoiding scale formation on the multiple effects.

This result is due to the fact that certain magnesium salts, particularly the sulphate, are soluble, whereas the same salts of lime are insoluble and the magnesia tends to be substituted for the lime in the compounds formed.

Due to the lower molecular weight, 1 kg of magnesium oxide (**MgO**) replaces 1.390 kg of quicklime (**CaO**), or 1.837 kg of hydrated lime (**Ca(OH)₂**). The best results are obtained generally when 50 - 80% of the lime (expressed in terms of pulverised hydrated lime) are replaced by the corresponding quantity of MgO. If we assume that the factory conducted its clarification with, for example, 0.8 kg (1.8 lb.) of lime per t.c., these quantities correspond to:

0.4 kg (0.9 lb.) of lime + 0.2 kg (0.5 lb.) of MgO, or a 65 – 35% mixture, or:

0.15 kg (0.35 lb.) of lime + 0.35 kg (0.78 lb.) of MgO, or a 30 – 70% mixture
both of these being equivalent to 0.8 kg (1.8 lb.) of lime used previously.

It is convenient for a start to commence, for example, with the latter mixture, then each week to reduce it and replace 10 g (0.025 lb.) of magnesia by using an additional 18 g (0.045 lb.) of lime, so long as the tubes of the multiple effects remain **clean**, so as to arrive at the minimal proportion of Magox which will give the desired result.

The tubes in fact remain clean and brilliant. The expense of Magox is thus involved, as it costs 2 – 3 times as much as the corresponding quantity of lime; however, the savings, in addition to the lime replaced, include the labour of descaling (generally at overtime rate at weekends), the cost of descaling tools (*cf.* p. 548), which wear rapidly, also the cost of flexible drives from motor to the descaling tools; the cost of replacement motors, costs of soda and hydrochloric acid for scale removal, as well as wear of the tubes due to cleaning. Furthermore, there is the advantage of a multiple effect the efficiency of which remains constant throughout the week. Some users of this method quote in addition an improved purity rise in clarification, a clearer juice, and even a better exhaustion of molasses. These latter effects are scarcely significant and are somewhat doubtful, but it must be remembered that the use of Magox permits the solution of certain problems due to tenacious scale, and it is easy to

work out the balance of advantages and economies as opposed to the expense due to the cost of the material.

Care must be taken on account of the low rate of reaction of magnesia, especially if the volume of the heaters allows only a short time of contact. The pH must be watched; if it continues to rise after the heating tanks, for example, in the heated juice, it is because the Magox has not had time to react. The pH should be equal in the various compartments of the subsider. If it increases from the top compartment to the bottom one, it is due to an excess of the clarifying agent. In the same way, if the pH of the mud is higher than that of the clarified juice, it indicates excess of Magox. In general, it is advisable to allow 20 minutes contact before reheating¹.

Any pulverised burnt magnesia may be used, provided that the powder is sufficiently fine, 99% passing through a 325-mesh Tyler screen.

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29. *Subsidiation*

When the juice has undergone the desired treatment - simple defecation, sulphitation, or carbonatation - it must be allowed to settle in order to separate the clear juice from the precipitate formed in the body of the juice. This settling is effected in continuous decanters or "clarifiers".

Dilution

The main factor controlling settling is the difference in density between the material to be settled and the liquid in which it is mixed. The area and the shape of the particles thus play an important part, as well as the viscosity of the liquid.

It follows that the time necessary for settling depends mainly on the difference in density between the juice and the particles of the precipitate.

At Clewiston, in Florida, tests have been made on the time necessary for settling as a function of the dilution of the juice¹.

- (a) Mixed juice: 78 min
- (b) Mixed juice + 10% of water: 38 min
- (c) Mixed juice + 20% of water: 28 min

In addition to the effect of difference in density, the viscosity of the juice falls with decreasing brix, thus assisting settling.

Salinas², in Cuba, takes as basis a juice of 16° brix, and estimates that it is necessary to allow a settling area 2% greater for each degree brix above 16, in other words, a settling time approximately 2% longer.

Although the Clewiston results appear much more striking than what is observed in normal operation, it is important to remember that increasing imbibition improves the subsidiation: the gain realised in the time of settling much more than compensates for the increased volume of juice.

Speed of settling

The speed of settling of an average juice is approximately 75 mm (3 in.) per minute, until the concentration of the muds near the bottom has reached a maximum. Then the speed of settling diminishes rapidly and approaches zero.

Surface area and volume of subsidiers

The speed of settling of the particles comprising the precipitate is constant for a given juice. It follows that the settling time is proportional to the depth of juice. In other words, the capacity of a subsidier is independent of its depth: a subsidier of twice the height will contain

twice as much juice but will take twice as long to settle. Thus only the surface area is important.

Subsider capacities should therefore be expressed in area rather than in volume. They are often found, however, still indicated in terms of volume, but this is a designation liable to introduce errors. However, in the case of rapid (tray-less) crystallisers, the volume does play an important role.

CLARIFIERS OR SUBSIDERS

“Clarifier” or “subsider” is the name given to a continuous settler.

A continuous subsider is a vessel into which the juice to be settled is fed uniformly and continuously, and which is large enough to reduce the velocity of flow and of circulation of the juice to such a low value that it does not prevent settling from taking place. The clear juice obtained is similarly withdrawn from the upper part of the subsider in an equally uniform and continuous manner, as also are the muds from the lower portion.

Description of a clarifier

Clarifiers are generally divided into several compartments, so as to increase the area for settling.

Since clarifiers are similar in principle, and vary in details only, we shall describe the “Rapidorr”, which is perhaps the most widely used.

It is provided with a central hollow shaft, rotating very slowly (12 rev./hour), which carries scrapers of sheet metal which slowly brush the bottom of each compartment.

The juice to be clarified enters tangentially at the top, into a compartment half the diameter of the main clarifier; this is termed the feed compartment or flocculation chamber. Here some scum rises to the surface; this is eliminated by a special scraper which pushes it into a small lateral discharge canal leading to the mud outlet box.

The Rapidorr 444 consists of 4 superimposed compartments, each forming a complete clarifier independent of the others, and fed separately by a rotating central tube by means of openings situated in the upper part of the compartment. The entering juice encounters baffle plates designed to ensure good distribution of the juice. It flows radially towards the outer wall, the velocity decreasing in inverse ratio to the radius. Settling of the mud proceeds simultaneously, and the muds deposited on the bottom plate are moved by scrapers mounted on arms connected to the central tube and pushed towards the centre; there they settle into a mud tray from which they are extracted by diaphragm pumps at controllable rates; the mud from the various trays is pumped to a mud tank, from which it goes to the filtration plant.

The clear supernatant juice is withdrawn from each compartment by a circumferential internal pipe with several openings which withdraw the juice close to the roof of the compartment. The juice then passes through an overflow box, by vertical pipes fitted with sliding sleeves which permit regulation of the rate and overflow level.

Vertical tubes through the roof of the equipment allow escape of gas from each compartment to atmosphere.

The Rapidorr 444 does not require any addition of flocculant, but such addition does im-

prove the subsidation and reduces the necessary settling time. The subsider has a capacity slightly greater than that of previous models.

The clarifier is enclosed, except for a door giving access to the flocculation chamber. It is completely lagged; thanks to this protection, and especially to its dimensions, which give it a very small ratio of external surface to volume, the juice leaves at about 99°C (210°F). After being shut down over Sunday, the juice is still at 90 - 95°C (195 - 200°F) when starting up on Monday morning. This represents a cooling rate of 0.16 - 0.22°C/h (0.3 - 0.4°F/h) in general, for a clarifier of 100 - 200 m³ (3,500 - 7,000 cu.ft.).

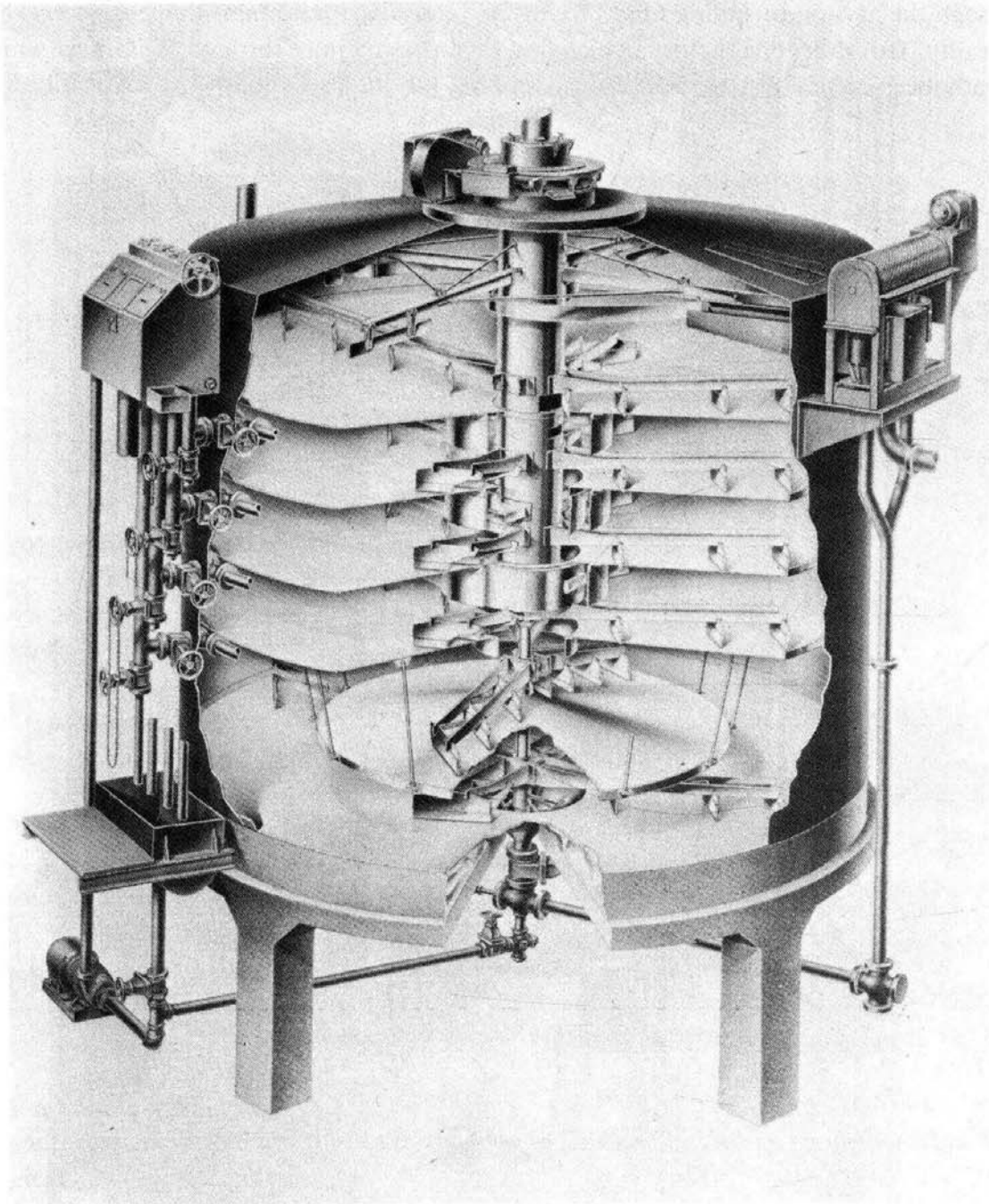


Fig. 29.1. Dorr "Multifeed" clarifier.

The Rapidorr was preceded by the Dorr "Multifeed" (Fig. 29.1), which consisted of three to five compartments, only the bottom one being arranged for collecting muds. It was succeeded after the war by the A.T.V. model, designed for the beet sugar industry but also utilized with cane; then by the A3T model, which consisted of two superimposed subsidors each of two compartments; and was finally replaced by the Rapidorr 444 (Fig. 29.3).

A clarifier very similar to the Dorr is the Graver (John Seip) (Fig. 29.2). The muds in this design move down, no longer by the centre, but around the periphery; the settling area of the compartments is therefore slightly less than the cross-section of the clarifier. The clear juice is withdrawn from the top of a bell in such a way that the juice must theoretically pass through the curtain of falling mud. We do not consider that things happen exactly this way in reality. However, one factory, which had tried passing juice through the Graver which had already been settled rapidly in ordinary settlers, has indicated that the clarification became

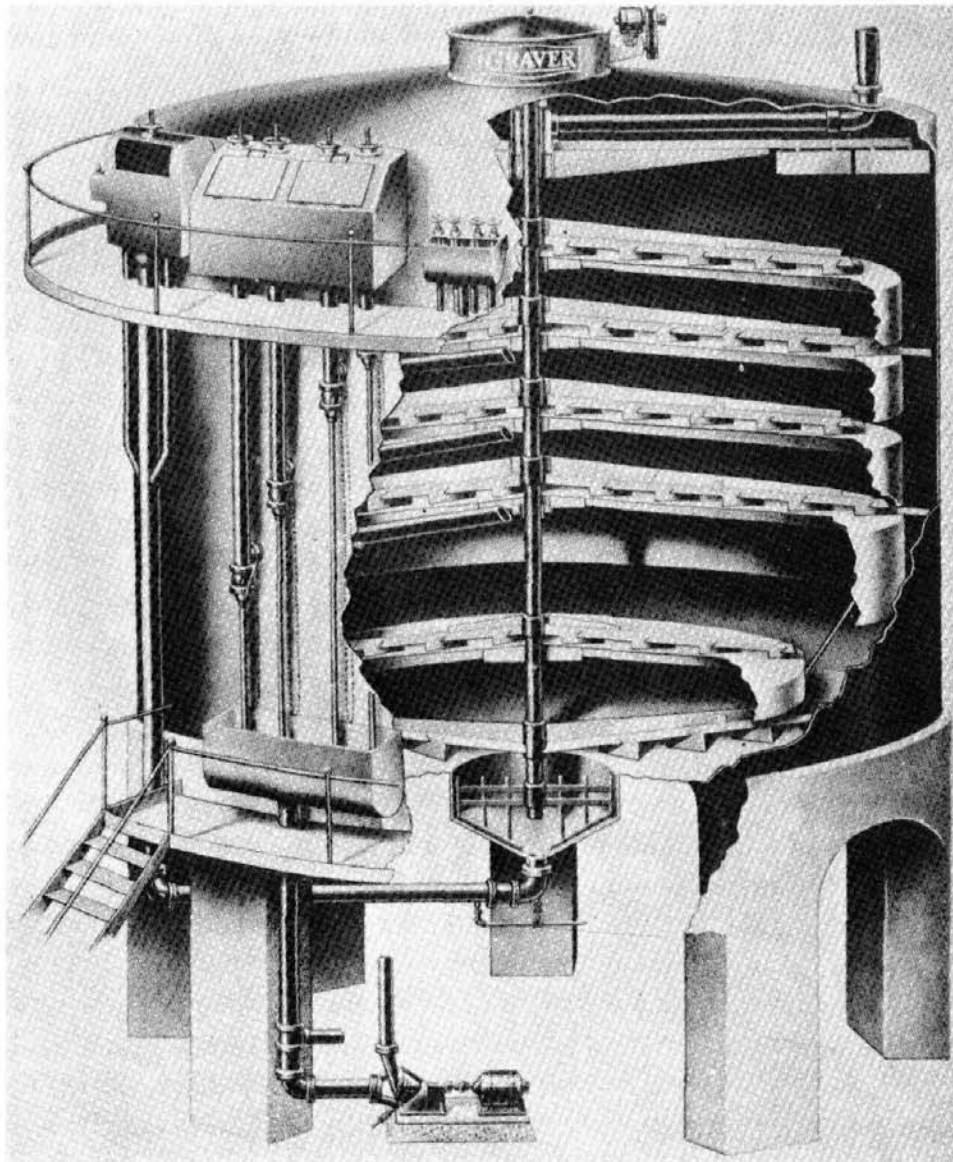


Fig. 29.2. Graver clarifier.

better as the juice entering carried more mud; which tended to confirm the theory of the inventor: "upward sludge filtration", that is, filtration of the rising juice through the descending curtain of settling mud.

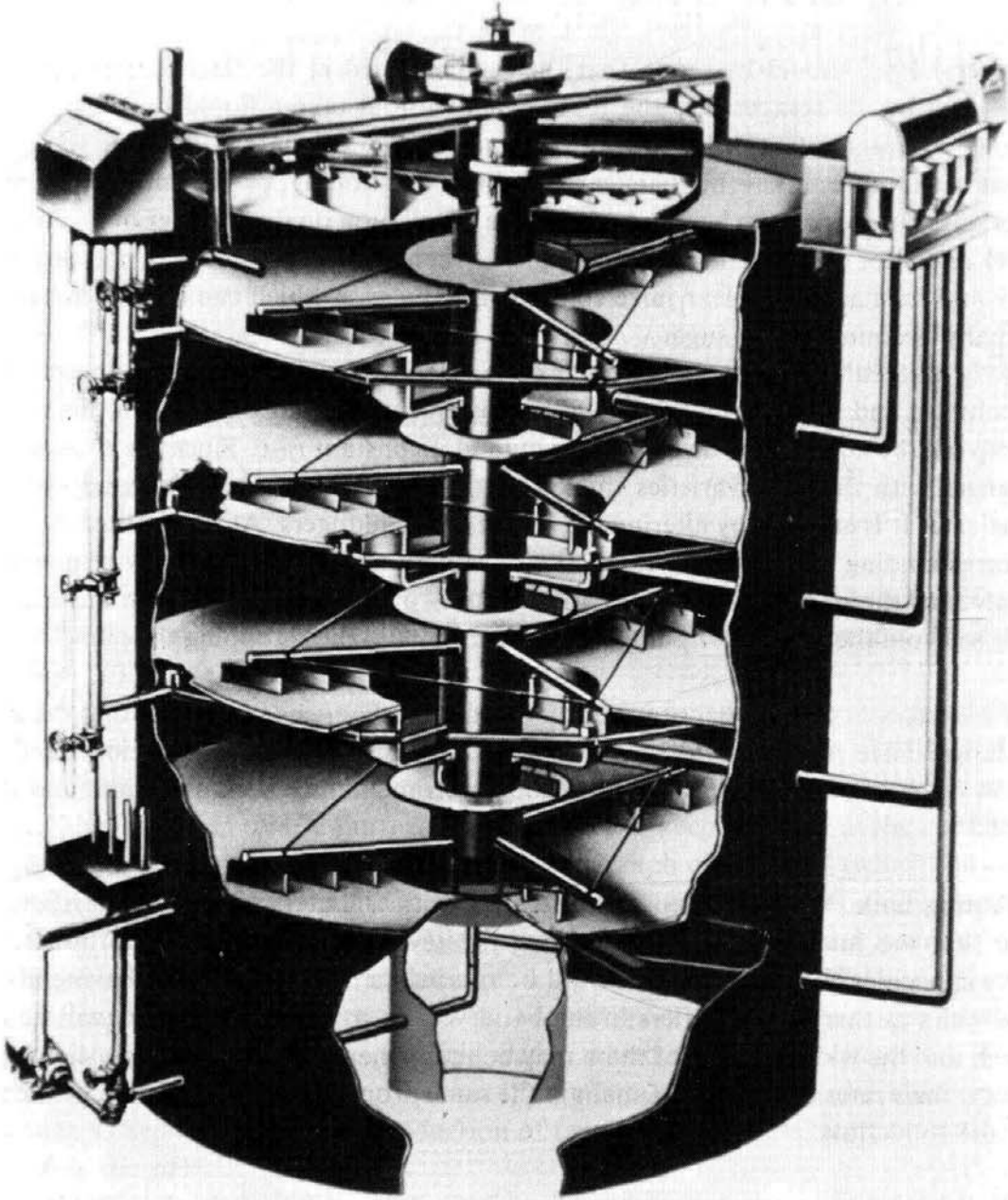


Fig. 29.3. Rapidorr 444 clarifier (Dorr - Oliver)

Operation of a clarifier

When the factory is started up, the clarifier becomes filled with juice up to the overflow level of the clear juice outlet pipes.

This level may be regulated by a handwheel, each turn of which raises all the sliding tubes simultaneously by a small amount of the order of a millimetre. In addition, each tube may

be adjusted by means of an individual handwheel which permits the output of the corresponding compartment to be increased or decreased.

Correct adjustment should give practically equal juice flow from the several compartments. Adjustments should be made by very small amounts of one turn of the handwheel at a time. Correct regulation is recognised by the fact that further adjustments are not necessary.

The general level should be set so that the scum formed in the **flocculation** chamber is readily pushed by the scraper into the foam canal without taking liquid juice with it. This necessitates constant attention by the operator in charge of the clarification station; the author has often noticed that the foam canal was not functioning, or was functioning at very long intervals. For this reason he recommends the installation on the **clarifier** of an automatic juice-level regulator (Masoneilan, for example). When removal of scum is not carried out smoothly and regularly, the clear juice contains fine bagasse which cannot be removed and which finally is found in the sugar.

In the “**Multifeed**” and most of the older designs, the compartments are connected by the central column, and there is only one mud compartment, the bottom one. In this case, the mud pump will be adjusted so as to remove mud at a constant rate. Since the proportion of mud changes with the cane varieties treated or their state of maturity, this adjustment is rather delicate. It is effected by altering the stroke of the plungers. At the limit of the adjustment, corresponding to very small rates, the sensitivity may be increased by removing one of the valves of the pump, which is generally of two- or three-throw type. In any case, it is advisable to avoid the method of pumping all the mud, and then stopping the pump for some time.

When it is necessary to force the output of the **pump**, it happens sometimes that the strokes of the plunger have repercussions on the general level of the liquid, producing a pulsation in the flow of clarified juice. This phenomenon is detrimental to the subsidation, and should be avoided.

Distributing tank. When a factory is operating with a battery of several clarifiers, it is desirable that the juice be distributed among the several clarifiers in proportion to their respective capacities. For this, the juice will be run into a distributing tank having as many overflow weirs as there are clarifiers in the battery. The crests of these weirs will be at the same level, and the width of each of them may be set by means of an adjustable sliding plate. In this way, their rates of flow will remain in the same proportions, independent of the variations in the total flow.

Liquidation and losses during weekly shutdown. On account of the magnitude of their volume, it is generally not possible to liquidate the clarifiers during the weekly shutdown of the plant. One or more clarifiers are accordingly left full; but this cannot be done without some losses of sucrose. These losses have been studied by Noble and Henderson in Australia^{3, 3a}. The losses are of two kinds:

- (a) Inversion losses, more marked at high temperatures, above 80°C (175°F)
- (b) Losses due to action of micro-organisms, which are more marked at low temperatures, below 70°C (160°F).

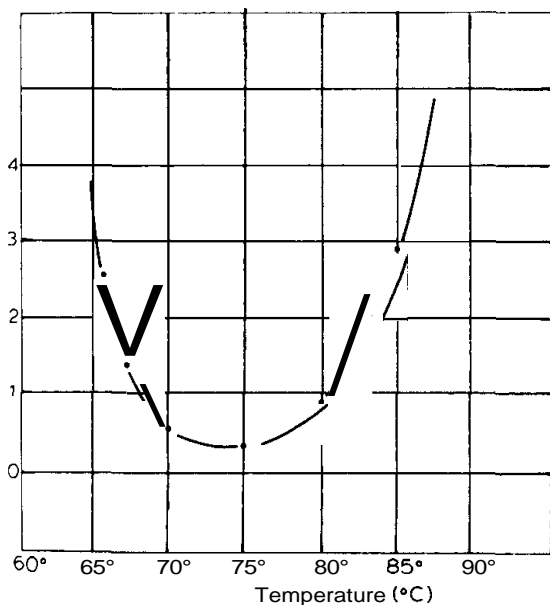


Fig. 29.4. Purity drop over 40-hour period, as a function of temperature.

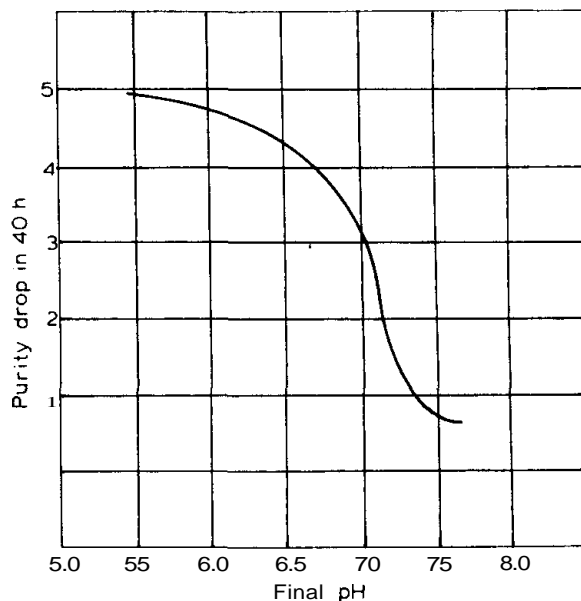


Fig. 29.5. Purity drop over 40-hour period, as a function of pH.

These losses are measured by purity drop, which will thus be minimal between 70 and 80°C (Fig. 29.4). They are also closely related to the pH, and increase when the pH of the juice falls below 7.3 (Fig. 29.5).

In order to reduce these losses to a minimum, it is recommended that lime should be added to the juice, before shutting down, at the rate of 150 p.p.m. of juice (0.35 lb./t). Lime is cheaper and twice as active as sodium carbonate (St. Antoine⁴). This amount corresponds to raising the initial pH to 8.7. Furthermore, in order to keep the juice at the conditions corresponding to the most advantageous portion of the graph of Fig. 29.4 (which, however, has been established without any addition to the juice), it is advisable to cool the juice which is to be left in the clarifier, that is, the juice treated during the last 3 or 4 hours of crushing. With this object, Noble suggests regulating its temperature to 80°C (175°F) or, after shutting down, passing the juice through a heater the vapour space of which has been provided with admission and outlet connections for cold water, which thus can be circulated. This lower temperature will not harm the clarification if care is taken to add at the same time a flocculant such as Separan AP 30. The reduction of temperature to 80°C is much more effective than raising the pH⁴.

Once the juice has been adjusted to 80°C or below, formalin may be added, in the ratio of 0.5–0.6 p.p.m. of juice. According to Noble, this represents only 6% of the value of the sucrose which would otherwise be lost. It is advisable to mix it well, by adding it for example from a charging tank at the entry to the clarifier. There is no point in adding the formalin above 85°C (185°F), as it would volatilise rapidly. For the same reason, it is desirable not to add it until the juice has been reduced to 70°C (160°F) or below.

As the pH indicates very clearly the purity drop, the latter may be detected by observing the pH when it falls below 7.2: deterioration becomes evident.

It is preferable, however, to carry out two or preferably several liquidations in the course of a season. Advantage will be taken of these to clean the vessel thoroughly. For this purpose, times will be chosen when the factory has to be shut down for more than 40 hours (in Reunion, for example, the traditional holidays of 1st–2nd November, possibly the 15th August, when it happens to give a 2-day holiday, etc.).

Inversion losses. During the subsidation, the pH of the juice changes, falling by about half a unit, for example, from 7.5 to 6.9. This drop is more marked in the bottom compartment than in the upper ones.

In spite of all the precautions taken, some inversion losses take place in the clarifier. Honig⁵ estimates such losses at 0.2 or 0.3% of the sucrose, in general, but they can increase sometimes to 1% (*cf.* p. 541).

The muds deteriorate more rapidly than clear juice. If they are maintained at a temperature of approximately 95°C (203°F) and a pH of 6–6.5, the purity drop observed⁶ is of the order of:

$$\Delta p = \frac{t^2}{100} \quad (t \geq 10) \quad \Delta p = \frac{t}{10} \quad (0 < t \leq 10) \quad (29.1)$$

Δp = total purity drop

t = residence time, in hours.

Decomposition of reducing sugars. During the period the juice is in the clarifier, at high temperature, some decomposition of reducing sugars always occurs. Honig⁷ estimates this at about 1% of the reducing sugars, per hour, at 100°C (212°F), when the pH is in the neighbourhood of 7. However, this decomposition can reach 3% per hour, if the pH is increased to about 8.

Velocity of juice. The juice, at the entry to the clarifier, should have a maximum velocity of 15 m/h (50 ft./h). At higher rates, it creates eddies which are detrimental to the subsidation.

In the interior of a compartment, the settling takes place under the following conditions, depending on the velocity V of the liquid:

$V = 3-6$ m/h (10-20 ft./h): perfect laminar flow

$V = 6-12$ m/h (20–40 ft./h): steady flow, giving excellent subsidation

$V = 2-15$ m/h (40-50 ft./h): irregularities begin: subsidation still possible

$V = 15$ m/h (50 ft./h) or more: flow becomes turbulent: subsidation is no longer possible.

Other types of clarifier

In addition to the Dorr and Graver, there are other types of clarifiers, notably the Bach (Fig. 29.6), rather widely distributed in British countries, and of a design broadly similar to that of the Dorr. However, it is generally not covered, the surface of the juice being simply covered by a layer of oil with the object of avoiding convection currents at the surface; the top compartment then takes part in the subsidation process like the others.

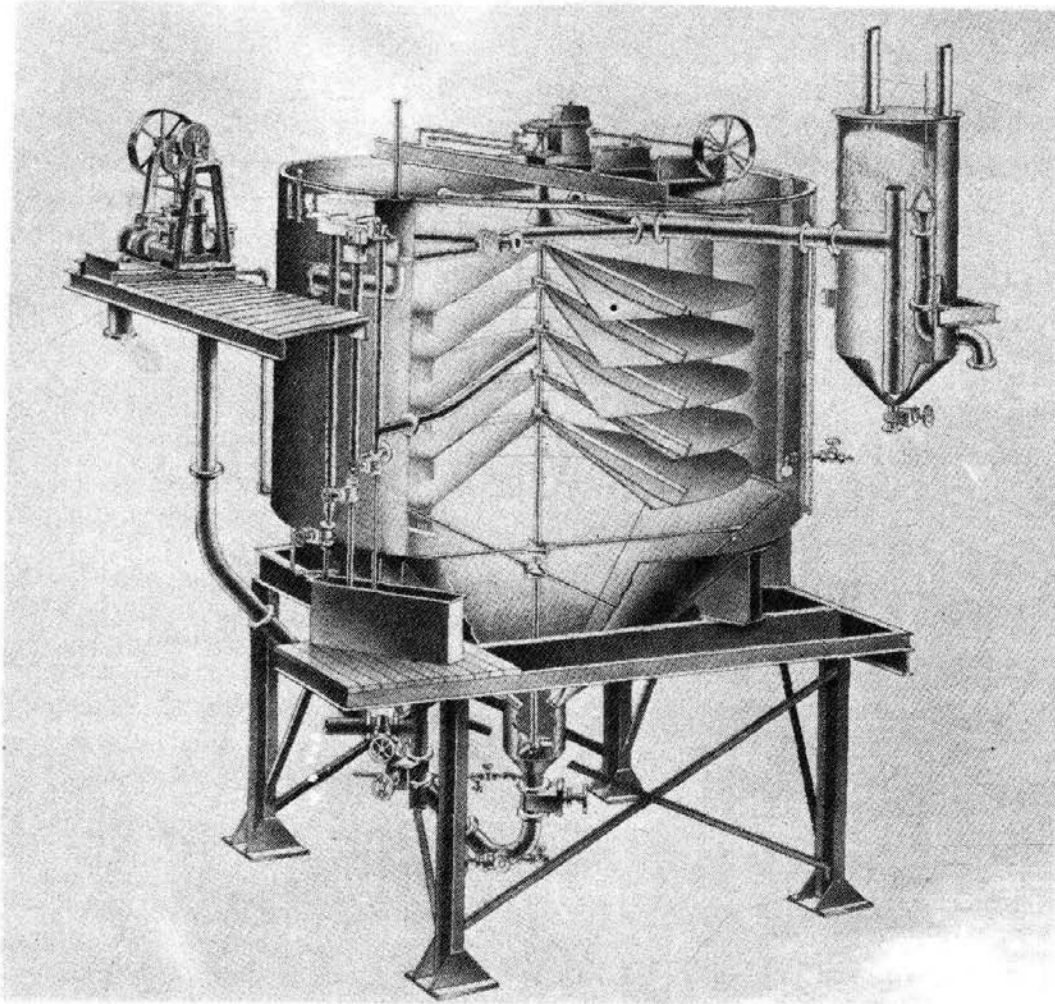


Fig. 29.6. Bach clarifier (Mirrlees-Watson).

Rapid clarifiers

Since the introduction of continuous clarifiers, some defects have been apparent. It was found that in the laboratory, or with certain pilot units, the speed of settling could be much higher. In seeking the causes of this inferior performance of equipment of industrial dimensions, it was found that certain very simple details, once modified, gave a substantial improvement in the settling time of juice in the equipment. These were:

(a) Introduction of the juice to the subsider with the minimum of disturbance, at a much reduced velocity, without eddies; and regulating the flow as much as possible, for example, by provision of a storage tank (20 m^3 (700 cu.ft.)/10Gt.c.h.) to smooth out fluctuations in the rate of flow.

(b) Reduction of length of travel of the juice between entry to the equipment and exit.

(c) Provision of more points for juice outflow, in order to reduce disturbance at those points.

(d) Arranging for the juice to travel upwards through the flocculated juice, so that the particles in suspension are retained in their flow by the existing flocs ("upward sludge filtration").

(e) Division into compartments was found to be detrimental to performance.

These considerations have led to the development of the S.R.I. clarifier, designed by Hale & Whayman⁶ at the Sugar Research Institute in Australia (Figs. 29.7, 29.8 and 29.9). Entry and distribution of the juice are effected by a circular channel of radius equal to two-thirds of that of the subsider; it overflows by a notched channel into an annular downtake, which delivers the juice at the mid-height of the vessel; it then meets a deflector of double slope

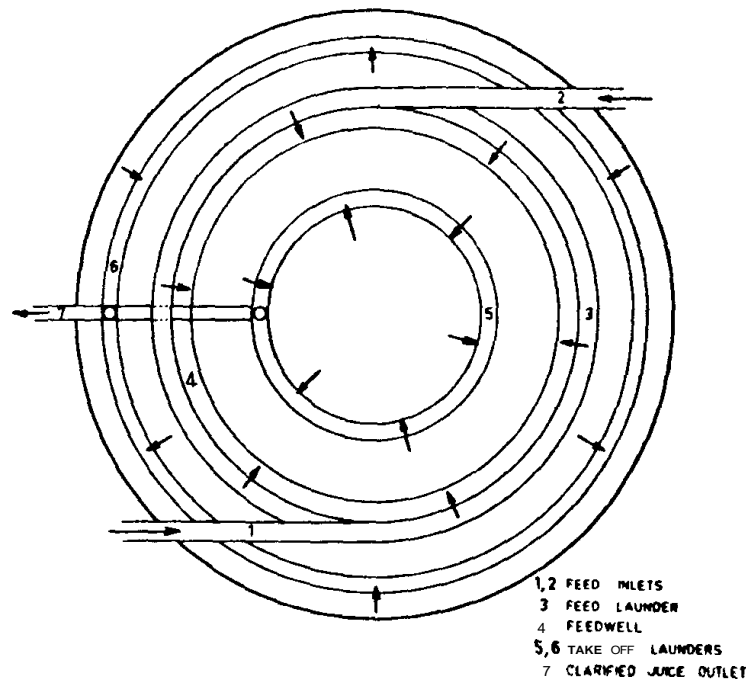


Fig. 29.7. S.R.I. clarifier (plan).

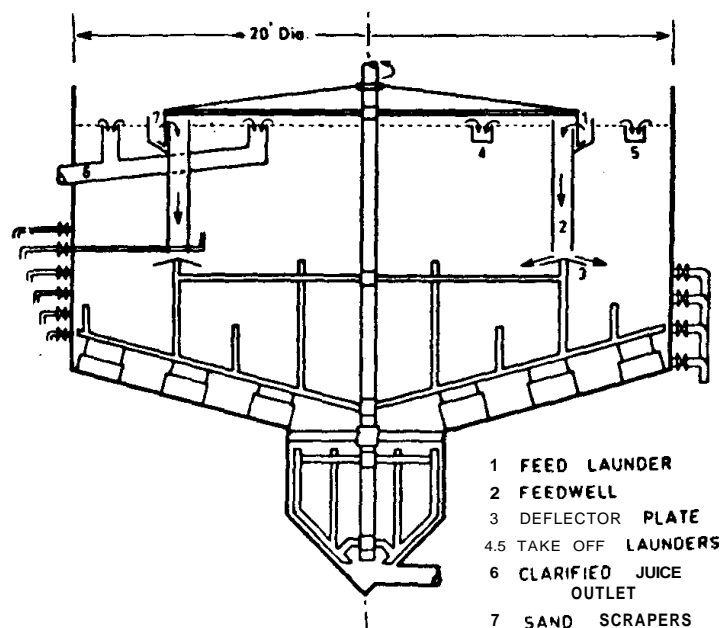


Fig. 29.8. S.R.I. clarifier (section).

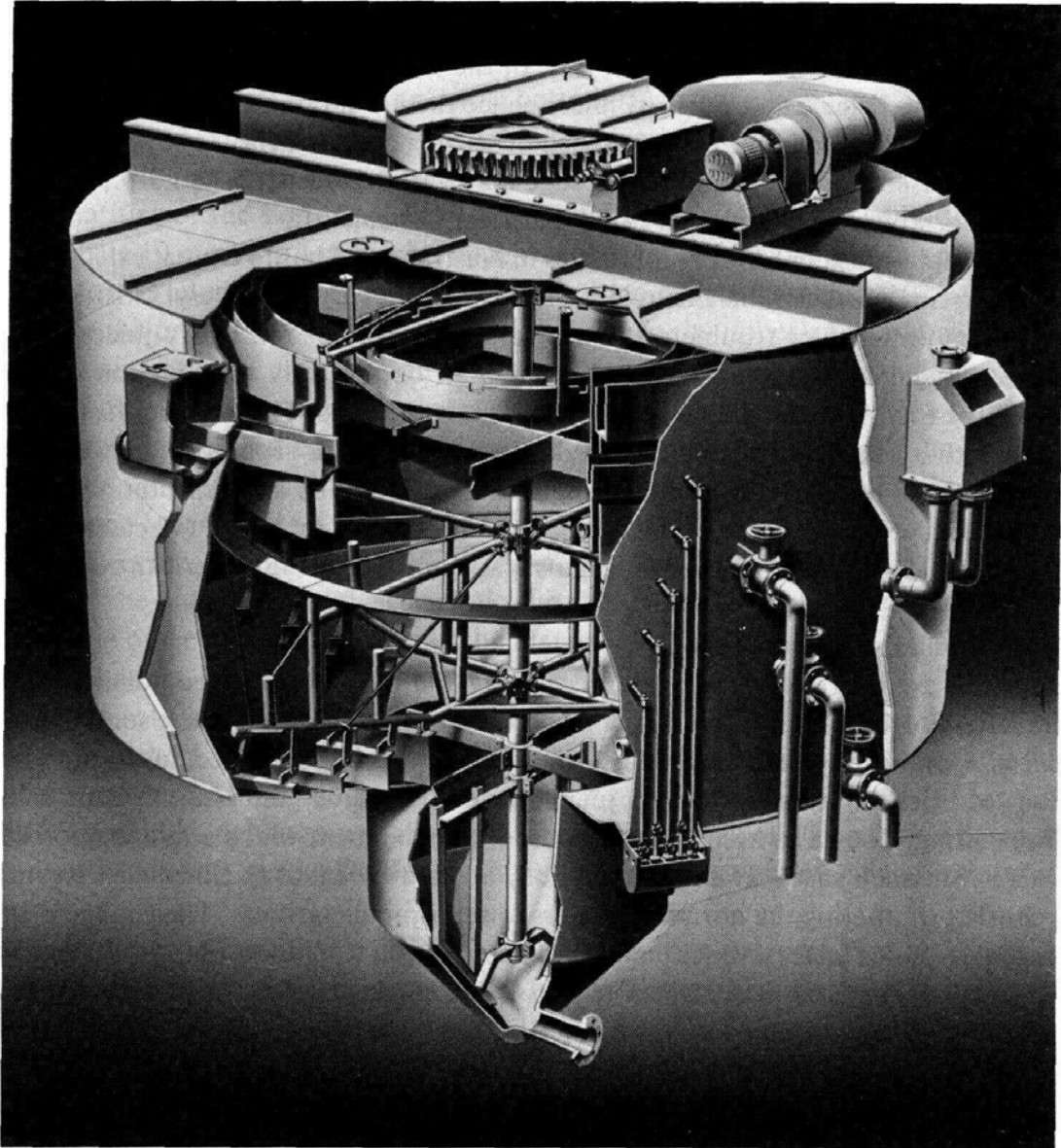


Fig. 29.9. S.R.I. clarifier (broken section] (Fletcher and Stewart).

which distributes it laterally in the equipment. The settled mud accumulates on the bottom, where it is scraped towards a mud well and removed. The scrapers are carried on rotating arms.

The clarified juice overflows by two notched gutters, both circular and concentric; the first or inner one is of radius one-half or two-thirds that of the entry channel, the second approximately half way between the latter and the outer wall of the tank. Fixed scrapers avoid accumulation of mud on the double-sloped deflector which is fixed to the rotating arms.

The rate of settling is of the order of 10 cm (4 in.) per minute, calculated on the interior cross-section of the vessel reduced by that of the deflector. Area may be reckoned as about $0.2 \text{ m}^2/\text{t.c.h.}$ (2 sq.ft./t.c.h.). The depth between the bottom of the annular feed channel and the conical bottom is approximately 1.20 m (4 ft.). The residence time of the juice is approximately 20 min, that of the mud some hours.

Reckoning on the total cross-section of the vessel, we should allow about $0.15 \text{ m}^2/\text{t.c.h.}$ ($1.5 \text{ sq.ft./t.c.h.}$). On account of the brief time of the juice in the subsider, the pH falls only by 0.1 between limed juice and clarified juice; hence there is a decreased risk of decomposition of juice. These considerable gains in speed of settling and residence time are made possible only by the use of a flocculant. The dosage may vary between 1.5 and 3 p.p.m. It is in fact flocculants which have rendered rapid clarifiers possible.

The S.R.I. rapid subsider now predominates in Australia and has to a great extent replaced older clarifiers in the Australian factories. These factories express every satisfaction and report no inconvenience or trouble resulting from stale cane or refractory juices.

In the beet sugar industry, a similar gain has been obtained by introduction of the Eis clarifier, of the Enviro-Clear Co., following the same principles. This clarifier has moreover been fitted with a photoelectric analyser to detect the surface of separation between cloudy and clear juice and maintain it steady. This clarifier has attained a sedimentation rate close to 600 kg of solid material per m^2/h (120 lb./sq.ft./h), or $12.7 \text{ m}^3/\text{m}^2/\text{h}$ ($42 \text{ cu.ft./sq.ft./h}$) of juice, and has also been installed in cane sugar factories in the American zone.

Modification of existing subsiders

Many factories are interested in the gain in capacity and in space rendered possible by rapid clarifiers. On the other hand, they hesitate on account of troubles which have sometimes been experienced with the use of these clarifiers. They may in fact be concerned due to more severe conditions of brief duration: refractory juices, cane affected by gumming, or cane held too long since cutting. They also hesitate to dispense with equipment which is still in good condition. There is in such cases an elegant solution, applied by Lindsay Lincoln in Reunion.

This consists of altering the normal clarifiers to semi-rapid clarifiers. The change is simple and inexpensive. In a factory which had a Multifeed and a Rapidorr, both of 7.3 m (24 ft.) diameter, the Rapidorr has been altered and is sufficient for a capacity increase to 200 t.c.h. ; in another factory which had four clarifiers, a 1.2-m (14-ft.) and a 5.5-m (18-ft.) Multifeed, a 6.1-m (20-ft.) Graver, and a 6.7-m (22-ft.) Rapidorr, the Rapidorr has been altered and is sufficient alone for a capacity increase to 150 t.c.h. We may comment that the juice of a variety R, general in Reunion, is rather difficult to clarify, without being really refractory.

The idea has been not to seek a truly rapid clarifier, which may risk sometimes becoming insufficient, but to aim at a semi-rapid unit, that is, one calculated for a residence time of **50–60 min.** The following, for example, describes the alteration of an older clarifier:

(a) Remove the intermediate plates to convert the equipment to a two-compartment clarifier. It would be permissible to remove all the compartments, but maintaining two compartments permits operation as an ordinary clarifier in case of a lack of flocculant for any reason.

(b) Increase the volume of the flash tank, which would become too small for the quantity of juice to be handled (see later calculation for flash tanks).

(c) For the same reason, enlarge all the piping and openings for juice feed.

(d) Locate the openings for feed of dirty juice to the two compartments, so that they are placed at the level of the top of the mud pit.

(e) Widen and deepen the mud pits at the bottom of the two compartments which are retained. Replace their vertical sides by plates inclined at 45° .

(f) Modify the scrapers and the supporting arms so that they fit the bottom plates of the mud pit and the compartment.

(g) Locate the scrapers so that the whole bottom surface is swept, and incline them at 45° to the direction of rotation, giving them a slight overlap.

(h) Arrange the scrapers, not vertically, but pivoted and trailing, so that they follow better the irregularities of the bottom plate. They should be terminated at the bottom by a small vertical rebate 10 mm ($\frac{1}{2}$ in.) in height, which scrapes more effectively than an inclined plate. The slope to be adopted relative to the bottom plate is approximately 60° .

(i) Place the openings for discharge of clear juice in the upper angle of each of the compartments, and provide an increased number of such openings. For example, placing one every 15° on the two semicircular pipes so located, this corresponds to 24 openings. These openings would be placed on the upper surface of the pipe, but every 30° an orifice on the lower face of the pipe should be located to ensure complete draining of the pipe for cleaning. Each half pipe covers 180° ; it should be of uniform diameter throughout its length. The juice outlet openings are circular, without restriction, obtained by simple cutting. There are six on each side of the outlet pipe to the clear-juice box, of which the three closest to the outlet are $D/3$, and the other three $D/2$, D being the diameter of the pipe, which remains open on its lower third, at the end.

(j) Ensure satisfactory removal of the mud from the bottom of each mud well.

(k) Place at the top of each compartment two degassing pipes discharging at 0.50 m (20 in.) above the roof of the subsider, 180° apart.

(l) It is desirable to maintain at the top of the subsider a small flocculation compartment, the level of which will be regulated automatically so as to maintain a slight accumulation of scum.

The clarifier thus modified will have a minimal horizontal path for juice and an even smaller vertical path, with the departing juice having to pass through the settling flocs already formed. The use of a flocculant is essential. It will be added after the flash tank at the rate of 2–3 p.p.m., according to its efficacy. This can be increased to 4 p.p.m. if considered desirable. It should be introduced by metering pumps at a controlled rate.

Since such alteration of clarifiers is somewhat complex, it is recommended that technologists wishing to make such changes should enquire from Tate & Lyle (Process Technology), sending them plans of the old equipment to be altered. They will furnish designs permitting the necessary modifications.

Flash tank

Clarifiers are normally preceded by a flash tank. This is a simple cylindrical tank located just above and ahead of the clarifier, with a flue open to the atmosphere. The juice from the heaters discharges tangentially into this tank; since the juice has been brought to $101.5 - 104.5^\circ\text{C}$ ($215 - 220^\circ\text{F}$), it partially flashes into vapour when discharged into this vessel at atmospheric pressure. This flashing removes from the suspended particles the air bubbles attached to them, which, if not removed, would prevent particles of bagasse from settling during the clarification process.

Maintenance of clarifiers

At the end of the crushing season, the interior of the clarifiers is sometimes painted; this is very useful for preserving them during the slack season but, even if the paint selected is considered to be resistant to acid juices, the paint disappears during the first weeks of the next season.

Height and number of trays

We have already remarked (*cf.* p. 431) that the volume of settlers does not influence the sedimentation; the area only determines the operating capacity. Thus, there might be a tendency to reduce the height of the trays. This is limited, however, by the necessity to allow for inspection and maintenance of the apparatus.

Generally, the height per tray is made not less than 60 cm (24 in.). The usual height, for the Dorr Multifeed, was 69 cm (27 in.). The bottom compartment, where the muds collect, was two or three times higher than the other. In the Rapidorr, the upper compartments are 122 cm (4 ft.) in height, and the mud compartments 183 cm (6 ft.).

The Multifeed was made in 3, 4 or 5 compartments; the Rapidorr always has 4; and the Graver generally has 4.

A clarifier is moreover more economical, in cost and space occupied, if it has more compartments. It has sometimes been maintained that the juice was clearer if the clarifier had fewer compartments. However, in factories having a set of clarifiers with differing numbers of compartments, a consistent and detectable difference between their juices has scarcely ever been detected. In particular, the juice of the Rapidorr is not superior to that of the Multifeed types.

Evolution of clarifiers

The foregoing considerations on compartmented clarifiers establish the point that, since the appearance of continuous decantation, subsidors have evolved continually in the direction of a decrease in the number of compartments for simple decanting of juice relative to the number for thickening of mud. The evolution of the Dorr clarifier is very definite from this point of view: the Multifeed had 5, 4 or 3 compartments and only one mud thickener. Then, in 1956, Sabi⁸ commented that the surface required for thickening of mud was the important factor in the capacity of the equipment. He estimated accordingly that, according to the nature of the juice and the cane variety, it required 0.50–1 m², averaging 0.75 m² (7.5 sq.ft.), for thickening of mud per t.c.h. Since the speed of settling is approximately 1 cm/min = 60 cm/h (2 ft./h), and since 1 t.c.h. corresponds roughly to 900 dm³/h of juice, an area of 0.75 m² would permit the settling of 0.75 x 0.60/0.900 = 0.5 t.c.h.; consequently, the capacity per tray for settling would be half that for thickening. Hence the Rapidorr A3T, which has two trays for settling for each thickening tray. This would make 1–2 m² (10–20 sq.ft.) of total area/t.c.h., averaging 1.50 m²/t.c.h. (15 sq.ft./t.c.h.).

Progress has continued in the efficacy of the equipment, and Dorr has finally produced the Rapidorr 444 with 4 independent compartments superposed, all providing decantation and thickening at the same time. The rapid clarifiers without compartments are the outcome of this evolution.

Capacity

This evolution explains why it is difficult to give figures for capacity applicable to all clarifiers. The scatter of values is particularly marked where volume is concerned; it is less marked for surface area for decantation. The values given have evolved parallel with the conception and the efficacy of the equipment. Commencing from the older multicompartiment models, capacities are as follows:

	Volume		Area	
	($m^3/t.c.h.$)	($cu.ft./t.c.h.$)	($m^2/t.c.h.$)	($sq.ft./t.c.h.$)
Hawaii ⁹	2.5-3.8 (av. 3)	90 – 136 (av. 108)	–	–
Cuba ¹⁰	2.4	86	2.5	27
Puerto Rico ¹¹	2.1 – 3.8	75 – 136	–	–
S. Africa ¹²				
Old clarifiers	2.5-3.7	90 – 133	–	–
Rapidorr 444				
(one factory)	2.4	86	1.7	19
S.R.I. only				
(3 factories)	0.65 – 0.92	23 – 33	0.22-0.33	2.4-3.6

	Juice flow per unit vol.		Juice flow per unit area	
	($kg/h/m^3$)	($lb./h/cu.ft.$)	($kg/h/m^2$)	($lb./h/sq.ft.$)
Australia				
1968 ¹³ : Bach	550	34	500	102
Dorr ATV	450	28	650	133
1969 ¹⁴ : Bach	400	25	500	102
ATV	500	31	750	154
Philippines ¹⁵				
AMS	650	41	–	–

(In some cases it is stated that capacity varies with refractory juices.) Table 29.1 gives figures for the Rapidorr 444 Series.

Surface area. The area of the clarifiers for decantation is reckoned on the gross cross-section, that is, the interior gross area of the vessel, without deduction for the central tube, multiplied by the number of compartments. In the case of the Graver, the total cross-section is similarly taken, including the peripheral channel, since it participates in the settling.

Consequent on the remarks indicated concerning compartments, the capacity is better when the ratio of total number of compartments to number of thickening compartments is lower; for example, Multifeed < A3T < 444.

TABLE 29.1

RAPIDORR 444 SERIES

Diameter		Volume		Area		Capacity	
(m)	(ft.)	(m ³)	(U.S. gal.)	(m ²)	(sq.ft.)	(t.c.h. in favourable conditions)	(nominal, short tons/h)
3.048	10	41	10,800	29	312	18	16
3.658	12	59	15,600	42	452	26	23
4.267	14	81	21,300	57	614	36	32
4.877	16	105	27,800	75	804	47	42
5.486	18	133	35,200	94	1,016	59	53
6.086	20	164	43,400	116	1,254	73	69
6.706	22	199	52,700	141	1,516	88	79
7.315	24	237	62,700	168	1,830	105	94
7.925	26	278	73,400	198	2,128	124	111
8.534	28	322	85,000	229	2,460	143	128
9.144	30	369	97,600	262	2,824	164	147
9.754	32	421	111,200	299	3,220	187	168
10.363	34	474	125,200	337	3,630	211	189
10.973	36	533	140,800	378	4,068	236	212

In these clarifiers, the height of the cylindrical portion is 5.5 m (18 ft.); for the earlier Rapidorr clarifiers, it was 6.1 m (20 ft.).

Dorrclones

Modern methods of cane harvesting send to the factory, with the cane, not only trash proper with cut ends and roots, but also foreign matter of various kinds, some of which are dangerous, such as stones (see p. 48). These items consist mainly of soil, sand, and gravel, of which only the finest particles are conveniently handled with the muds in filtration.

The largest particles contribute to wear of the pumps, and interfere with the fabrication. The Dorr – Oliver Company offers equipment designed for eliminating them: the "Dorrclone". It is composed of cyclone separators into which the mixed juice, unscreened, is pumped under a pressure of about 1.5 kg/cm² (22 p.s.i.). The juice is introduced tangentially and spun in the equipment. The clear juice leaves above the point of entry, while the heavy particles fall to the bottom of the cyclone and pass into a sand box, where they are extracted intermittently by an automatic device.

On account of the wear to which they are subjected, these cyclone separators are lined with rubber, which normally lasts a full season.

The operation takes place in two stages. After passage through a first set of cyclones, the solid matter from the equipment is deposited in a bath where it is washed with water to remove the entrained sucrose. The mixture is then passed to a second cyclone, the refuse from which is discarded and the clear juice sent for use as imbibition, since it contains a little sucrose.

Since the equipment must function under pressure and at a constant rate, Dorrclones cannot be subjected to stoppages of the factory: hence they are arranged to operate with recirculation, assuring a constant rate of throughput.

It is preferable that the cyclone be installed vertically, but they may also function in an inclined position, provided that the slope of the cone remains at a positive angle.

There are three sizes of Dorrclones, designated by the diameter of the entry chamber:

$D = 152 \text{ mm (6 in.)}$: capacity = 11 – 25 m³/h (47 – 110 U.S. gal./min)

$D = 305 \text{ mm (12 in.)}$: capacity = 42- 84 mVh (185- 350 U.S. gal./min)

$D = 610 \text{ mm (24 in.)}$: capacity = 162-345 mVh (710 – 1,520 U.S. gal./min)

They are often installed in batteries of 3 cyclones for the first stage, with 1 for the second stage.

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30. Juice Heating

We have seen that in the course of treating the juice, it must be heated at least once. High-pressure steam being very valuable, exhaust steam is used for this heating or, if possible, preferably bled vapour from the evaporators. It is thus necessary to have a heat exchanger between vapour and juice: this is provided by the juice heaters.

The juice heater (Fig. 30.1) consists of an assembly of tubes: the juice circulates through the tubes, and the vapour outside them. Suitable headers force the juice to pass a certain number of times from bottom to top and from top to bottom of the heater by restricting the juice each time to a few of the tubes.

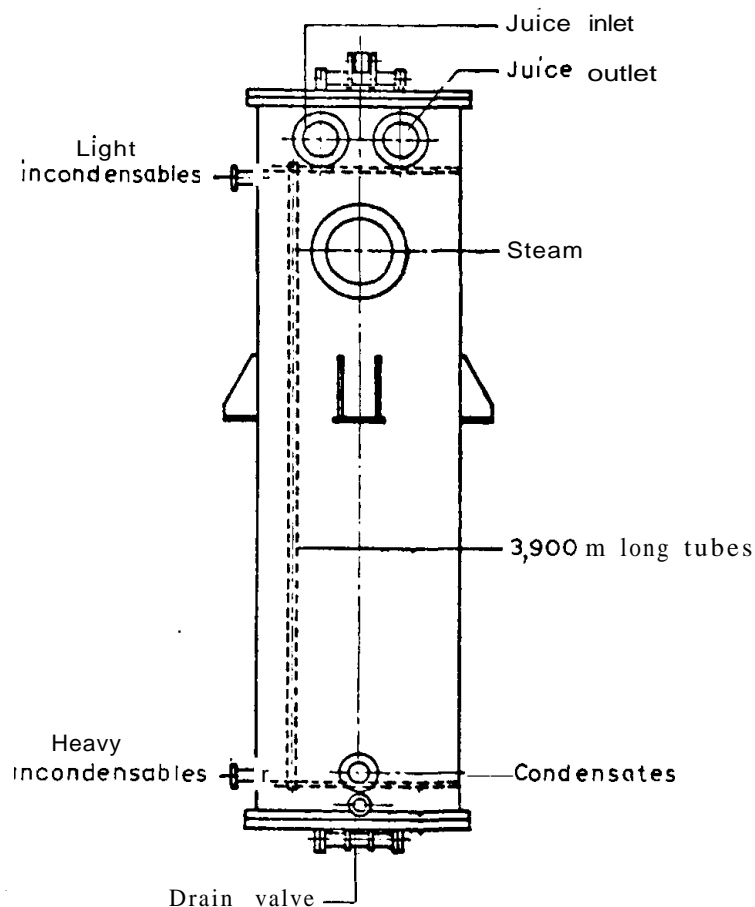


Fig. 30.1. Juice heater (Fives Cail - Babcock).

Specific heat of sugar solutions

The specific heat c of sucrose solutions is given, to a close approximation, by the equation:

$$c = 1 - 0.006 B \quad (30.1)$$

B = brix of the solution.

This formula may be applied without serious error to juices, syrups and molasses of different purities.

It follows that a typical mixed or defecated juice, of 16° or 17° brix, will have a specific heat of approximately 0.9. Using this value for juice will never involve a serious error. For greater precision, when the brix differs appreciably from the mean, it will be preferable to use the value given by eqn. (30.1), which moreover is simple and readily calculated.

According to Gucker and Ayres¹, more accurate values would be obtained by replacing the coefficient 0.006 by 0.0056 in eqn. (30.1), and by taking into account the temperature t (°C) and purity P of the juice:

$$c = 1 - \left[0.6 - 0.0018 t + 0.0008 (100 - P) \right] \frac{B}{100} \quad (30.2)$$

Heat losses

According as the lagging of the heater is more or less effective (and, for batteries of heaters, the lagging of the piping connecting the heater to the following one), the loss of heat, which determines the efficiency of the heat-exchange operation, will range from about 4 to 8%, averaging 5% for a heater suitably lagged and covered with wooden battens.

CALCULATIONS FOR HEATERS

Calculations for heaters are complicated by the fact that while one of the fluids, the vapour, is at constant temperature, the other, *i.e.* the juice, is at a varying temperature in its passage from entry to exit. This introduces an integral which is expressed by a logarithm. This renders the calculation somewhat lengthy, but any precise calculation would not be possible otherwise.

It will be seen, moreover, in the following example, that the resulting complication is indeed minimised: there is not even need for a table of logarithms; a simple slide rule will furnish results sufficiently precise for practical requirements.

The whole calculation for heaters is contained in the 3 equations following:

(a) *Quantity of heat transmitted:*

$$M = pc (T - t_0) (1 - e^{-kS/pc}) \quad (30.3)$$

(b) *Temperature obtained:*

$$t = T - (T - t_0) e^{-kS/pc} \quad (30.4)$$

(c) *Heating surface:*

$$S = \frac{M}{k} \ln \frac{T - t_0}{T - t} \quad (30.5)$$

M = quantity of heat transferred to the juice, in kcal (B.T.U.)

S = heating surface of the heater, in m^2 (sq.ft.)

p = weight of juice to be heated, in kg/h (lb./h)

c = specific heat of the juice (approx. 0.9)

T = temperature of the heating vapour, in $^{\circ}\text{C}$ ($^{\circ}\text{F}$)

t_0 = initial temperature of the cold juice, in $^{\circ}\text{C}$ ($^{\circ}\text{F}$)

t = final temperature of the hot juice, in $^{\circ}\text{C}$ ($^{\circ}\text{F}$)

k = coefficient of heat transfer, in kcal/ $m^2/^{\circ}\text{C}/h$ (B.T.U./sq.ft./ $^{\circ}\text{F}/h$).

Recall that the expression $y = e^{-x}$ reduces to:

$$-x = \ln y = 2.3 \log y \quad (30.6)$$

Hence:

$$\log e^{-kS/pc} = -0.4343 \frac{kS}{pc} \quad (30.7)$$

and that, when a logarithm has a negative value, we must write, for example:

$$\log x = -0.372 = \bar{1}.628$$

Value of heat-transfer coefficient

All authors are in agreement in indicating the marked influence exerted on the heat-transfer coefficient k by the velocity V of the juice circulation in the tubes; but opinions differ concerning the law expressing the relationship of these two quantities.

Hausbrand² gave:

$$k = 750 \sqrt[3]{V + 0.007} \quad (k = 103 \sqrt[3]{V + 0.023}) \quad (30.8)$$

k = heat-transfer coefficient of the heater, in kcal/ $m^2/^{\circ}\text{C}/h$ (B.T.U./sq.ft./ $^{\circ}\text{F}/h$)

V = velocity of juice in the tubes, in m/s /ft./s).

However, his work was done on clean tubes.

An expression previously used in Australia³ was:

$$k = 900 \sqrt{V} \quad (k = 100 \sqrt{V}) \quad (30.9)$$

Speyerer⁴ gives for $V = 1$ m/s (3.28 ft./s):

$$k = 222 + 3.6 r \quad (k = 45 + 0.41 (T - 32)) \quad (30.10)$$

However, these formulae have the disadvantage of taking into account only the effect of heat transfer from tube (or scale) to juice, in which the juice velocity is important. It is desirable to take into account equally the resistance offered to heat transfer from vapour to metal. The complete and exact theoretical formulae are much too complicated for practical usage. They involve the viscosity and the Nusselt, Grashof and Prandtl numbers, at various powers. The multiplicity of practical formulae offered and the great variability of the coefficient k in practice have prompted the author to propose a formula which should be the simplest possible, while taking into account the results now gathered on the influence of velocity U of juice. We propose accordingly:

$$k = 6 T \left(\frac{U}{1.8} \right)^{0.8} \quad \left(k = 0.7 (T - 32) \left(\frac{U}{6} \right)^{0.8} \right) \quad (30.11)$$

which gives, for example, with heating steam at 115°C (239°F):

For $U = 1.5$ m/s	(5 ft./s):	$k = 596$ kcal/m ² /°C/h	(125 B.T.U./sq.ft./°F/h)
For $U = 1.8$ m/s	(6 ft./s):	$k = 690$ kcal/m ² /°C/h	(145 B.T.U./sq.ft./°F/h)
For $U = 2.1$ m/s	(7 ft./s):	$k = 780$ kcal/m ² /°C/h	(164 B.T.U./sq.ft./°F/h)

Influence of scale on heat transmission

Formation of scale on the heating surfaces has a marked influence on the coefficient of heat transfer. On an exchanger with straight tubes, we have in effect:

$$\frac{1}{k} = \frac{1}{\kappa_1} + \frac{1}{\kappa_2} + \frac{1}{\kappa_3} \quad (30.12)$$

κ_1 = heat-transfer coefficient, steam to tube

κ_2 = heat-transfer coefficient across the tube = e_2/λ_2 (e_2 = thickness of wall)

κ_3 = heat-transfer coefficient from tube to juice.

Staub and Paturau⁵ give the following values, which are expressed here for tubes 1.2 mm in thickness (35.6 X 38 mm, for example):

$$\kappa_1 = 10,546 \text{ kcal/m}^2/\text{°C/h} \text{ (2,160 B.T.U./sq.ft./°F/h)}$$

$$\kappa_3 = 1,758 \text{ kcal/m}^2/\text{°C/h} \text{ (360 B.T.U./sq.ft./°F/h)}$$

$$\lambda_2 = 37 \text{ kcal/m}^2/\text{°C/h/m} \text{ (25 B.T.U./sq.ft./°F/h/ft.) for steel}$$

$$\lambda_2 = 94 \text{ kcal/m}^2/\text{°C/h/m} \text{ (63 B.T.U./sq.ft./°F/h/ft.) for brass}$$

and:

$$\kappa_2 = 30,830 \text{ kcal/m}^2/\text{°C/h} \text{ (6,350 B.T.U./sq.ft./°F/h) for steel}$$

$$\kappa_2 = 78,300 \text{ kcal/m}^2/\text{°C/h} \text{ (16,000 B.T.U./sq.ft./°F/h) for brass}$$

Hence:

$$k = 1,437 \text{ kcal/m}^2/\text{°C/h} \text{ (294 B.T.U./sq.ft./°F/h) for steel}$$

$$k' = 1,477 \text{ kcal/m}^2/\text{°C/h} \text{ (303 B.T.U./sq.ft./°F/h) for brass}$$

This shows that the influence of the metal used for the tubes is insignificant, contrary to general ideas.

On the other hand, if scale is formed, the factor $1/k_3$ in eqn. (30.12) should be replaced by $1/k_4$ and $1/k_5$, for the transmission from metal to scale, and from scale to juice, respectively. Now if these coefficients are much lower than k_3 , sometimes of the order of several units, depending on the material and composition of deposits, they will cause a substantial drop in the final overall coefficient k , as the deposition of scale increases.

Formula (30.11) makes a large allowance for scale, and the coefficient 6 (0.7) can often be replaced by a figure up to 8 (0.9), if there is little fear of scale formation.

Very approximately, we may say that the heat-transfer coefficient of a heater cleaned regularly is of the order of $600 \text{ kcal/m}^2/\text{°C/h}$ ($125 \text{ B.T.U./sq.ft./°F/h}$), and that it would fall to approximately $400 - 300$ ($80 - 60$) if it were operated for two or more weeks without cleaning.

Giving a well-studied formula and taking into account the number of hours H since the last cleaning of the heater, Wright⁶ gives, for tubes of stainless steel (metric units):

$$\frac{1}{K} = \frac{D^{0.25}}{4.35} + \frac{1}{13.5} + 0.0035 H^{0.8} \left[\frac{10,763}{U^3} \right] + \frac{1}{0.283 D^{-0.2} U^{0.8} t_m^{0.467}} \quad (30.13)$$

D = diameter of tubes, in m

K = heat-transfer coefficient, in $\text{kW/m}^2/\text{°C}$

t_m = mean temperature of juice, in $^{\circ}\text{C}$; we may take: $t_m = (t + t_0)/2$

U = velocity of juice, in m/s.

If it is a case of a primary heater, the figure 4.35 may be replaced by 3.8, and 0.283 by 0.295. For brass tubes instead of stainless steel, 13.5 would be replaced by 64.

If it is desired to ascertain the lowest efficiency of the heater in the course of the week taking into account Wright's results, eqn. (30.11) may be modified, taking:

$$k = 5 \left(\frac{U}{1.8} \right)^{0.8} \left(\frac{135 - H^{0.8}}{100} \right) \left(k = 0.6 (T - 32) \left(\frac{U}{6} \right)^{0.8} \left(\frac{135 - H^{0.8}}{100} \right) \right) \quad (30.14)$$

2nd formula. We give another formula which avoids the use of the power 0.8, while still taking into account the increasing scale formation and which gives analogous results. The derivation of the formula follows.

In most sugar countries, the factory is operated continuously during the week and is shut down on Sundays to clean heaters and evaporator vessels. These items of equipment are thus practically clean on Monday morning and accumulate scale throughout the week (or throughout the fortnight, where cleaning is done only at alternate weekends). It is interesting to study the variation of this coefficient of heat transmission during that period.

It is logical to assume that the thickness of scale in the heaters and evaporators is proportional to the weight of liquid heated or evaporated, thus to the quantity of heat transmitted. We may thus write:

$$k = \frac{1}{\frac{1}{k_0} + \lambda M} \quad (30.15)$$

t = time elapsed since previous cleaning

k — heat-transfer coefficient at time t

k_0 = heat-transfer coefficient at $t = 0$

M = total quantity of heat transferred since time $t = 0$

X = coefficient.

Hence:

$$M = \frac{1}{X} \left(\frac{1}{k} - \frac{1}{k_0} \right) \quad (30.16)$$

M , however, is proportional to the coefficient k and to time:

$$M = \alpha k t \quad (30.17)$$

a = numerical coefficient.

From (30.16), we have:

$$dM = - \frac{dk}{\lambda k^2}$$

and, from (30.17):

$$dM = \alpha k dt$$

Hence:

$$\alpha k dt = - \frac{dk}{\lambda k^2} \quad - \frac{dk}{k^3} = \alpha \lambda dt$$

Integrating:

$$\frac{1}{k^2} = 2 \alpha \lambda t + b$$

where b = constant

Or:

$$k = \frac{1}{\sqrt{b + at}}$$

placing $2 \alpha \lambda = a$.

The relationship just established for the apparent heat-transfer coefficient k may be equally well established for the evaporation coefficient c (cf. p. 571).

If we express t in days and it is desired to express the coefficients a and b as functions of figures generally established, we may write:

$$k = k_0 \sqrt{\frac{A}{A + J}} \quad c = C_0 \sqrt{\frac{A}{A + J}} \quad (30.18)$$

k_0 = heat-transfer coefficient at $t = 0$

C_0 = evaporation coefficient at $t = 0$

A = a number varying with the rapidity of scale deposition, which in turn depends on the juice, intensity of circulation in equipment, design of equipment, etc.

J = number of days elapsed since cleaning the heater or the vessel.

We may take for an ordinary heater or multiple effect:

For favourable conditions: $A = 6$

For average conditions: $A = 4$

For unfavourable conditions: $A = 3$

Example. A heater has an overall heat-transfer coefficient k_0 of 1,000 kcal/m²/°C/h (205 B.T.U./sq.ft./°F/h) on Monday morning. If it is operating under fairly good conditions, characterised by $A = 6$, the coefficient on Saturday night will fall to

$$k = 1,000 \sqrt{\frac{6}{6 + 6}} = 1,000 \times 0.707 = 707 \text{ kcal/m}^2/\text{°C/h} \quad (145 \text{ B.T.U./sq.ft./°F/h})$$

If it is desired to take into account the variation in heat-transfer coefficient over the course of the week, we may take:

$$k = 7T \left(\frac{T}{1.8} \right)^{0.8} \sqrt{\frac{A}{A + J}} \quad \left(k = 0.8 \left(\frac{T}{32} \right)^{0.8} \sqrt{\frac{A}{A + J}} \right) \quad (30.19)$$

an equation taking into account the day of the week. We may also replace days by hours by substituting for:

$$\sqrt{\frac{A}{A + J}} \quad \text{the fraction:} \quad \sqrt{\frac{24A}{24A + H}}$$

H = hours elapsed since cleaning.

Variability in the coefficient

The following table, quoted by Perk⁷, gives an indication of the variability of the heat-transfer coefficient k , as ascertained in an investigation made in Java in 1940 on the value of the overall coefficient k for various vertical heaters in 9 factories:

	Value of k	
	(kcal/m ² /°C/h)	(B. T. U./sq.ft./°F/h)
Heated by exhaust steam:	225 – 1,127	46-231
Heated by 1st-effect vapour:	212 – 1,080	43 – 221
Heated by 2nd-effect vapour:	201 - 630	41 – 129
Heated by 3rd-effect vapour:	129- 612	26-125
Heated by last-effect vapour:	276- 517	57-105

Webre⁸ reckons approximately 1,200-1,500 kcal/m²/°C/h (250-300 B.T.U./sq.ft./°F/h), values appreciably higher than those quoted by Perk.

On account of these variations, it is advisable to determine the coefficient k for the existing heaters in the factory, to deduce from it the value of a factor to apply to the second term of eqn. (30.11) or (30.19) and, using the various values found, to apply these equations with the factor which appears most suitable for the case under consideration.

It may be recalled that at the beginning of the week, that is, with clean tubes, a correctly designed heater, with sufficient juice velocity, has a heat-transfer coefficient of the order of 1,000 kcal/m²/°C/h (200 B.T.U./sq.ft./°F/h).

Circulation velocity

We have seen the important effect of juice velocity in the tubes on the efficiency of a heater. This is the reason for dividing heaters into compartments separated by baffles.

For effective use of this equipment, it is advisable that the juice velocity should not fall below 1.5 m/s (5 ft./s). Otherwise, not only would the heat-transfer coefficient be lower on Monday morning, but the heater would foul more rapidly, and the temperature of the hot juice would fall all the more rapidly during the week.

On the other hand, at high velocities the passage of the juice through the heater causes a marked pressure drop, which rapidly becomes prohibitive. For this reason, a velocity of 2.2 m/s (7–7.5 ft./s) is seldom exceeded, and the best velocities to be aimed at, from the economic viewpoint, are between 1.5 and 2 m/s (5 and 6.5 ft./s).

Pressure drop

The pressure drop suffered by the juice in its passage through a heater is calculated according to the following considerations:

(/) The kinematic viscosity of the juice, $\nu = \eta g / \bar{\omega}$, is of the order of $(0.5 - 1) \times 10^{-6}$ m²/s at the reheating temperatures (40- 100°C) (c/. p. 1117):

ν = kinematic viscosity, in mVs

η = absolute viscosity, in kg s/m²

ω = specific gravity, in kg/m³

$g = 9.81$ m/s².

(2) The Reynolds number $R = VD/\nu$ (dimensionless - *cf.* p. 1116) is of the order of 50,000–200,000. The flow is therefore turbulent (flow is laminar when R is less than 2,320, turbulent if R is above 3,000):

V = velocity of juice in the tubes, in m/s

D = diameter of tubes, in m.

(3) The coefficient of resistance X (*cf.* p. 1118) varies considerably according to the condition of the tube. When it is new or clean, it may be considered as smooth; the coefficient X (dimensionless) is then of the order of 0.016–0.020. As soon as the tube becomes scaled or dirty, it should be considered as rough, and the coefficient then varies between 0.025 and 0.100. It may be assumed that at the end of the week it often reaches 0.05.

Since head losses are generally used in calculating pump requirements, it is desirable to base them on the values assumed by X at the end of the week.

(4) The loss of head due to the 180° change of direction at the end of each pass may be considered as equivalent to a length of 1.5 times the diameter of the tube. Since there are 2 changes of direction for each tube, one at each end, the equivalent tube length, taking these into account, becomes:

$$L = l + (2 \times 1.5) D = l + 3D$$

The head loss is thus given by:

$$\Delta p = X \frac{\omega}{2g} U^2 \frac{L}{D} \quad (30.20)$$

Δp = loss of head in the whole heater, in kg/m²

L = total equivalent length of travel in the heater, taking into account comment (4), with:

$$L = n (l + 3D)$$

n = number of passes in the heater

l = length of tube, in m.

Taking the value at the end of the week for X and the mean values for the terms:

$$\Delta p = 0.05 \times \frac{1.050}{2 \times 9.8} \times V^2 \times \frac{n (l + 3D)}{D} \approx 2.5 V^2 \frac{n (l + 3D)}{D} \text{ kg/m}^2 \quad (30.21)$$

If it is desired to express Δp in metres (feet) of water, the above values must be divided by 1,000. We have then:

$$J = 0.0025 n U^2 \left\{ \frac{l}{D} + 3 \right\} \quad \left\{ 0.00075 n U^2 \left\{ \frac{l}{D} + 3 \right\} \right\} \quad (30.22)$$

J = pressure drop caused by passage through the heater, in m (ft.) of water

- V = velocity of juice in the tubes, in m/s (ft./s)
 n = number of passes in the heater
 l = length of each tube, in m (ft.)
 D = inside diameter of the tubes, in m (ft.).

Temperature margin

Practical application of juice-heater calculations shows that, if excessive values of heating surface are to be avoided, it is desirable to arrange for a certain margin between the temperature T of the heating vapour and the temperature t required for the heated juice leaving the heater.

With the object of economy, one should strive to limit the temperature t required, in such a way as to maintain the margins of temperature given in Table 30.1.

TABLE 30.1

MARGIN OF TEMPERATURE TO BE ALLOWED IN JUICE HEATERS		
Heating medium	Temperature margin $T - t$	
	(°C)	(°F)
Exhaust steam	5 - 8	10 - 15
Vapour from 1st vessel	10 - 12	18-22
Vapour from other vessels	15-20	27-36

Otherwise, the excessive heating surface which will be necessary to obtain a hotter juice would be out of proportion to the gain in temperature so obtained.

Juice heating is generally done in stages, at least in the main battery of heaters, by taking vapours from the various vessels of the multiple effects in turn, and finishing with exhaust steam; thus a battery will be obtained having a reasonable number of heaters of optimal heating surface.

DESIGN OF A BATTERY OF HEATERS

Data. Suppose:

Crushing rate of the factory:	100 t.c.h.
Weight of mixed juice produced:	100% on cane
Density of mixed juice:	1.05
Temperature of cold juice:	30°C (86°F)
Back pressure:	0.5 kg/cm ² (7 p.s.i.)
Quadruple effect: 4 equal vessels each	720 m ² (7,750 sq.ft.)

We shall assume that this quadruple effect operates under the following scale of temperatures:

	Temp. T		Latent heat r	
	(°C)	(°F)	(kcal/kg)	(B.T.U./lb.)
Steam to 1st effect	111	232	532	957
Vapour from 1st effect	102	216	538	967
Vapour from 2nd effect	92	198	544	979
Vapour from 3rd effect	80	176	551	992
Vapour from 4th effect	55	131	566	1,019

and that the evaporation capacity of each effect is according to the following scale (*cf.* p. 580):

	H.S.		Evap. rate		Total evap.	
	(m ²)	(sq.ft.)	kg/m ² /h	(lb./sq.ft./h)	(kg/h)	(lb.h)
1st effect	720	7,750	38	7.8	27,360	60,450
2nd effect	720	7,750	32	6.6	23,040	51,150
3rd effect	720	7,750	24	4.9	17,280	37,975
4th effect	720	7,750	22	4.5	15,840	34,875

Choice of heating stages

We shall strive to utilise to the best advantage the possibilities of each vessel by bleeding from each the quantity of vapour which it can supply above that necessary for the following vessel. We shall assume that no vapour bleeding is used for pans.

However, we shall not make use of the excess of the 3rd effect over the 4th, which is not worth the trouble. We shall suppose in this case that the last 2 vessels equalise and each furnish 16,500 kg/h (36,400 lb./h).

Heater No. 1. The surplus of the 2nd effect is therefore:

$$23,040 - 16,500 = 6,540 \text{ kg/h (14,750 lb./h)}$$

Using this in a heater receiving the cold juice, we may heat this juice to a temperature t_1 such that:

$$6,540 = \frac{100 \times 1,000 \times 0.9 (t_1 - 30)}{544 \times 0.95}$$

whence:

$$t_1 = 67^\circ\text{C (154}^\circ\text{F)}$$

Heater No. 2. In its turn, the 1st effect has a surplus of $27,360 - 23,040 = 4,320$ kg/h (9,300 lb./h) and could thus heat the juice leaving No. 1 heater to a temperature t_2 such that:

$$4,320 = \frac{100 \times 1,000 \times 0.9 (t_2 - 67)}{538 \times 0.95}$$

whence:

$$t_2 = 91^\circ\text{C} (196^\circ\text{F})$$

Heater No. 3. It remains then to raise the juice from 91°C to $103 - 104^\circ\text{C}$ ($218 - 220^\circ\text{F}$) by means of exhaust steam.

Calculation of heating surfaces

We may anticipate that the required heaters will be found for example in a series of 75 m^2 (807 sq.ft.) with 14 passes each of 13 stainless-steel tubes of $35.6 \times 38 \text{ mm}$.

The volume of juice to be heated is:

$$Q = \frac{100 \times 1,000}{1.05} = 95,240 \text{ l/h} (3,420 \text{ cu.ft./h})$$

As these heaters give an output of $46,600 \text{ l/h}$ for a juice velocity of 1 m/s (or 502 cu.ft./h for 1 ft./s), the velocity for an output of $95,240 \text{ l/h}$ of juice will be:

$$V = \frac{95,240}{46,600} = 2.04 \text{ m/s} (6.8 \text{ ft./s})$$

This is a very high velocity, but is acceptable if no increase in tonnage is planned for the factory. It is desirable to seek velocities of the order of $1.6 - 1.9 \text{ m/s}$, which will leave a slightly greater margin from this point of view. The principal difficulty with a velocity of 2 m/s (6.5 ft./s) lies in the pressure drop, which necessitates the interposition of a second pump, after the first two or three heaters, in order to avoid bursting of the bottom doors by excess pressure.

No. 1 heater from second effect. The heat-transfer coefficient of the first heater will be (eqn. (30.11)):

$$k_1 = 6T \left(\frac{U}{1.87} \right)^{0.8} = 6 \times 92 \times 1.333^{0.8} = 610 \text{ kcal/m}^2/\text{C/h} (125 \text{ B.T.U./sq.ft./}^\circ\text{F/h})$$

and its heating surface (eqn. (30.5)):

$$S_1 = \frac{pC}{k_1} \ln \frac{T - t_0}{T - t_1} = \frac{100 \times 1,000 \times 0.9}{610} \ln \frac{92 - 30}{92 - 67}$$

or:

$$S_1 = \frac{1,000 \times 100 \times 0.9}{610} \times \frac{62}{25} = 134 \text{ m}^2 (1,484 \text{ sq.ft.})$$

We shall choose two heaters each of 75 m² giving 150 m², which will give a good margin of safety.

Normally, we shall thus obtain, instead of the temperature of 67°C (154°F) sought, a temperature t_1 (eqn. (30.4)) given by:

$$t_1 = 92 - (92 - 30) e^{-n}$$

where:

$$n = \frac{610 \times 150}{90,000} = 1.01667$$

$$\log e^{-n} = -0.4343 \times 1.01667 = -0.44146$$

Hence:

$$e^{-n} = 0.3618$$

$$t_1 = 92 - (62 \times 0.3618) = 92 - 22 = 70^\circ\text{C} (158^\circ\text{F})$$

In fact, it is wise not to count on the margin of safety thus calculated, but to reckon only on the 67°C (154°F) originally adopted.

No. 2 heater from first vessel. We have in the same way:

$$k_2 = 6 \times 102 \times \left(\frac{2.04}{1.8}\right)^{0.8} = 676 \text{ kcal/m}^2/^\circ\text{C/h} \quad (138 \text{ B.T.U./sq.ft./}^\circ\text{F/h})$$

$$S_2 = \frac{90,000}{676} \ln \frac{102 - 67}{102 - 91} = 154 \text{ m}^2 (1,600 \text{ sq.ft.})$$

We shall take again two heaters each of 75 m², identical to the two preceding ones.

No. 3 heater using exhaust steam. We have in the same way:

$$k_3 = 6 \times 111 \times 1.105 = 736 \text{ kcal/m}^2/^\circ\text{C/h} \quad (155 \text{ B.T.U./sq.ft./}^\circ\text{F/h})$$

$$S_3 = \frac{90,000}{736} \ln \frac{111 - 91}{111 - 103} = 112 \text{ m}^2 (1,229 \text{ sq.ft.})$$

We shall take again, to complete the standardisation, two heaters each of 75 m². It is likely

that one of these two will suffice on Monday and Tuesday, to attain a temperature of 102°C (216°F), the other on Wednesday and Thursday, and the two together to finish the week. We shall thus have a large and homogeneous battery, which is valuable for provision of replacement tubes.

Comments. (7) In each case we could, instead of starting again for each stage from the temperature chosen for the previous stage, calculate the temperature which the heating surface chosen will give, and commence with that temperature as *to*. The degree of accuracy of the heater calculations does not justify this.

(2) Our eqn. (30.11) corresponds to a conservative average. Even at the end of the week, better heat-transfer coefficients are sometimes obtained. If these are sufficiently assured, the second term in this equation could be multiplied by an appropriate factor, such as 1.1 or 1.2.

(3) To avoid risk of depriving the following evaporator vessel of part of the vapour which it requires, it will be necessary either to avoid providing too great a margin in the choice of heating surfaces, or to throttle the vapour valve so as not to exceed the set temperature.

(4) When the back pressure is sufficient (above about 0.7 kg/cm² (10 p.s.i.)), a temperature of 103°C (218°F) may be attained with vapour from the first vessel and thus exhaust steam will not be required. It is thus advisable to provide for the last heater an exhaust-steam connection regulated by a controller of Schlumberger, Amri, Masoneilan or similar type, admitting exhaust steam when the juice temperature falls below 101 or 102°C (214 or 216°F).

(5) Some heaters may advantageously be provided with two vapour connections; second and first effects, or first effect and exhaust, for example. At the beginning of the week, the vapour of lower temperature may be used; when the heater becomes fouled and does not give the required temperature, a change will be made to the higher-temperature vapour.

Final temperature

A temperature of 105°C (222°F) leaving the final heater should not be exceeded. Webre⁹ considers that the higher the juice temperature, the greater is the risk that waxes, molten at such temperatures, will be emulsified by the ebullition occurring in the flash tank ahead of the clarifier. They then become very difficult to remove.

Definition of heating surface

The heating surface of a juice heater refers to the *inside* area of the tubes. This convention is not universal, but it is general in France, and it is logical, since it is the coefficient of transfer from tube to juice that is the limiting factor; transmission from vapour to the outer surface of the tube is much more rapid. It is therefore the area of the boundary between tube and juice which determines the capacity of the heater, and it is certainly that surface which is the best measure of capacity.

The area of tube plate between the tubes is neglected. However, certain manufacturers, who neglect this, take it into account in some measure by calculating the area from the overall length of the tubes, between the outer faces of the tube plates. This is the system adopted in particular by French manufacturers.

English manufacturers calculate the heating surface from the outside of the tubes (Perk¹⁰). American manufacturers (Meade¹¹) follow a similar practice. It is as well to be aware of this; with tubes of 31 x 35 mm, the difference between the two areas exceeds 11%.

Total heating surface required

What is the total heating surface required in an ordinary cane sugar factory, using defecation or sulphitation?

Deerr¹² specifies 4 m²/t.c.h. (45 sq.ft./t.c.h.), Tromp¹³ 3-4 m²/t.c.h. (35-45 sq.ft./t.c.h.).

In South Africa¹⁴ installations vary from 2 to 9 mVt.c.h. (24 – 100 sq.ft./t.c.h.), 4 being considered a standard figure, but more recently¹⁵ increased to 8 mVt.c.h. (84 sq.ft./t.c.h.) including the reheaters before the evaporators.

In Cuba¹⁶ 3.5 mVt.c.h. (38 sq.ft./t.c.h.) was recommended for a juice velocity of 1.8 m/s (6 ft./s).

In Puerto Rico¹⁷ the mean for all factories for the 1948 campaign gave 3.3 m² (36 sq.ft.), the extreme figures being 2 and 5.1 mVt.c.h. (22 and 56 sq.ft./t.c.h.).

If heating is to be done in stages, with vapour bleeding from at least 2 effects, it is necessary to reckon on:

4.5 – 6 m²/t.c.h. (50-65 sq.ft./t.c.h.) for the normal battery

1 – 1.5 mVt.c.h. (10- 15 sq.ft./t.c.h.) more, in the case of fractional liming and double heating.

Addition of a heater-condenser would increase these figures appreciably.

CONSTRUCTION OF HEATERS

The cylindrical shell containing the tube plates is extended at each end beyond the tube plate, the extended portion being divided into compartments by baffles.

Except for the first compartment, by which the juice enters, and the last, or outlet, both of which are located in the top upper recess for vertical heaters, each compartment provides for 2 passes: upward and downward. If there are 10 tubes per pass, for example, there will be 20 tubes for each compartment, 10 for upward and 10 for downward flow. We give (Fig. 30.2) a view from above of the top compartment and of the bottom cover, showing the mode of circulation.

The shell is generally of mild steel plate. The extensions at both ends were previously of cast iron, but are much stronger if made of cast steel. The doors also should be of steel, if they are to withstand the pressures produced by the pressure drops corresponding to high velocities in long batteries of heaters.

Tubes

When the tubes reach or exceed 4 m (13 ft.) in length, there is a risk of trouble due to the difference in expansion between tubes and shell, since the former are always hotter than the latter; it is then advisable to construct the shell in two portions separated by an expansion joint. Certain manufacturers go as far as 4.5 m (15 ft.) without a joint, but it is then found that the tubes become bent and wear more rapidly with cleaning.

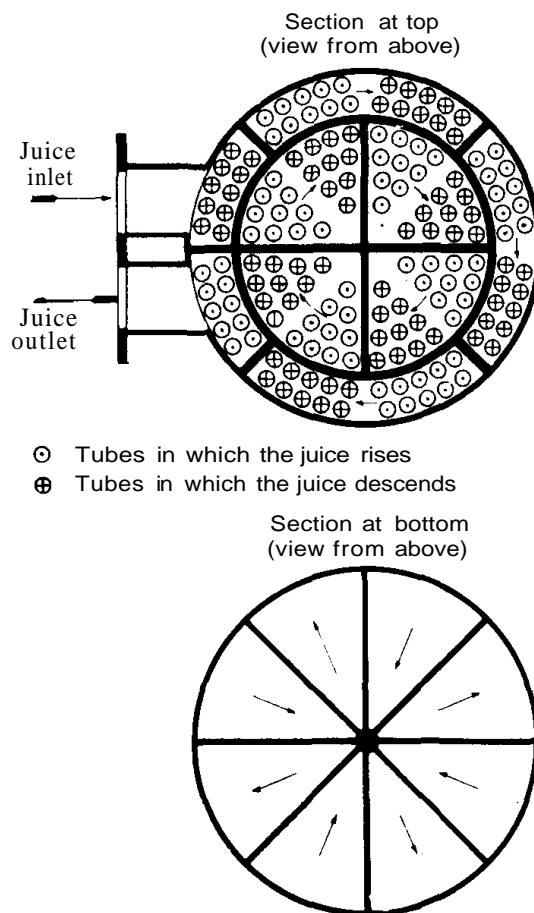


Fig. 30.2. Juice circulation in a heater.

TABLE 30.2

SERIES OF HEATERS (FIVES CAIL - BABCOCK)

Series 1

Short Vertical Heaters

12 Passes: Length of Tubes 3.9 m (35.6 x 38.1 mm)

	Heating surface (m ²)							
	40	50	60	75	100	120	150	200
Number of tubes	96	120	144	180	240	288	360	480
Tubes per pass	8	10	12	15	20	24	30	40
Ext. dia. <i>D</i> (mm)	740	780	850	890	1020	1100	1220	1350
Tube cross-section (dm ²)	0.796	0.995	1.194	1.493	1.991	3.389	2.986	3.981
Juice flow (m ³ /h) (<i>v</i> = 1 m/s)	28.7	35.8	43	53.7	71.7	86	107.5	143.3

Series 2
Long Vertical Heaters
8 Passes: Length of Tubes 6 m (35.6 x 38.1 mm)

	<i>Heating surface (m²)</i>					
	<i>150</i>	<i>200</i>	<i>250</i>	<i>300</i>	<i>350</i>	<i>400</i>
Number of tubes	240	320	392	472	552	632
Tubes per pass	30	40	49	59	69	79
Ext. dia. <i>D</i> (mm)	1071	1196	1326	1401	1511	1586
Tube cross-section (dm ²)	2.986	3.982	4.877	5.873	6.868	7.863
Juice flow (m ³ /h) (<i>v</i> = 1 m/s)	107.5	143.3	175.6	211.4	247.3	283.1
Factory capacity (t.c./day)	3,000	4,000	5,000	6,000	7,000	8,000

Fives Cail – Babcock did not exceed 3.9 m in tube length, but now make heaters with 6-m (20-ft.) tubes (Table 30.2) without apparent trouble due to expansion. The same long tube heaters are supplemented by a "sweeper" series, in which the juice travels from top to bottom and bottom to top in concentric annular spaces, thus assuring effective vapour – juice circulation and reducing the loss due to incondensable gases.

The tube plates should preferably be of the same metal as the tubes, to avoid electrolytic effects.

The tubes were previously in brass, the composition of this metal being for example: 70% Cu, 29% Zn, 1% Sn. In France, brass designated U-Z 36 = 64% Cu, 36% Zn was often used. The tubes are now generally made in stainless steel type 430 or A151304, which gives the longest life. The commonest diameters in brass are:

32 X 35 mm	35 X 38 mm	42 x 45 mm	46 x 50 mm
(1 $\frac{1}{4}$ X 1 $\frac{3}{8}$ in.)	(1 $\frac{3}{8}$ x 1 $\frac{1}{2}$ in.)	(1 $\frac{5}{8}$ x 1 $\frac{3}{4}$ in.)	(1 $\frac{13}{16}$ x 2 in.)

and for stainless steel, generally 35 or 35.6 x 38.1 mm (1 $\frac{1}{2}$ x 1 $\frac{1}{2}$ in.).

It is not logical to retain different diameters so close together and it would be desirable to adopt a single standard diameter, such as 35.6 x 38.1 mm (1 $\frac{3}{8}$ x 1 $\frac{1}{2}$ in.), as recommended by Fives Cail – Babcock.

For this exterior diameter of 38.1 mm (1 $\frac{1}{2}$ in.), the thicknesses recommended are:

Brass:	1.6 mm (0.063 in.)
Copper:	1.6 mm (0.063 in.)
Steel:	2.0 mm (0.079 in.)
Stainless steel:	1.2 mm (0.047 in.)

Each heater should be furnished with two thermometers, designed for easy and immediate reading, giving the entry and exit temperatures of the juice.

While tubes of stainless steel are to be recommended for a new heater, they are not indicated for replacing a set of tubes in a used heater; they are less adaptable to being expanded

into tube plates which are somewhat worn. The replacement tubes should be of brass or copper, and should all be replaced at the one time.

Horizontal and vertical heaters

The British practice is to build heaters with the axis horizontal, while the French use vertical types. The latter arrangement generally allows the heaters to be accommodated and grouped more readily. On the other hand, horizontal heaters are considered to have a higher heat-transfer coefficient: Buchanan¹⁸ estimates this difference at 13-28%, according to the degree of cleanliness of the heating surfaces (28% for clean surfaces, 13% for scaled surfaces), or about 20% as an average. Heaters of diameter greater than 1.2 m (4 ft.) are generally arranged horizontally.

The type of heater adopted very generally in Australia is somewhat different from the normal type as shown in Fig. 30.1. These are heavy horizontal cylindrical heaters without compartments; the juice passes through several tubes in parallel, and flow is repeatedly reversed by headers fixed to the end tube plates, which take the place of end covers in the more common type of heater. The heaters are normally not opened for cleaning, cleaning being effected by chemical means, with circulation through the tubes of a 7% solution of caustic soda.

Incondensable gases. Heaters using exhaust steam are generally provided with a simple incondensable-gas pipe discharging to atmosphere, and it is sufficient to leave this just "cracked" open.

Heaters working on bled vapour, on the other hand, demand a generously designed incondensable-gas pipe. The withdrawal should be made with a drop of one stage of pressure when the heater is close to the evaporator (the incondensables from a heater using 1st-effect vapour should be taken to the top of the 2nd effect), but of 2 stages if the heater is at a distance (1st to 3rd effect).

The incondensables should be withdrawn from the top as well as from the bottom of the shell. The withdrawal pipe serving the bottom of the heater should terminate 10 cm (4 in.) from the bottom, in order to avoid picking up condensate.

The incondensable-gas pipe should have a cross-section of at least 1 cm² per 10 m² (1 sq.in. per 700 sq.ft.) of heating surface.

Condensates. Condensate outlets from the heater should be sufficient to ensure that the velocity of flow of the water does not exceed 1 m/s (3 ft./s).

Vapour pipes. The steam and vapour pipes should be so designed that the velocity of the vapour does not exceed 30 m/s (100 ft./s).

The vapour entry should be placed about one-quarter of the length down from the top of the heater (in the case of a vertical heater). This arrangement avoids excessive vibration and breakages of tubes, and facilitates escape of condensate along the tubes (Perk¹⁹).

In long tube heaters, the shock of vapour entry to the tubes may be avoided, and steam distribution improved, by the installation of a skirt in the heater, surrounding the tubes. This cylindrical skirt is more effective if placed eccentric to the body of the heater, forming a space

proportional to the quantity of steam remaining: *i.e.* maximum space near the entry and minimum at the opposite side of the heater.

Pressure test. Heaters are tested, according to the intended vapour pressure:

Vapour side: at 5 bars (73 p.s.i.)

Juice side: at 9 bars (130 p.s.i.).

The permissible working pressure is 2.5 bars on vapour side, 6 bars on juice side.

Pressure on bottom doors

The pressure on the lower doors and bottom portions of the heaters is calculated as follows: we take the delivery head of juice from the heater outlet to the final juice-discharge level; we add the height of juice in the heater, the loss of head in the heaters following it; and it is assumed that the pressure due to the loss of head in the heater under consideration is equal to half the total loss of head for that heater.

Generally the difference in density between juice and water is neglected, and the height of juice above the bottom is taken as equal to the height of the tubes.

Example. To calculate the pressure acting on the bottom of the first heater (obviously the heaviest loaded) of a battery of 3 heaters of 12 passes each of 19 tubes of 31 X 35 mm ($1\frac{1}{4}$ x $1\frac{3}{8}$ in.) heating 100,000 l/h (3,600 cu.ft./h) of juice.

Height of juice discharge above the outlet of the heaters:

$$h = 2 \text{ m (6 ft.)}$$

Length of tubes of each heater:

$$l = 3.7 \text{ m (12 ft.)}$$

Solution. Cross-section of juice passage in the heaters:

$$s = 19 \times \frac{\pi \times 0.31^2}{4} = 1.43 \text{ dm}^2 \text{ (0.1620 sq.ft.)}$$

Juice flow:

$$Q = \frac{100,000}{3,600} = 27.8 \text{ l/s (1.00 cu.ft./s)}$$

Velocity of juice in the heaters:

$$V = \frac{27.8}{1.43} = 19.4 \text{ dm/s} = 1.94 \text{ m/s (6.17 ft./s)}$$

Loss of head in each heater (eqn. (30.22)):

$$J = 0.0025 \times 12 \times 1.94^2 \left(\frac{3.7}{\dots} + 3 \right) = 13.80 \text{ m (40 ft.)}$$

Pressure acting on the bottom of the first heater:

	<i>m</i>	<i>ft.</i>
Head above heaters	2.0	1
Height of juice in heater:	3.70	12
Loss of head:		
In delivery pipe (estimated):	1.0	3
In the last two heaters = 13.8 x 2:	27.60	80
Mean in 1st. heater = 13.8 ÷ 2:	6.90	20
	<hr/>	<hr/>
	41.20	122

If it were desired to calculate the delivery pressure at the pump pumping the juice through the heaters, we should have:

	<i>m</i>	<i>ft.</i>
Discharge head: pump to heaters:	4.50	15
heaters to final level:	2.0	7
Losses of head:		
(a) in delivery pipes (pump to heaters + heaters to discharge):	1.50	5
(b) in three heaters (13.8 x 3):	41.40	120
	<hr/>	<hr/>
	49.40	147

It will be seen that high velocities of circulation lead to high pressures for the tube sizes normally employed by French manufacturers. The heaters and their pumps should be designed accordingly.

Heater-condenser

It will be seen in Chapter 32 (see p. 566) that the further advanced the vessel from which the vapour bleeding is done, the greater is the steam economy. If it is the vapour from the last effect that is utilised, economy will be complete, since this vapour would otherwise go to the condenser and be lost.

Further, by utilising this vapour, the load on the condenser is reduced by reducing the weight of vapour to be condensed.

Hence a heater is sometimes interposed, called a heater-condenser, in the vapour pipe between the last effect and the condenser.

This heater can work only on cold juice, since the temperature of the hot juice which it can deliver is limited by that of the vapour, that is, by the vapour temperature corresponding to the vacuum in the condenser (50 – 60°C; 120 – 140°F). It is difficult, in these conditions, to maintain an economic margin of temperature between vapour and hot juice, and this leads to large heating surfaces, and hence to an expensive unit. It will be necessary to balance its cost against the small gain in heat units to be expected from it. Similarly, the extra length of juice piping required, sometimes rather long and complicated, must be taken into account.

The heater-condenser is calculated as for an ordinary heater. When one is installed, it amounts in itself to about 3 m²/t.c.h. (32 sq.ft./t.c.h.).

Series of heater sizes

We gave in Table 30.2 data for the series of heaters supplied by Fives Cail – Babcock. It is obviously possible to fabricate heaters with tubes of any diameter desired, but the series indicated is recommended as standard.

All the heaters have similar tubes:

Outside diameter:	38 mm (1½ in.)
Total length:	3,900 mm (12 ft. 10 in.) and 6,000 mm (19 ft. 8 in.)
Effective length between plates:	3,835 mm (12 ft. 7 in.) and 5,935 mm (19 ft. 6 in.)
Thickness (stainless steel):	1.2 mm (0.047 in.)

The true heating surface, which depends on the thickness, will depend on the metal chosen for the tubes.

The output Q' in hl/h of juice for a juice velocity V' is deduced from the output Q given for a velocity of 1 m/s by multiplying the latter by V : $Q' = Q \times V'$.

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31. Filtration

The clarification process separates the juices treated into two layers or two parts:

- (a) The clear juice, which rises to the surface
- (b) The mud, which collects at the bottom.

The clear juice goes to manufacture, that is, in general, direct to the evaporators. The mud has first to be filtered, in order to separate from the juice the suspended matter which it contains, with the insoluble salts formed and the fine bagasse entrained with them.

CONDITIONS FOR GOOD FILTRATION

Filtration is sometimes a tricky operation, and difficult to control. To combine the best chances of carrying it out satisfactorily, it is necessary to observe several points:

(1) *Temperature.* The viscosity of the juices and especially that of the gums and waxes, which have to pass through the filtering surface, decreases as the temperature increases. It is therefore of advantage to filter the juice at a high temperature. Preferably, temperatures above 80°C (175°F) will be used.

(2) *Reaction.* Alkaline juices filter better than acid or neutral juices; hence it was general custom to add lime to muds before sending them to the filtration process, raising the pH to 8-8.5. This was necessary with filter presses, but is not done for rotary vacuum filters.

TYPES OF FILTER

The following types of filter have been used in the sugar factory:

Filter presses

Mechanical or leaf filters

Continuous rotary vacuum filters.

The first two types have disappeared from the defecation and sulphitation factories, to their benefit. However, filter presses still remain the standard system for carbonatation factories, for the filtration of juices after first and second carbonatations. We shall therefore give a brief review of them.

FILTER PRESSES

Description

A filter press (Fig. 31.1) consists of a series of plates (Fig. 31.2) separated by frames of the

same thickness, separate and movable, which slide on two supporting bars and can be held firmly against each other by means of a screw device or by hydraulic pressure.

When pressure has been applied and the filter is "closed up", the openings *a* and *b*, situated in the lugs of the plates and frames, form a continuous passage. The juice to be filtered enters by the passages *a*. Between each plate and the adjacent frame is interposed a filter cloth. The juice passes into the frame, traverses the cloth and leaves by the passage *c* (Fig. 31.3) after leaving behind on the cloth, in the interior of the frame, the suspended matter which it contained.

The solid mass formed by this suspended matter, when all the available space has been filled, is called the filter cake.

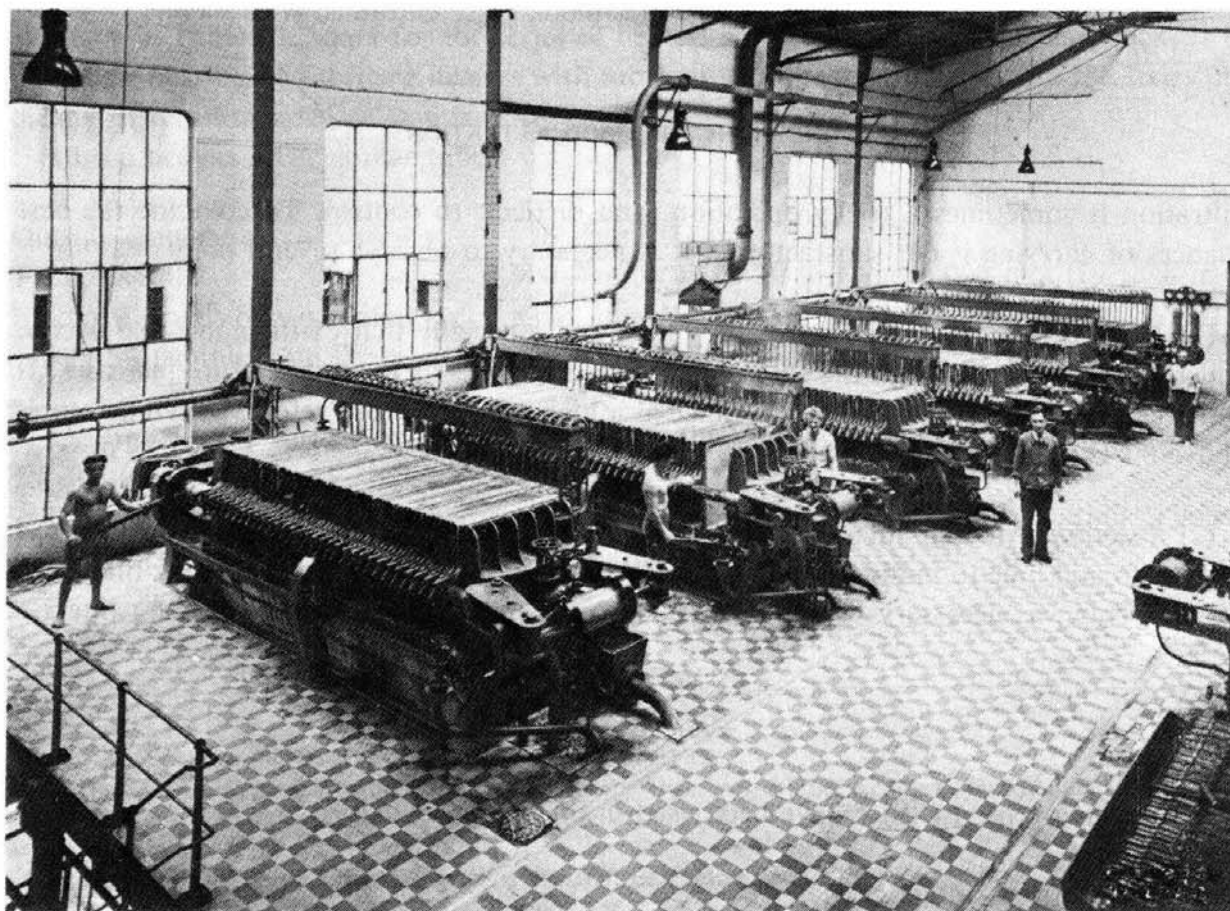


Fig. 31.1 Filter presses (Fives Cail - Babcock).

Plates. The plate consists of a machined border, and a thinner central portion, the surface of which is in the form of ridges or designs in relief, between which the juice can flow in spite of the pressure, which tends to force the cloth against the plate.

Frame. The frame has a similar machined border, but its interior is open.

The thickness of the frames and plates is of the order of 32 - 38 mm ($1\frac{1}{4}$ - $1\frac{1}{2}$ in.). The greater the thickness, the longer the time taken to form the cake.

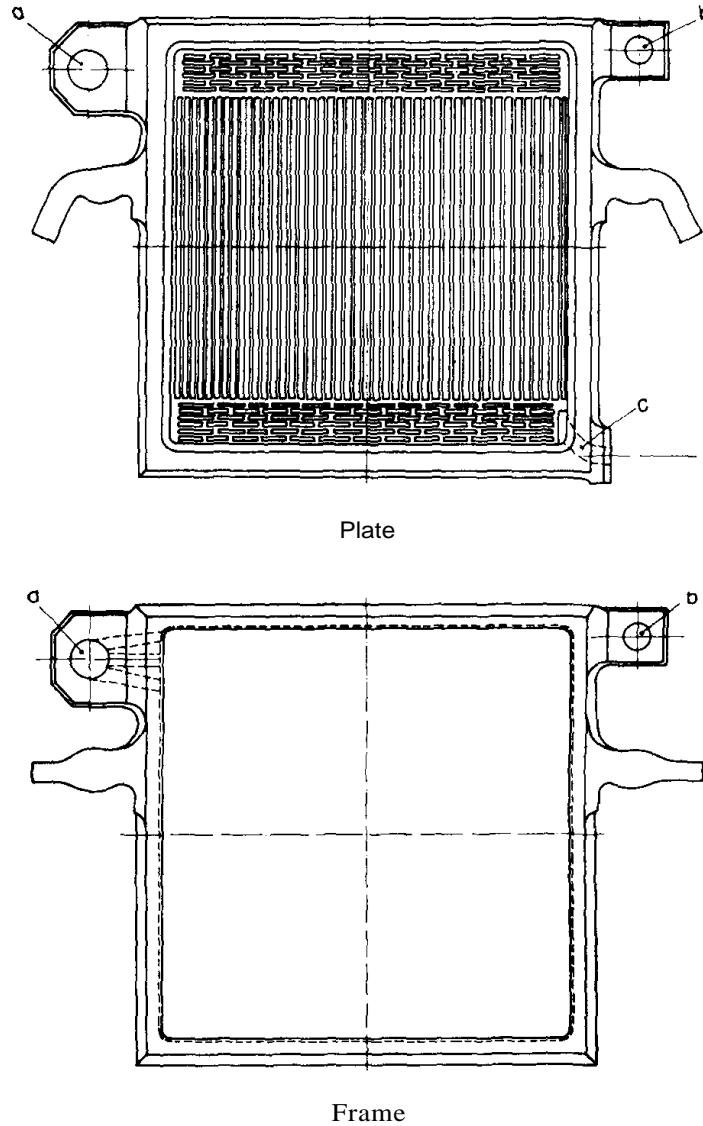


Fig. 31.2. Plate and frame of filter press.

Frames and plates are always of cast iron.

The number of movable plates, which is one less than the number of frames, is generally of the order of 40-45.

Plates and frames are supported by two strong bars of steel which also serve as cross-braces and absorb the tensile force produced between the two end members by the pressure exerted in closing the press.

The cocks fitted to each plate discharge into a longitudinal trough by which the clear filtered juice is removed.

Cycle

It requires about $\frac{1}{2}$ – 1h to open up, discharge, clean, re-assemble and close up a filter press.

Filtration proper lasts $1\frac{1}{2}$ – 2 h.

Washing with water and steam requires about $\frac{1}{2}$ h.

Hence the duration of the cycle:

Discharging and re-assembling:	$\frac{3}{4}$ h approx.
Filtering:	$1\frac{3}{4}$ h
Washing:	2 h
Total cycle:	<hr/> 3 h

Filtering area

The filtering area is equal to the useful area available for the passage of the dirty juice through the filter cloths.

For plate and frame filter presses, the filtering area is given by the expression:

$$S = 2NHL \quad (31.1)$$

S — filtering area of the press, in m^2 (sq.ft.)

N = number of frames in the press

H = interior height of the frame, in m (ft.)

L = interior width of the frame, in m (ft.).

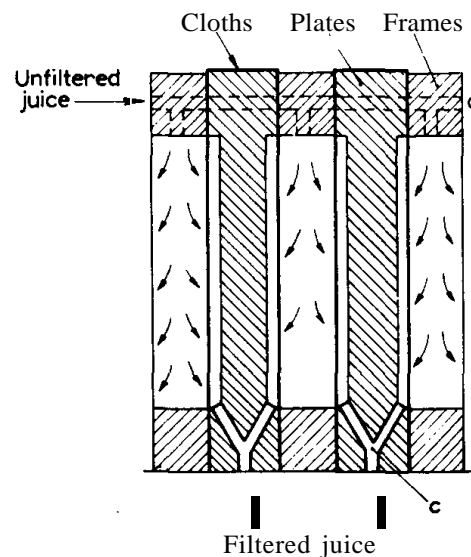


Fig. 31.3. Operation of filter press.

Normal dimensions

The most common size corresponds to a filter press of 40, 42 or 45 frames of 1,016 X 1,016 mm (40 x 40 in.).

These are the external dimensions of the frame. The corresponding filter area is about 915 x 915 mm (36 x 36 in.), say about 1.7 m^2 (18 sq.ft.) per frame, or 70 m^2 (750 sq.ft.) for one filter. The overall area of the plates, here about 1.8 m^2 (20 sq.ft.), has no meaning as regards filtration.

Filtration pressure

The speed of filtration depends on the pressure under which the juice is forced into the filter. This pressure is provided by the mud pump.

It is normally 2.8 – 3.5 kg/cm² (40 – 50 p.s.i.). It may fall as low as 2.1 kg/cm² (30 p.s.i.) in the most favourable cases, and may rise to 4.6 kg/cm² (65 p.s.i.), at the washing period, and in unfavourable cases.

Capacity

The filter area adequate for a factory per t.c.h. varies greatly according to the carbonatation process adopted and according as more or less complete washing of filter cake is adopted (cf. p. 474).

Deerr¹ and Tromp², respectively, specify the areas given in Table 31.1.

TABLE 31.1

FILTER AREA REQUIRED (PRESSES) PER t.C.h.

	<i>Deerr</i>		<i>Tromp</i>	
	(m ²)	(sq.ft.)	(m ²)	(sq.ft.)
1st carbonatation (de Haan)	8	90	6-8	70-85
1st carbonatation (ordinary)	10-13	110-135	9-11	100-120
2nd carbonatation	4-5	40-50	4-5	40-50
Filtration of syrup	3-4	30-40	4-5	40-50

Tromp advises adopting the lowest figures when kieselguhr is used but, if the quality of juice to be handled is not known, it will be prudent to increase these figures.

Filter cake

When filtration is good, the cake is dry and porous; when it is bad, the cake is semi-liquid and slimy; density approx. 1.3.

Water content. The water content of the cake obviously depends on its physical condition. If it is good, *i.e.* dry, it is about 50 – 60%; when it is slimy, it rises to 70%.

Sugar content. Before washing, the water content of the cake corresponds to its juice content. In the same way, its sugar content originates entirely from the juice; if it contains 70% of juice (say about 55 - 60% of water), its sugar content will be 70% of that of the juice from which it originates. Hence the advantage of filtering only muds derived from weak juices; hence also the advantage of substituting water for the juice in the cake, by washing, and by carrying the washing as far as possible.

Weight of cake. The weight of cake obtained in a carbonatation factory goes up to 6 - 12% on cane.

Loss of sugar in the cake. With good effective washing it is possible to reduce the loss of sugar in the cake to 1% on weight of cake or less, say 0.05–0.15% on cane or 0.5–1.2% of the sugar content of cane.

Washing of the cake

Pumping of mud to the filter press is stopped when the cake is completely formed, which is observed by the decrease in flow of clear juice from the drain cocks of the plates. However, better conditions for washing are obtained if it is arranged to stop slightly sooner than this, while there still remains a cavity in the cake.

At that moment, the juice which impregnates the cake and which forms more than half its weight, is the juice from the carbonatation, which generally contains 12–15% of sugar. An effort is made therefore to displace it, by driving it out with water or steam. This is what is called "washing".

It may be effected in two ways:

(a) **Simple washing.** Here water is simply substituted for juice: it follows the same path in the filter and leaves by the juice outlet cocks.

(b) **Systematic washing.** We have seen (Fig. 31.2) that, in addition to the juice passage *a* located in the rear lug of the plate, the frames and plates are provided with a passage *b* in the front lug.

Furthermore, the even and odd numbers of plates are differentiated by giving them alternately a short and a long outlet cock. This renders the handling of these cocks easy, which otherwise would become confusing.

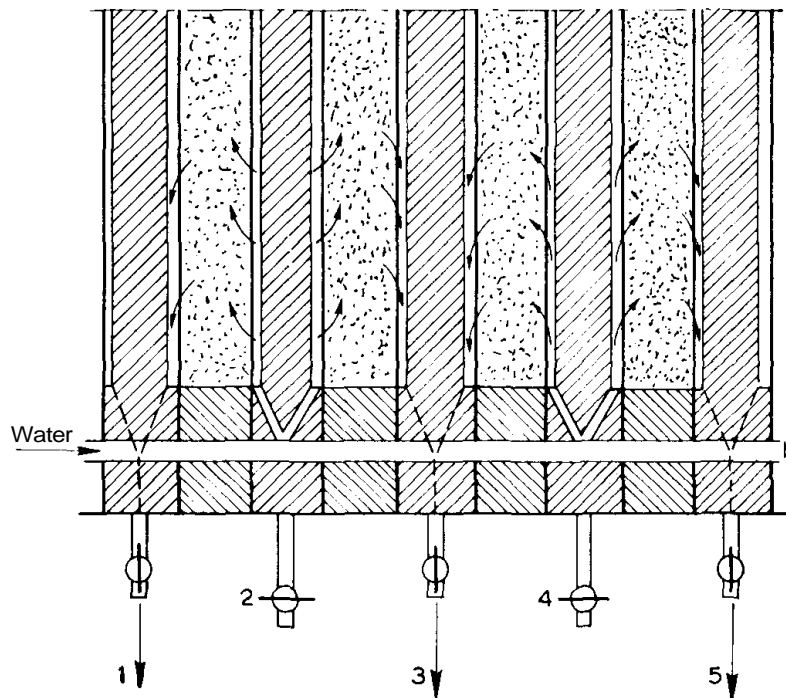


Fig. 31.4. Systematic washing.

The even plates communicate with the passage *b*.

Systematic washing is then conducted as follows: water is admitted by the passage *b*, and the clear-juice cocks on the even plates are closed (all the long cocks, for example). The water then enters by the even plates, traverses the cloths and the adjacent cakes, and leaves by the cocks of the odd plates, which are left open (Fig. 31.4).

This procedure gives better results than the preceding one.

French manufacturers furnish filter presses with the water passage *b* communicating with the interior of all the plates. This practice has the object of:

- (1) Permitting of simple washing by using the passage *b*
- (2) Maintaining interchangeability of the plates.

It is then necessary to block the small connecting passage into all the even plates, for example. It is of advantage to use for this a resinous cement, readily removed, rather than plastic metal.

Limit to washing. When at the finish of the filtration, the juice is replaced by water for the washing procedure, the brix of the clear juice flowing from the outlet cocks at first remains the same; this is the juice contained in the cake, which then flows out, displaced by the water. Then, the brix falls suddenly; this is the water coming through. According to the filtering area available, the washing is then stopped, at the moment when the brix falls to 3°, 2°, or 1°. At 3° brix there would be a high loss of sugar in the cake, but there is no advantage on the other hand in going lower than 1° brix; this would recover more impurities than sugar.

Since the wash water is alkaline, it is not mixed with the juice, and is preferably reserved for mixing the milk of lime.

After washing, the cycle is concluded by forcing out the water in the cake by passing steam into the filter.

External washing. The cake may also be washed outside the filter press. In this method, it is dropped from the press into a receiver, where it is thinned out with 90% of its weight of water, at 60°C (140°F). It is mixed well and re-filtered. The second filtration is rapid, and furnishes a cake containing not more than 1 – 2% of sugar.

Difficulties in filtration

If the filtration is difficult, and the cakes slimy, this could be due to the waxes contained in the juice which, solidifying towards 66°C (150°F), deposit on the cold cloths when the hot juice is introduced into the filter. To avoid this effect, the filter press may be put on steam before admitting juice.

If the cake remains slimy, the trouble could be ameliorated by increasing its content of fine bagasse, either by introducing fine bagasse, separated by a special screen, directly into the muds to be filtered, or by replacing the perforated screens of the cushion separator by screens with larger holes.

Filter aids

Kieselguhr may also be added. This is a siliceous deposit obtained from mines in North

Africa and in America, and derived from accumulated deposits of diatoms. The very fine structure of these minute creatures gives to their siliceous skeletons useful properties as filter aids.

The kieselguhr is mixed with the juice to be filtered. Tromp³ gives a figure of 0.43 kg/m^2 filtering surface/h ($0.09 \text{ lb./sq.ft./h}$).

Filter cloths

Filter cloths consist of fabric which may be cotton, linen and cotton, hemp and cotton, jute, or of nylon, the latter material having a much longer life.

They are supplied either in strips of the desired width cut at the factory in pieces of the desired length, or in separate cloths ready made up. Generally, the cloth is supplied with the gloss removed and guaranteed unshrinkable. However, it is of advantage to provide some excess of width and length. We give in Fig. 31.5 the best dimensions to be adopted for cloths intended for filter presses of $1 \text{ m} \times 1 \text{ m}$. The reinforcing bands shown give twice the thickness of cloth at the points where it is most liable to wear. These double its life without doubling its cost, and are to be recommended. They should never be done with lap joints, since each extra thickness causes leakages of juice at the point where it comes between plate and frame.

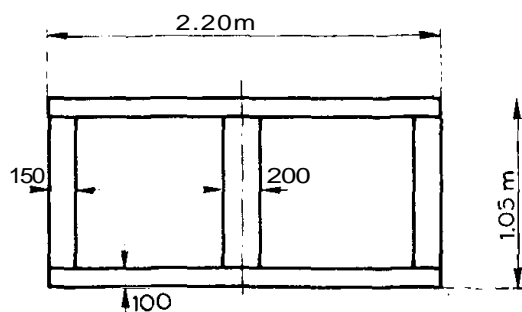


Fig. 31.5. Made-up cloth for filter press $1 \times 1 \text{ m}$ (unshrinkable).

For shrinkable cloths it is necessary to use dimensions of 2.25 and 1.07 times the side of the plate.

A filter cloth weighs about $0.8 - 1.0 \text{ kg/m}^2$ ($2.6 - 3.2 \text{ oz./sq.ft.}$) according to quality, not including reinforcements.

CONTINUOUS ROTARY VACUUM FILTERS

The typical and most widely used filter of this class is the Oliver - Campbell. It is this type which we shall describe.

Description

The filter is composed of a hollow drum rotating about a horizontal axis (Fig. 31.6) and partly submerged in the liquid to be filtered. The periphery of the drum serves as the filtering surface; it is divided into 24 independent sections, each occupying 15° on the circumference,

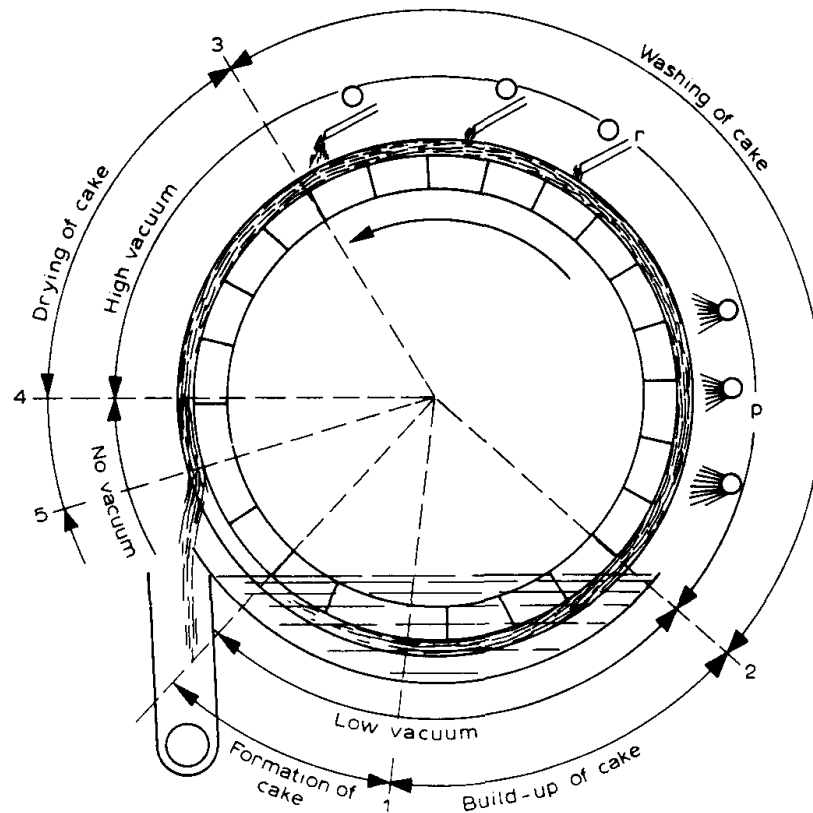


Fig. 31.6. Diagrammatic operation of a continuous rotary vacuum filter.

and extending along the full length of the drum. Each of these sections is connected individually to a vacuum system by a small metal pipe terminating in a distributing valve situated at one end of the drum and carrying three different sectors:

- (a) One without connection to the vacuum, but connected to the atmosphere
- (b) The second communicating with a chamber where a low vacuum, of the order of 15–30 cm (6–12 in.) of mercury, is maintained
- (c) The third connecting to a chamber where a higher vacuum is maintained, of the order of 40–50 cm (15–20 in.) of mercury.

The exterior surface of the drum consists of a sheet of copper or stainless steel with very fine perforations, covering the 24 sections (Figs. 31.7 and 31.8).

Operation

As the filter rotates, the section which first enters the liquid is immediately connected with the low vacuum. This produces an aspiration of the liquid, which passes through the fine perforations. However, the latter soon become coated with pieces of fine bagasse and suspended matter. The first juice which passes through is thus obviously badly filtered and very turbid; it is called "cloudy filtrate" and is directed to a special receiver.

Having now been coated with a thin layer which forms the beginning of the filter cake, the section next passes to the valve sector connecting to the higher vacuum. It is indeed necessary to increase the vacuum, since the resistance offered by the cake increases with its thickness. However, the juice which now passes through is more effectively filtered by its own impurities and by the fine bagasse which it contains, which thus themselves form the

necessary filtering surface and which collect on the surface of the drum. This filtrate then constitutes the "clear filtrate", and is sent to another receiver. The filter cake continues to build up, until the section concerned emerges from the liquid. The juice picked up at the surface flows back into the tank, while the juice contained in the body of the cake which has formed continues to be drawn through into the compartment of the drum.

The filter section then passes under several rows of atomisers p which spray hot water on the cake. This is the beginning of the washing. The vacuum draws the water through slowly, and the makers assert that this washing has been designed in such a way that the water has just sufficient time, on the fraction of the rotation remaining, to pass through the cake and to displace the juice. Actually the de-sugaring is very effective, and yet the brix of the clear juice is only very slightly lower than that of the cloudy filtrate.

After the sprays, the washing is continued by pipes and sheet-metal distributors r which allow water to drip on the cake, to balance the tendency of the surface to dry out as the water is drawn through it.

After the last drip pipe, drying commences: the water is drawn towards the inside without addition of water on the outside.

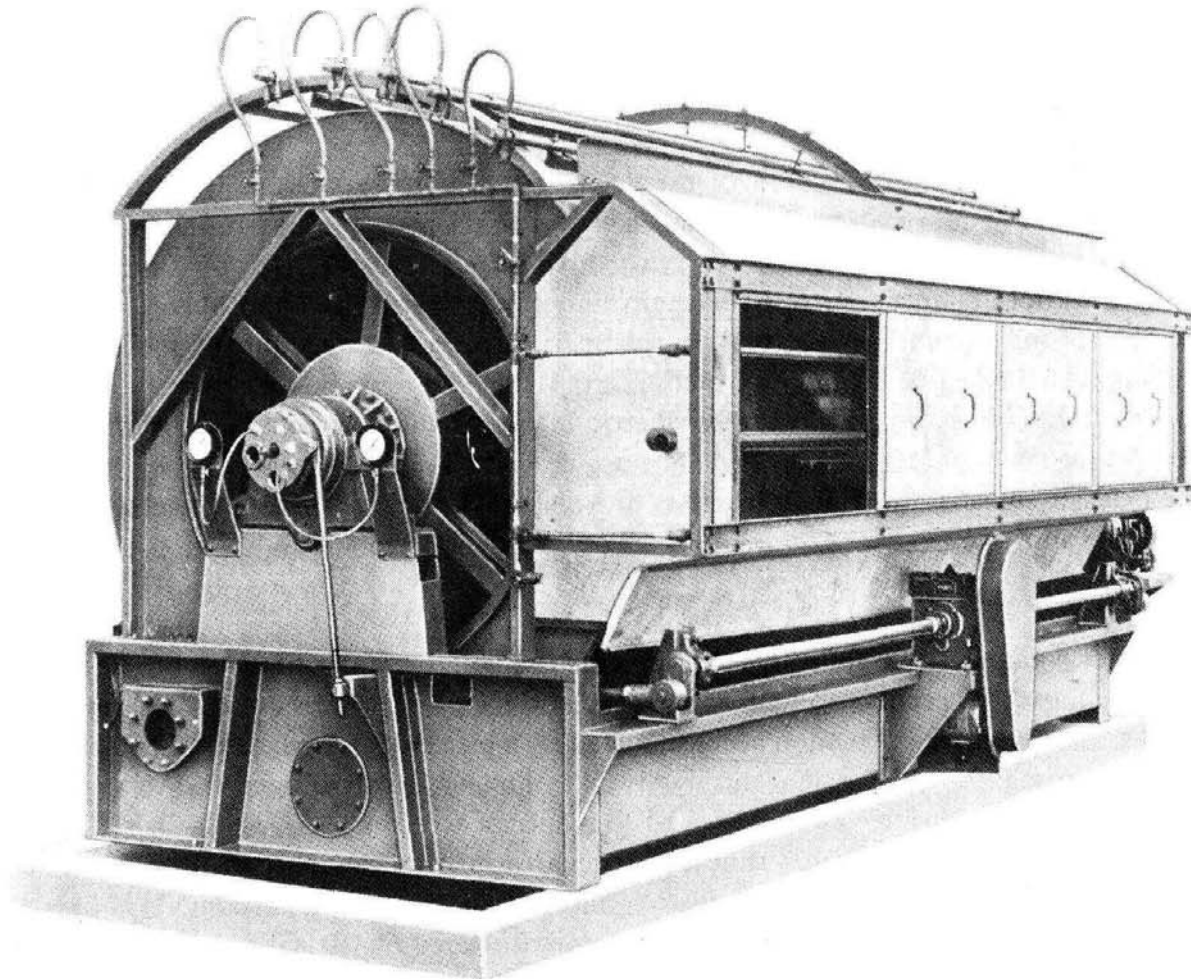


Fig. 31.7. Oliver - Campbell filter 2.4 X 4.9 m (8 x 16 ft.) (view showing sprays and drip pipes).

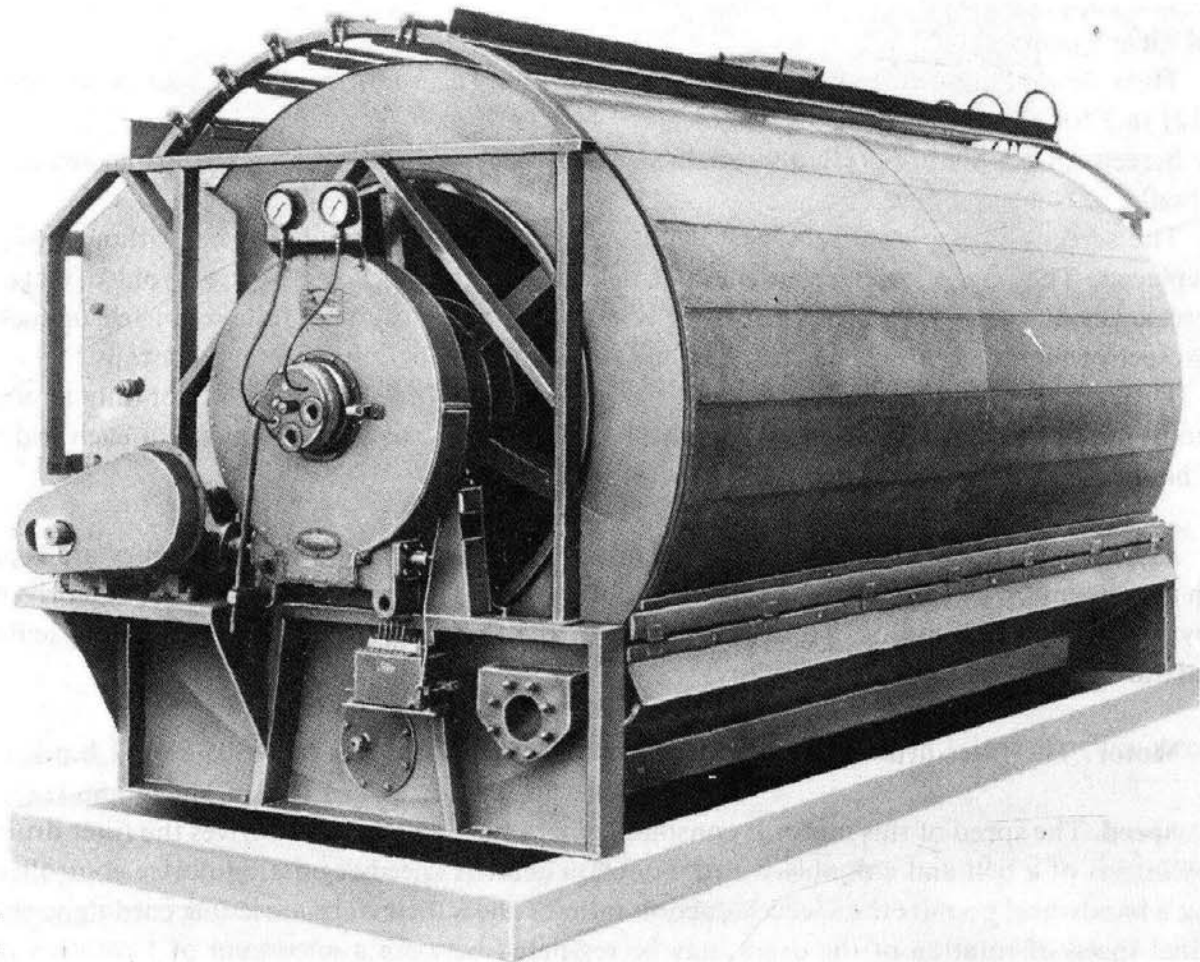


Fig. 31.8 Oliver – Campbell filter: view from cake discharge side. On the distributing valve are seen the regulating valves for the two suction pipes for low and high vacuum.

Just before the section re-enters the liquid in the tank, it arrives at a scraper formed by a rigid strip of rubber belting which bears lightly on the copper screen. When the filter section is about to reach the scraper, the distributor valve breaks the vacuum which has held the cake against the filter surface. Then the slightest contact of the scraper detaches the cake *en bloc*, and it drops into a screw or belt conveyor which removes it outside the building (Fig. 31.9).

Out in the yard, it is dropped into a trailer which, as soon as it is filled, is taken by tractor for spreading the filter cake on the fields, and the trailer is immediately replaced by another one.

Dry substance in muds. The proportion of solids in the muds going to the filtration should be not less than 4.5%. The recommended proportion is between 5.5 and 6%.

Perforated screen. The perforated screens are of copper, of brass or of stainless steel, perforated with 625 holes per sq.in., of 0.5 mm (0.020 in.) in diameter.

The copper screens are 0.45 mm in thickness, 1 mm between centres of holes, and 23% of clear opening.

Their designation in the French series is RD 0.5 S. The width of the sections is 324 mm ($12\frac{3}{4}$ in.) for the Oliver or 345 mm ($13\frac{9}{16}$ in.) for the Eimco filter.

Screens of copper are preferable to those in stainless steel; they are more flexible and less fragile.

The screens suffer very little wear, and may be used for several seasons without being replaced. They are, however, fragile and delicate. In the slack season, to avoid damage, breakages due to cyclones, etc., it will be wise to cover the filter with a layer of felt or such protective material, which can be fixed around the drum by circumferential straps.

The capacity and operation of the filter are improved by feeding the mud not only at one point but at three points along the length of the filter tank; at the middle and at each end.⁴ The distribution on the drum is then more uniform.

Agitator. In order to avoid allowing the muds in the filter tank to settle out, they are kept in movement by an agitator oscillating to and fro, pivoted on the axis of the filter, and driven by a separate small motor. To avoid rupture of the flocculent mud particles, the bagacillo mixture should be fed by gravity^{4a}.

Motor. The filter drum is driven by a small electric motor of about 1.5 kW (2 h.p.).

Speed. The speed of this motor is constant: generally, 1,450 r.p.m. It drives the filter drum by means of a belt and a double-worm reduction gear. A speed-regulation device controlled by a handwheel permits the speed reduction ratio of the V-belt to be varied; accordingly, the final speed of rotation of the drum may be regulated between a maximum of 1 rotation in 3 min and a minimum of 1 in about 10 min. Thus a scale of speed regulation is provided allowing the speed to be varied between extreme values in the ratio of approximately 1 to 3. It is estimated in Louisiana^{4b} that, by doubling the speed, the filtration rate is increased only by a factor of 1.4. On average, a speed of 1 revolution per 5 min is used.

Filter cake

The filter cake obtained appears dry and porous, but nevertheless contains, 75 – 80% of water.

On account of the bagacillo which is added to the mud to facilitate filtration, it is necessary to reckon on about 3-4 kg of filter cake per 100 kg of cane (specific gravity 1.3 – 1.4).

The thickness of the cake is variable, but generally of the order of 7 - 13 mm ($\frac{1}{4}$ – $\frac{1}{2}$ in.). It has been found in Louisiana^{4c} that optimal exhaustion of the cake was obtained with a thickness of 7 mm, since washing was most effective at that thickness.

The filter turns out 60-75 kg of filter cake per m²/h (12-15 lb./sq.ft./h), or more precisely, 125 kg/m²/h per cm (64 lb./sq.ft./h per inch) of cake thickness. In South Africa⁵, a rate of 75 – 80 kg/m²h (15 – 16 lb./sq.ft./h) is reported.

Temperature. The mud for filtration should be at least at 85°C (185°F) after addition of the fine bagasse. Below that temperature, there is a risk of waxes blocking the gauze. In this

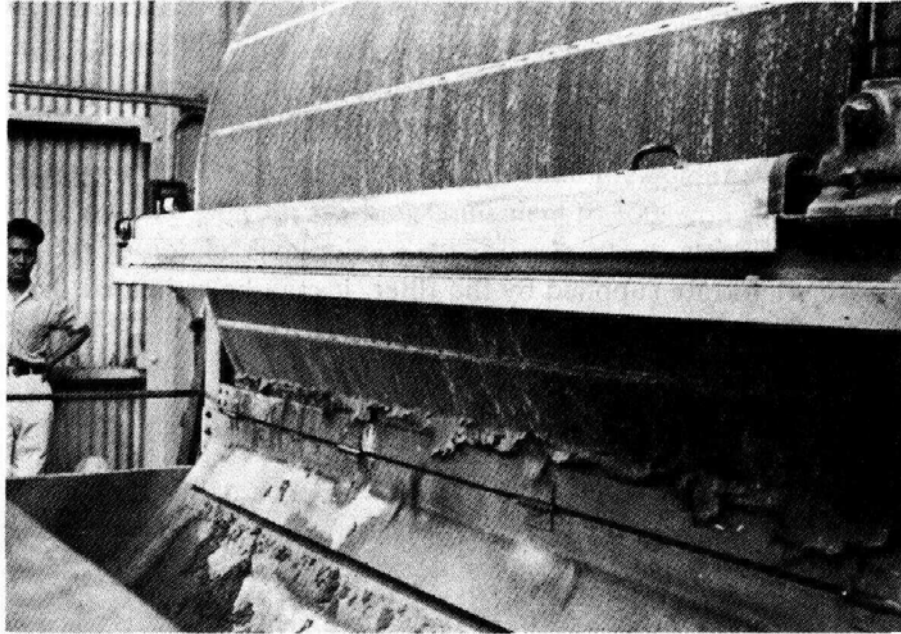


Fig. 31.9. Oliver – Campbel filter: cake discharge.

connection, overflow of mud from the filter tank should be avoided, since that involves recycling, which in turn causes cooling of the material.

Sugar loss in cake. The filter cake contains 0.5 – 3% of sugar, averaging 1 -2%. This corresponds to a pol loss of about 0.2 – 0.8%, averaging say 0.5%.

Filtration rate

A filtration rate of 250 – 400 l/m² (5 – 8 gal./sq.ft.) of filtering surface/h may be expected. However, the capacity of filtrate pumps is reckoned as 480 – 500 l/m²/h (10 gal./sq.ft./h).

Capacity

The filter area considered necessary varies in different countries. The following figures have been quoted:

	Average		Extremes	
	(m ² /t.c.h.)	(sq.ft./t.c.h.)	(m ² /t.c.h.)	(sq.ft./t.c.h.)
Cuba ^e	0.30	3.2		
Philippines ⁷	0.44	4.8		
Queensland ⁸	0.55	6.0		
Natal ⁵	0.64	7.4	0.33-1.0	3.6-11
Hawaii ^{1c}	0.83	9.1	0.67-1.0	7.4-11
Puerto Rico ^{1c}	0.59	6.5	0.20-2.2	2.2-24

Generally, an area of 0.6 m²/t.c.h. (6.5 sq.ft./t.c.h.) is specified in the American hemisphere.

We estimate that, even with juices difficult to treat, a capacity of 0.3 mVt.c.h. (3.3 sq.ft./t.c.h.) would be satisfactory.

With the “Rapifloc” system (*cf.* p. 485), Dorr–Oliver estimates that a figure of 0.35 mVt.c.h. (3.8 sq.ft./t.c.h.) will be sufficient instead of 0.6 (6.5). It is even possible to go as low as 0.25 mVt.c.h. (2.8 sq.ft./t.c.h.).

Proportion of cloudy filtrate

For every 100 parts of filtrate supplied by the filter, it may be estimated that there will be:

30-60% of cloudy filtrate
70 - 40% of clear filtrate.

These two filtrates together may contain between 2 and 12% of suspended matter expressed as dry material¹².

Disposal of filtrates

It is often difficult to decide what is the best way to deal with these two filtrates.

(a) For the cloudy filtrate, the first idea which comes to mind is to send it to the filter tank or to the mud tank preceding it. However, this dilutes the filter feed, and is not to be recommended. It is necessary to return the cloudy juice ahead of the clarifier. In compound clarification, there is a choice between returning it before the primary clarifier or before the secondary. The former solution is preferable.

(b) As regards the clear filtrate, unfortunately it has neither the brilliance nor the clarity of the clarified juice. It is greyish and still slightly cloudy. It would not be desirable to send it to manufacture with the primary clear juice. Accordingly, it is generally returned to process ahead of the clarifier.

In simple clarification and often in compound clarification, it is therefore common practice to mix the two filtrates and return the mixed filtrate to process ahead of the clarifier.

This is the disadvantage of this type of filter: it overloads the clarification station. When the latter is working near the limit of its capacity, it is undesirable to increase the volume of the juices, and consequently their speed of circulation in the clarifier, by a fraction which represents approximately:

Simple clarification:		12-20%
Compound' clarification	}	Primary clarifier: 10-20% Secondary clarifier: 10 - 15%

One solution consists of re-clarifying the two filtrates in a special small clarifier, after reheating and re-liming. A very clear juice is thus obtained, together with thick mud, which is returned to the rotary filter¹³. The filtrates may also be treated in a centrifugal separator, of the Westfalia type.

Washing

For the reason indicated in discussing the operation of this filter (*cf.* p. 477), the quantity

of wash water which passes into the clear filtrate represent only a small fraction of the water used at the filter, about 20–25%. The greater part remains in the filter cake. It is necessary therefore to determine the quantity of water to be used for washing according to the dilution of the clear filtrate, and not according to the quantity of water sent to the filter.

The efficiency of washing is of more importance than the quantity of water used. In South Africa, one factory using water for washing equivalent to 100% on filter cake obtained 0.3% pol in cake, whereas another employing half as much obtained 0.4%.

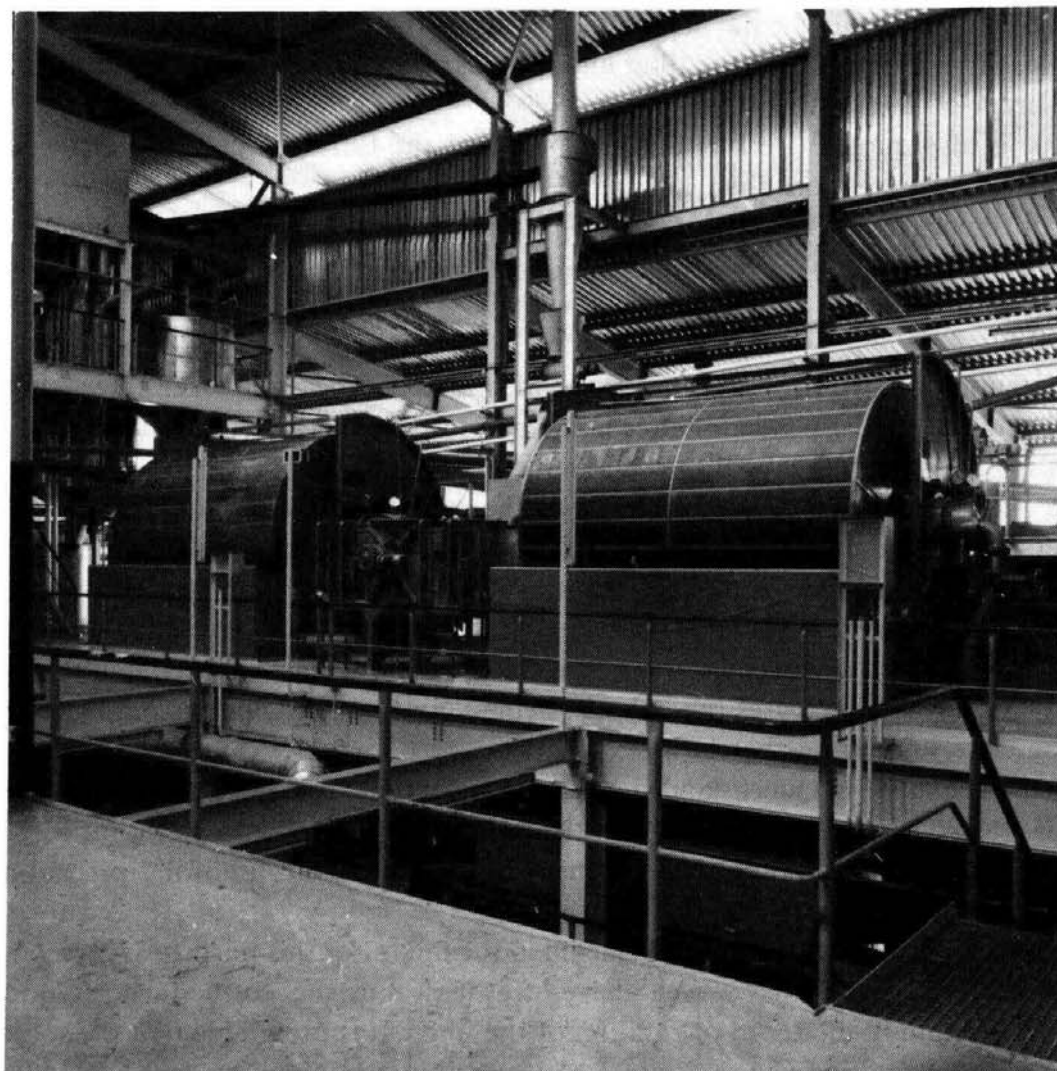


Fig. 31.10. Filtration station B.M.A. filters.

Generally, wash water amounting to 100–150% on cake is used. In Iran, wash water is reckoned as 1.38% on cane by weight. In general, one may assume 14–20 kg/t.c. (30–45 lb./t.c.), or 1.4–2.0%. One South African factory reports 12.6 kg/t.c. (28 lb./t.c.).

This would preferably be water at 75–80°C (167–176°F). However, Webre¹³, considers that it is desirable not to exceed 60°C (140°F) to avoid risk of entraining waxes contained in the filter cake. We do not share this point of view, for a reason completely analogous to that demonstrated by Honig for hot imbibition (*cf.* "Waxes", p. 308). In Louisiana, the

minimum temperature for wash water is stated as 82°C (180°F) and the optimal temperature as 88–90°C (190–194°F), at a pressure of 3–4 kg/m² (0.6–0.8 lb./sq.ft.). The small nozzles for water spray should be spaced about 30–40 cm (12–16 in.) apart, and about 16 cm (6 in.) from the surface of the filter. They should have an opening of 3–3.2 mm.

The washing could be continued so far as to reduce the polarisation to 0.2%, in compound clarification, or 0.3% in simple clarification; but it is doubtful whether this is desirable, since, in seeking such low figures, there is a risk of reducing the purity of the filtrate by washing to such an extent that the nett result is a loss of sugar.

Part of the soluble impurities are picked up during the washing, resulting in a drop in purity between the juice reaching the filter and the clear filtrate leaving it. This drop is generally of the order of 2–4 units.

A rule is also quoted (Perk¹⁴) that the purity of the filtrate should not differ from that of the clear juice by more than 1.5 units.

The difference in brix between these two filtrates, for a well-controlled washing, is between 30 and 40%.

Retention

The "retention" is the proportion of suspended matter extracted by the filter, expressed as a percentage of the suspended matter contained in the mud arriving at the filter.

This retention sometimes falls very low, below 50%.

Now the filter must extract the whole of the suspended matter supplied to it from the clarifiers, otherwise there would be an accumulation of suspended matter. Thus if the retention is only 50%, this means that the two filtrates maintain in circulation between clarifiers and filters a quantity of suspended matter equal to that which is removed with the filter cake. It is obvious that this involves a serious overload for the clarifiers, which greatly reduces their capacity, and which increases the risk of having very fine particles passing to manufacture with the clear juice from the clarifier.

Tests by Jenkins¹⁵ in Queensland showed that retention varied from 48 to 88%, averaging 70%. It was concluded from these tests that it was easy to attain a retention of 75–80%, representing a very substantial improvement over 50%, if the following precautions were observed:

(7) Increase in the proportion of bagacillo in the mud. For this purpose it is recommended that for the bagasse screen a woven gauze of at least 8 meshes to the inch should be used.

(2) A reduction in the speed of rotation of the filter. It is preferable that the speed should not rise above 1 rotation in 6 or 8 min, and if possible, the minimum speed should be maintained, which is of the order of 1 rotation in 10 or 11 min.

Later tests by Foster¹⁶ have shown that filter speed is not important, provided that the bagacillo content is sufficient; a high ratio of bagacillo to mud solids, and a high concentration of mud solids in the filter feed, are the main essentials for good retention.

It is considered in Mauritius (Houareau¹⁷) that the optimal concentration of insoluble material in muds is approximately 6%. If the concentration rises to 12%, the retention would fall to 60%; if it drops to 4%, the retention drops similarly. To obtain the optimal proportion of 6%, the bagacillo added should be increased to about 60% on the weight of insoluble matter, and retention can then exceed 80%.

Mud and scums. According to the method of harvesting and the state of the cane (straight stalks or mechanical harvesting), we may reckon 180-240 kg of mud per t.c. (400-540 lb./t.c.), giving 25 – 40 (55 – 90 lb.) kg of mud per tonne cane.

Dimensions

The normal diameter of the Oliver - Campbell filter drum is 8 ft. However, filters of 10 ft. are in use in Mexico, of 12 ft. in Australia and of 14 ft. in South Africa (Table 31.2).

TABLE 31.2

Diameter		Length (m)					
(m)	(ft.)	2.7 (9ft.)	3.7 (12ft.)	4.3 (14ft.)	4.9 (16ft.)	5.5 (18ft.)	6.1 (20ft.)
2.44	8	21 (226)	28 (302)	33 (352)	37 (402)		
3.05	10		35 (377)	41 (440)	47 (503)	52 (565)	58 (628)
3.66	12		42 (452)	49 (528)	56 (603)	63 (679)	70 (754)
4.27	14					74 (792)	

In new or expanding factories, the tendency is towards increased dimensions of the filters rather than increased number of filters. In Australia¹⁸ a filter of 240 m² (2,580 sq.ft.) has been installed.

“Rapifloc” filtration

In 1961, Dorr-Oliver patented a new system of filtration which it christened "Rapifloc". This system (Fig. 31.11) consists of adding a coagulant, “Separan AP 30”, manufactured

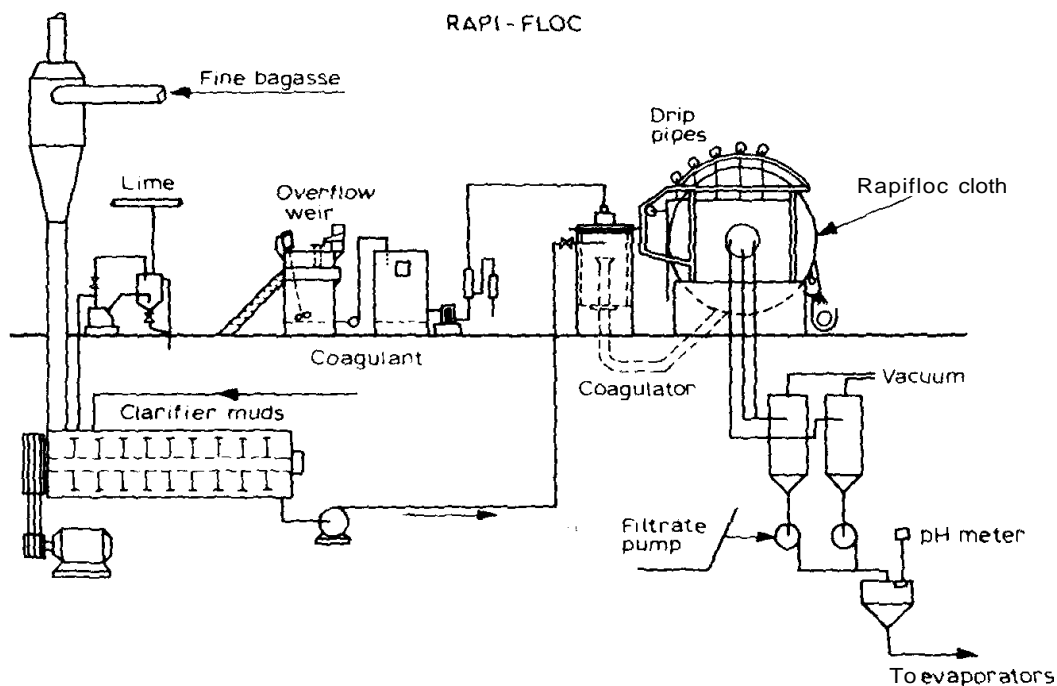


Fig. 31.11. Rapifloc system.

by Dow Chemical, Midland, Michigan, in the ratio of about 1.5 p.p.m. of juice. The coagulant is added in a special receiver, termed the coagulator, just before the mud goes to the filter. The latter may be an ordinary Oliver filter, which requires only minor modifications; it is covered with a porous cloth of felted polypropylene covering the perforated metal sheet, and is furnished with a new distributing valve and a device for washing the cloth between the position where the cake is detached and that where the drum re-enters the mud feed tank (Fig. 31.12).

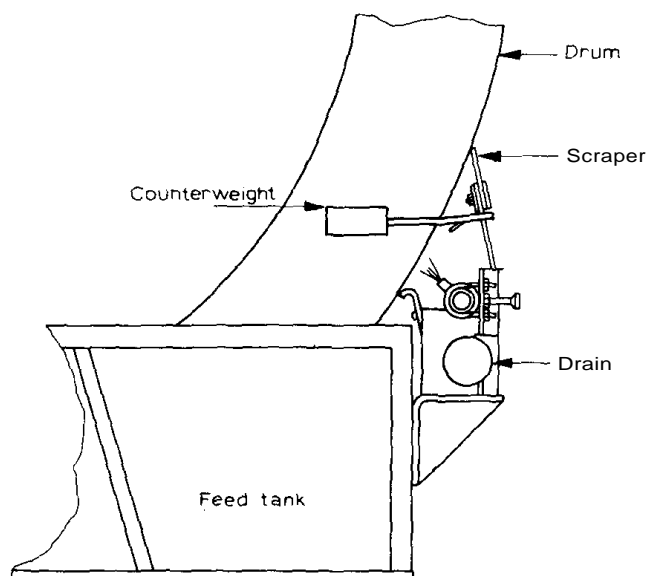


Fig. 31.12. Washing of cloth in Rapifloc filtration.

This new system gives a clear filtrate which may be sent directly to the evaporators, thus avoiding (1) overload of the clarification station and the filter due to recycling filtrates, (2) the long time which this portion of juice spends in these stations, with consequent losses by inversion.

The filter capacity is increased and, according to the manufacturers, almost doubled. It would be increased from about 100 to 170–195 kg/m²/h (35–40 lb./sq.ft./h) of cake at 80% moisture. The thickness of the filter cake is indeed greatly increased. From this point of view, the "Rapifloc" is clearly indicated when the increase in crushing rate of the factory is too much for the capacity of the existing filter plant, which will thus be sufficient with the new system. In South Africa¹⁹ a figure of 100-150 kg of filter cake per m²/h (20-30 lb./sq.ft./h) is assumed.

A certain gain in capacity may be obtained by replacing the normal perforated sheets (0.5 mm opening) by sheets with holes of 4–5 mm.

The water for washing increases the water requirement by approximately 2.3 l/s (0.5 gal./s); the brix of the filtrate is slightly lower, but the purity appreciably higher; a gain of about one point is obtained. Sucrose loss in the cake, however, shows hardly any decrease.

The life of the cloth is about 1–3 months for Dacron, 2–6 months for polypropylene.

The consumption of Separan may vary, according to conditions, from 1 to 5 p.p.m. Normally, 1.5–2 p.p.m. is sufficient.²⁰

Separan may also *be* replaced by the other flocculants cited (p. 404). In addition to propylene and dacron, a Belgian cloth (Lainyl) may also be used.

Difficulties. It happens at times that difficulty is experienced in stretching the cloth or in maintaining it tight on the drum; after operating for one or two days, it becomes loose and forms waves at the position of discharging the cake. A better tension will be obtained by keeping the filter under vacuum while placing the cloth in place; the cloth then stretches more readily and remains tight while the holding bands are secured.

To ensure that the washing is effective and does not lead to a choking of the cloth, it is necessary that the wash water should be at 80-90°C (175 – 195°F), under a pressure of 4 kg/cm² (55 p.s.i.), and should strike the drum at an angle of about 110° (that is, in a plane inclined at 20° to the vertical and against the movement of the drum). A water consumption of 1.5 l/s per m (1 gal./s per 10 ft.) length of drum may be assumed.²¹

Eimcobelt filter

This system was introduced before the Rapifloc, and is very similar to it.

The filter drum is surrounded by a tightly woven cloth of polypropylene which passes over a discharge roller, around an aligning roller and another tension roller, and then returns to the drum. The ends of the cloth are joined together by means of a metal lacing and a "lightning" fastener. The cloth is kept clean by means of two perforated pipes which wash it after the discharge roller, the wash water being recovered in a trough from which it may be removed and recycled or used for imbibition; its polarisation is approximately 0.2%. A third spray pipe washes the cloth from below just before the discharge roller, thus assisting discharge of the cake.

The cloth may have a life of 4 — 5 months. Separan AP 30, diluted to 0.5%, is mixed with the mud, to which bagacillo has been added. A good dosage of Separan is 2 p.p.m.; at 5 p.p.m., Rabe²² has found in South Africa that the cake does not adhere to the cloth; and at 1 p.p.m., the filtrate is dirty and the cloth chokes.

It is important that the wash water should be at least at 3 kg/cm² (40 p.s.i.) and 80°C (175°F), otherwise the cloth becomes blocked. Changing an ordinary Eimco filter to an "Eimcobelt" doubles its capacity. A capacity may be expected²³ of about 100 – 120 kg/m²/h (20 – 25 lb./sq.ft./h). This output can hardly be decreased below 60 without disadvantages nor increased above 155 kg/m²/h (32 lb./sq.ft./h).

The filtrate is sufficiently clear to be sent directly to the evaporators.

Efficiency. The "dressed" filters, Rapifloc or Eimcobelt, have a high retention; they reduce the suspended matter in the filtrate to 0.03 – 0.05%, compared with about 0.80% for the bare filters.²⁴

Air pump

Factories installing filters often attempt to avoid the small vacuum installation involved for

of wash water which passes into the clear filtrate represent only a small fraction of the water used at the filter, about 20–25%. The greater part remains in the filter cake. It is necessary therefore to determine the quantity of water to be used for washing according to the dilution of the clear filtrate, and not according to the quantity of water sent to the filter.

The efficiency of washing is of more importance than the quantity of water used. In South Africa, one factory using water for washing equivalent to 100% on filter cake obtained 0.3% pol in cake, whereas another employing half as much obtained 0.4%.

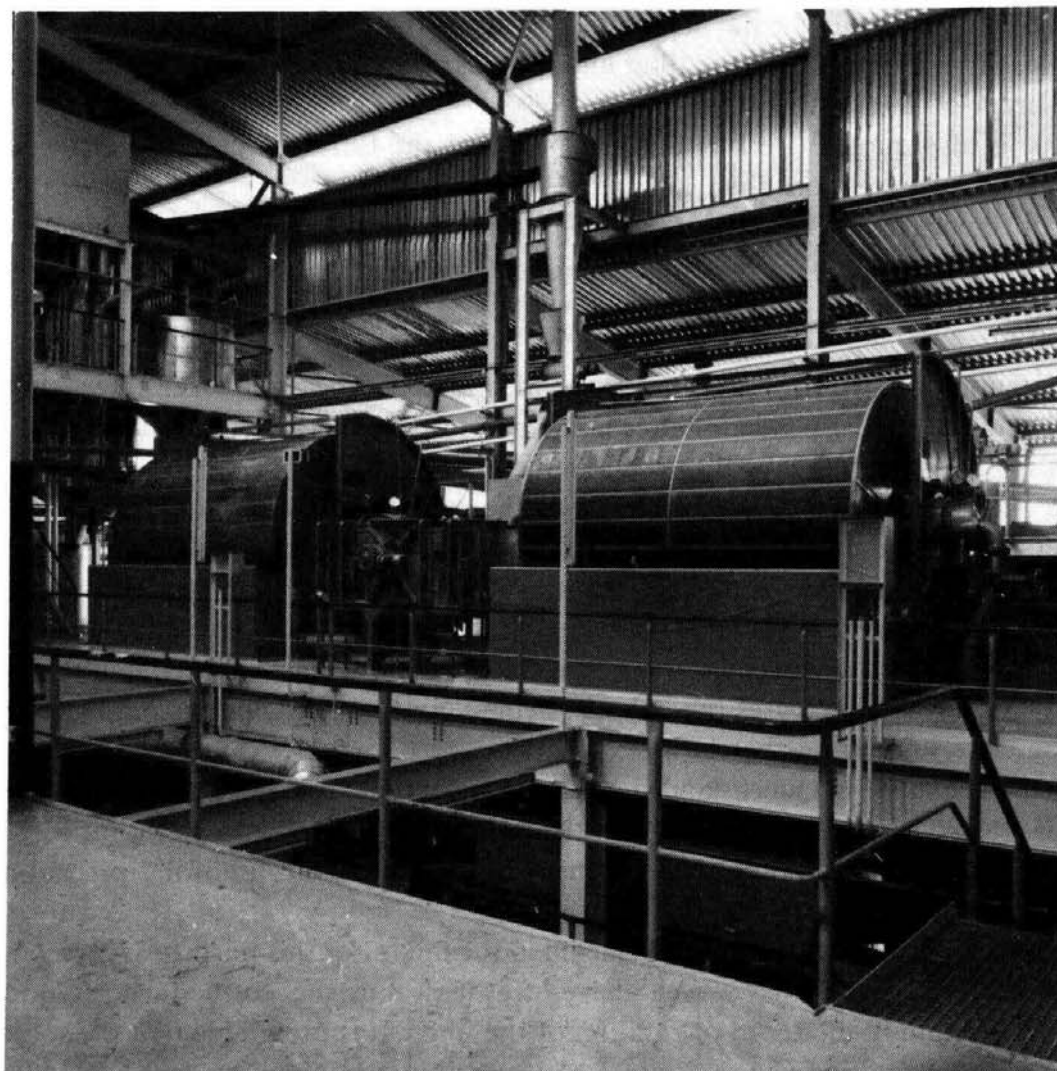


Fig. 31.10. Filtration station B.M.A. filters.

Generally, wash water amounting to 100–150% on cake is used. In Iran, wash water is reckoned as 1.38% on cane by weight. In general, one may assume 14–20 kg/t.c. (30–45 lb./t.c.), or 1.4–2.0%. One South African factory reports 12.6 kg/t.c. (28 lb./t.c.).

This would preferably be water at 75–80°C (167–176°F). However, Webre¹³, considers that it is desirable not to exceed 60°C (140°F) to avoid risk of entraining waxes contained in the filter cake. We do not share this point of view, for a reason completely analogous to that demonstrated by Honig for hot imbibition (*cf.* "Waxes", p. 308). In Louisiana, the

Bagacillo screens

Filters of the Oliver type generally do not furnish a satisfactory filtration unless a certain quantity of bagacillo is added to the muds. This is obtained by screening, and consists mainly of thread-like fibres about 3 cm (1¼ in.) in length and of pieces about 1 mm in diameter. The mixture of these two materials gives a substantial increase in the filterability of the other materials in suspension which constitute the muds. The fibrous threads in particular form a layer over the screen and so provide a network which forms the actual filter medium.

Quantity necessary. The makers estimate²⁵ that it is necessary to introduce about 7 kg (16 lb.) of bagacillo per tonne of cane.

In Cuba²⁶ a figure of approx. 6.5 kg (15 lb.) of fine bagasse at 48% moisture is stated.

In Louisiana^{4a} 6 – 10% of fine bagasse on weight of wet filter cake.

In Queensland²⁷ 4.5 kg/t.c. (10 lb./t.c.).

In South Africa²⁸ 4.5 - 14 kg/t.c. (10 – 32 lb./t.c.); more recently (Perk²⁹, 1960) 15 kg/t.c. (34 lb./t.c.).

In Mauritius, with an average value of 4 kg/t.c. (9 lb./t.c.), Wiehe³⁰ comments that a better result was obtained with 6.5 kg/t.c. (15 lb./t.c.), the loss of sugar in the cake falling by half.

As will be seen, this quantity is rather variable, and depends on the ease of filtration of the juice. We have found it possible to descend to zero while obtaining an excellent sugar recovery. As a general figure one can assume 3 – 6 kg/t.c. (7 – 14 lb./t.c.).

Installation. A shaker screen is located under the bagasse conveyor (Fig. 31.14), preferably between the two chutes feeding bagasse to the first two boiler furnaces. A slide similar to those feeding the furnaces allows the bagasse to fall on the screen, which allows the fine bagacillo to pass and rejects the remainder to the two neighbouring chutes.

In order to obtain the desired quantity of bagacillo which is necessary for a filter treating all the muds from the factory, it will be necessary to pass over the screen between 25 and 60% of the total bagasse produced by the factory. With compound clarification, 25 – 35% will be ample.

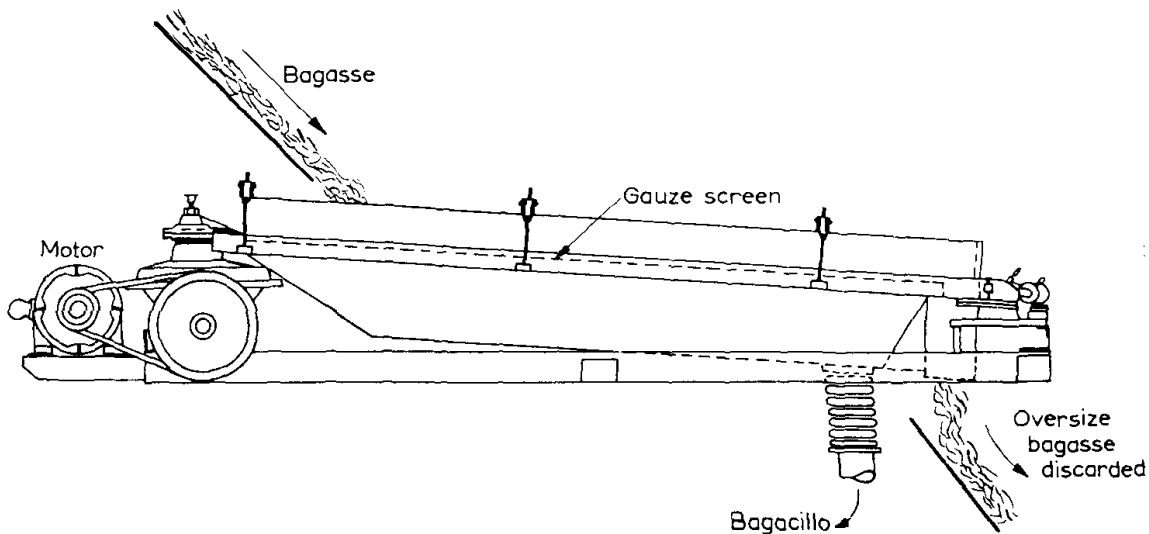


Fig. 31.14. Rotex shaker screen for fine bagasse.

However, in the great majority of cases, this small item of equipment may be omitted by simply replacing a certain length (2.5–5 m) of the bottom plate of the conveyor by a perforated plate with holes of 10 mm ($\frac{3}{16}$ in.) diameter. It is convenient to place this in an inclined section of the conveyor.

A fan conveys the bagacillo from the screen, or from the hopper situated under the perforated bottom plate of the conveyor, through a galvanised iron duct, and delivers it into a cyclone separator placed above a mixer (Fig. 31.15) into which are fed the muds passing to the filter. The quantity of bagacillo conveyed by the fan is controlled by a small slide or butterfly valve in the duct.

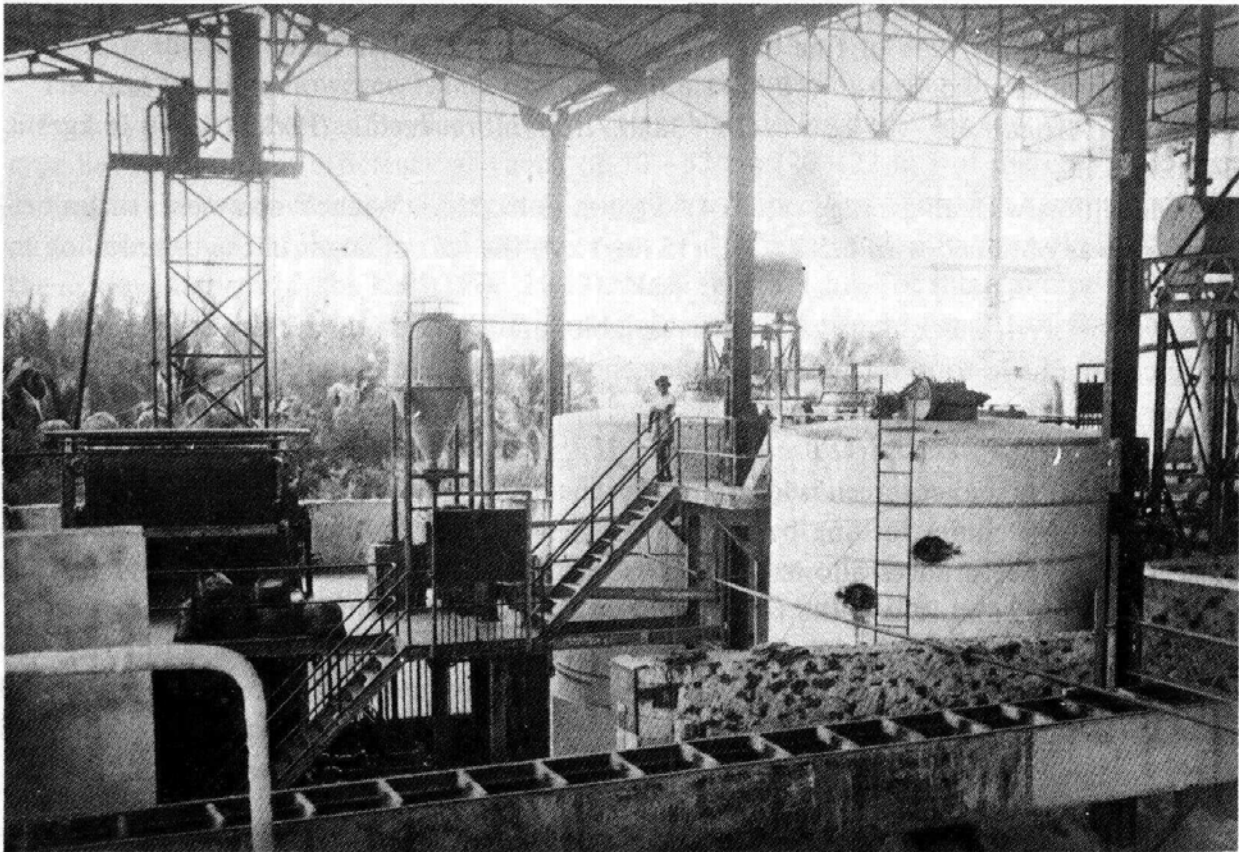


Fig. 31.15. Clarificator (right); Oliver filter (left); cyclone (centre); and condenser (above, left).

Screens. The four principal sizes of gauze used for screens are given in Table 31.3.

The finer screens will be used when the juice does not filter well, provided that the quantity of bagacillo obtained is sufficient. Gauzes coarser than No. 12 would not give such a good filtration; an attempt will be made, if possible, to use only No. 14. It is considered in South Africa³ that 90% of the bagacillo going to the filter should pass through a screen of 0.8 mm, and at least 70% through a 0.5-mm screen.

TABLE 31.3

GAUZES FOR BAGACILLO SCREENS		
<i>English no. (meshes per inch)</i>	<i>Meshes per dm</i>	<i>Proportion of bagacillo obtained, by weight</i>
8 mesh	32	100
10 -	40	90
12 -	48	60
14 -	56	25

TABLE 31.4

OLIVER SERIES OF SCREENS			
<i>mm</i>	<i>m²</i>	<i>inches</i>	<i>sq./ft.</i>
510 X 1,220 =	0.62	20 x 48 =	6.67
1,015 x 2,130 =	2.16	40 x 84 =	23.33
1,015 x 3,050 =	3.10	40 x 120 =	33.33
1,520 x 3,050 =	4.65	60 x 120 =	50

Capacity of bagacillo screens. A screen of 32 mesh per dm (8-mesh) will furnish 60–75 kg of bagacillo/m²/h (12–15 lb./sq.ft./h). In Queensland²⁷ a figure of 0.04 mVt.c.h. (0.4 sq.ft./t.c.h.) is given for an 8-mesh screen. It has been established³² that a screen of 16 mesh per dm (4-mesh) could be suitable and would give a proportion of fine bagasse lower than that furnished by one of 32 (8-mesh). Such screens furnish 133 kg/m²/h (27 lb./sq.ft./h) of fine bagasse in the case of fixed horizontal screens without suction, and 820 (168) with suction.

Sometimes lime is added to the mud, so as to raise the pH to 7.8 or 8.0. This facilitates flocculation and forms a more porous filter cake. Generally a screen of 0.06 mVt.c.h. (0.6 sq.ft./t.c.h.) is installed. The Oliver company supplies 4 sizes of screen, as listed in Table 31.4.

Fan. It is considered in Cuba³³ that, for a delivery duct of 75 m (250 ft.) in length, with three bends, 6.25 m³ of air will be required per kg of bagacillo to be conveyed (100 cu.ft./lb.); air velocity 25 m/s (82 ft/s).

Total power required

The total power required for the whole installation - filter and accessories - is approximately 0.8 kW/m² (1 h.p./10 sq.ft.) of filtering surface. For a filter of 28 m² (300 sq.ft.), for example, there will be required:

Filter drive:	1.9 kW
Vacuum pump:	9 kW
2 Centrifugal pumps for filtrates:	2.3 kW
Screw conveyor for filter cake:	1.5 kW
Bagacillo screen:	0.8 kW
Bagacillo fan:	3.8 kW
Mixer for muds and bagacillo:	0.4 kW
	19.7 kW (26 h.p.)

Operation of a rotary filter

The temperature of the muds to be filtered plays an important part. Care should be taken that this does not fall below 80°C (176°F), and it will be maintained preferably above 85°C (185°F), and if possible, 90°C (194°F). Provided this temperature is maintained, the screen remains clean and spotless, after discharge of the filter cake. At the end of the week, the screen is cleaned by a steam jet from a nozzle provided for the purpose. If this precaution were neglected, the perforations of the screen would become blocked, or the sections would fill with dried mud. In South Africa, it is even recommended³⁴ that the screens should be steam-cleaned every 12 hours.

FILTRATION OF SYRUPS

Filtration of syrup is normally carried out on special types of filter. The principal types are the following, all made under licence from American manufacturers. We indicate for each an order of magnitude of the filtering surface necessary:

- (a) Herfilco, made in France, 3–4 m²/t.c.h. (33–44 sq.ft./t.c.h.)
- (b) Niagara, made in Holland, 2–3 m²/t.c.h. (22–33 sq.ft./t.c.h.).

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32. Evaporation

GENERAL

Heat transmission in an evaporator

A sugar evaporator consists essentially of a tubular calandria serving as a heat exchanger: the heating steam surrounds the outside of the tubes and the juice to be evaporated circulates inside the tubes.

When two fluids of temperature t and t' are located on opposite sides of a solid surface, the quantity of heat transmitted from one to the other is given by the expression:

$$Q = kS(t - t') \quad (32.1)$$

Q = quantity of heat transmitted

k = coefficient of heat transfer

S = heat-exchange surface

t = temperature of the hot fluid

t' = temperature of the cold fluid.

The coefficient k is given by:

$$\frac{1}{k} = \frac{1}{a} + \frac{e}{b} + \frac{1}{c} \quad (32.2)$$

a - coefficient of heat transfer from the hot fluid to the surface

b - coefficient of heat transfer across the solid surface

c = coefficient of heat transfer from the surface to the cold fluid

e = thickness of the metal wall.

Whence:

$$k = \frac{1}{\frac{1}{a} + \frac{e}{b} + \frac{1}{c}} \quad (32.3)$$

This equation shows that the value of the coefficient k is determined by the magnitude of the smallest of the three coefficients a , b , c . If c , for example, is much smaller than a and b , we may neglect the latter and take:

$$k \approx c$$

This is the common case for transmission: vapour — metal tube → juice, the heat transfer coefficients from vapour to metal, and across the metal surface, being very high compared with the coefficient from metal to juice.

Deposits and incrustations form a layer on the tubes, the resistance of which to heat transmission is added to those given; now, the corresponding heat-transfer coefficient is much inferior to any of the three preceding ones. It follows that an evaporator with fouled tubes will function much less effectively than one with clean tubes. It is essential to maintain

TABLE 32.1A

RELATION BETWEEN PRESSURE AND TEMPERATURE OF DRY SATURATED WATER
VAPOUR AT PRESSURES BELOW ATMOSPHERIC (METRIC UNITS)

<i>Vacuum</i> (cm)	<i>Hg abs.</i> (cm)	<i>Abs. press.</i> (kg/cm ²)	<i>Temp.</i> (°C)	<i>Vacuum</i> (cm)	<i>Hg abs.</i> (cm)	<i>Abs. press.</i> (kg/cm ²)	<i>Temp.</i> (°C)
76	0	0.		38	38	0.517	81.7
75	1	0.0136	11.2	37	39	0.530	82.3
74	2	0.0272	22.1	36	40	0.544	83.0
73	3	0.0408	29.0	35	41	0.557	83.6
72	4	0.0544	34.0	34	42	0.571	84.2
71	5	0.0680	38.1	33	43	0.585	84.8
70	6	0.0816	41.5	32	44	0.598	85.4
69	7	0.0952	44.5	31	45	0.612	86.0
68	8	0.109	47.1	30	46	0.625	86.5
67	9	0.122	49.4	29	47	0.639	87.1
66	10	0.136	51.6	28	48	0.653	87.6
65	11	0.150	53.5	27	49	0.666	88.1
64	12	0.163	55.3	26	50	0.680	88.7
63	13	0.177	57.0	25	51	0.693	89.2
62	14	0.190	58.6	24	52	0.707	89.7
61	15	0.204	60.1	23	53	0.721	90.2
60	16	0.218	61.5	22	54	0.734	90.7
59	17	0.231	62.8	21	55	0.748	91.2
58	18	0.245	64.1	20	56	0.761	91.7
57	19	0.258	65.3	19	57	0.775	92.1
56	20	0.272	66.4	18	58	0.789	92.6
55	21	0.285	67.5	17	59	0.802	93.1
54	22	0.299	68.6	16	60	0.816	93.5
53	23	0.313	69.6	15	61	0.830	94.0
52	24	0.326	70.6	14	62	0.843	94.4
51	25	0.340	71.6	13	63	0.856	94.8
50	26	0.353	72.5	12	64	0.870	95.3
49	27	0.367	73.4	11	65	0.884	95.7
48	28	0.381	74.2	10	66	0.897	96.1
47	29	0.394	75.1	9	67	0.911	96.5
46	30	0.408	75.9	8	68	0.924	96.9
45	31	0.421	76.7	7	69	0.938	97.4
44	32	0.435	77.4	6	70	0.952	97.8
43	33	0.449	78.2	5	71	0.965	98.1
42	34	0.462	78.9	4	72	0.979	98.5
41	35	0.476	79.6	3	73	0.992	98.9
40	36	0.489	80.3	2	74	1.006	99.3
39	37	0.503	81.0	1	75	1.020	99.6
38	38	0.517	81.7	0	76	1.033	100.0

the tubes in a clean condition, and it is cleaning of the tubes which provides the most imperative reason for a weekly shutdown of a sugar factory which does not have spare evaporator vessels.

Superheated steam. If heat transmission from saturated steam to metal is excellent, superheated steam gives, on the contrary, a very poor heat-transfer coefficient.

TABLE 32.1B

RELATION BETWEEN PRESSURE AND TEMPERATURE OF DRY SATURATED WATER
VAPOUR AT PRESSURES BELOW ATMOSPHERIC (BRITISH UNITS)

<i>Vacuum</i> (in.)	<i>Hg abs.</i> (in.)	<i>Abs. press.</i> (p.s.i.)	<i>Temp.</i> (°F)	<i>Vacuum</i> (in.)	<i>Hg abs.</i> (in.)	<i>Abs. press.</i> (p.s.i.)	<i>Temp.</i> (°F)
29.92	0	0		15	14.92	7.328	178.9
29	0.92	0.452	76.5	14	15.92	7.819	181.8
28	1.92	0.943	99.8	13	16.92	8.310	184.6
27	2.92	1.434	114.1	12	17.92	8.802	187.2
26	3.92	1.925	124.7	11	18.92	9.293	189.8
25	4.92	2.417	133.1	10	19.92	9.784	192.2
24	5.92	2.908	140.3	9	20.92	10.275	194.5
23	6.92	3.399	146.4	8	21.92	10.766	196.7
22	7.92	3.890	151.8	7	22.92	11.257	198.9
21	8.92	4.381	156.7	6	23.92	11.749	200.9
20	9.92	4.872	161.1	5	24.92	12.240	202.9
19	10.92	5.363	165.2	4	25.92	12.731	204.8
18	11.92	5.855	169.0	3	26.92	13.222	206.7
17	12.92	6.346	172.5	2	27.92	13.713	208.5
16	13.92	6.837	175.8	1	28.92	14.204	210.3
15	14.92	7.328	178.9	0	29.92	14.696	212.0

Claassen¹ attributes to this latter coefficient 0.8% only of the value of the preceding, that is to say, 125 times less.

However, a slight superheat would not present any disadvantage. In France², evaporation has been conducted with beet juices, heating the first vessel with steam at 1.65 kg/cm² (23.5 p.s.i.) and 150°C (302°F), that is to say, with a superheat of 21 °C (38°F). The juice was boiling at 120°C (248°F), and the condensate left the calandria at 129°C (264°F), the saturation temperature corresponding to the vapour pressure. The heat transmission was normal.

Jaroschek³ states that superheat is permissible up to a temperature of 160°C (320°F). It would be only above that temperature that it would be necessary to cool the vapour. Webre⁴ indicates as a result of his own experience, after careful investigations, that degrees of superheat not exceeding 55°C (100°F) would not have any effect on heat transmission in multiple effects and vacuum pans.

Beale and Stewart⁵ report a test which indicated that it was possible to reach without disadvantage 45°C (80°F) of superheat and that the heat-transfer coefficient did not fall by more than 7.5% when the superheat rose to 80°C (180°F).

It is thus satisfactory to use steam with 30–50°C (50–90°F) of superheat.

Boiling point. (a) *Pressure.* The boiling point of water depends on the pressure. We give (Table 32.1) boiling points corresponding to various pressures.

(b) *Brix.* However, in a multiple effect in the sugar factory, it is not water which is boiling but juice. Now the boiling point of a solution of sugar in water, or of a juice, under a given pressure, increases with the concentration of this solution, or with the brix of this juice. Under atmospheric pressure, mixed juice, for example, boils at 100.25°C (212.5°F) instead of 100°C (212°F). Table 32.2 gives the value of the boiling-point elevation of solutions of pure sucrose (according to Claassen)

TABLE 32.2

BOILING-POINT ELEVATION OF SUGAR SOLUTIONS AND CANE JUICES
(°C) AT 760 mm PRESSURE (AFTER CLAASSEN AND THIEME)

<i>Brix</i>	<i>Purity</i>						
	100	90	80	70	60	50	40
10	0.1	0.1	0.1	0.1	0.2	0.2	0.2
15	0.2	0.2	0.2	0.2	0.3	0.3	0.4
20	0.3	0.3	0.3	0.4	0.4	0.5	0.6
25	0.4	0.5	0.5	0.6	0.7	0.8	0.9
30	0.6	0.7	0.7	0.8	1.0	1.1	1.2
35	0.8	0.9	1.0	1.1	1.3	1.4	1.6
40	1.0	1.1	1.3	1.5	1.7	1.9	2.1
45	1.4	1.5	1.8	2.0	2.2	2.4	2.7
50	1.8	1.9	2.2	2.5	2.8	3.1	3.4
55	2.3	2.5	2.8	3.1	3.5	3.9	4.3
60	3.0	3.2	3.6	4.0	4.4	4.9	5.4
65	3.8	4.1	4.5	4.9	5.5	6.0	6.5
70	5.1	5.5	6.0	6.5	7.1	7.7	8.3
75	7.0	7.5	8.0	8.6	9.4	10.1	10.8
80	9.4	10.0	10.5	11.3	12.3	13.1	14.1
85	13.0	13.7	14.4	15.3	16.4	17.4	19.1
90	19.6	20.5	21.2	22.4	23.7	25.3	
94	30.5						

and of cane juices (according to Thieme). The figures for low brix values and for a purity of 40 have been obtained by interpolation.

It will be seen that, for high purities, the boiling-point elevation is of the order of magnitude of:

$$e = \frac{2B}{100 - B} \quad \left(e = \frac{3.6B}{100 - B} \right) \quad (32.4)$$

e = boiling-point elevation, in °C (°F)

B = brix of the solution.

The boiling-point elevation for a pure solution may be calculated by the equation:

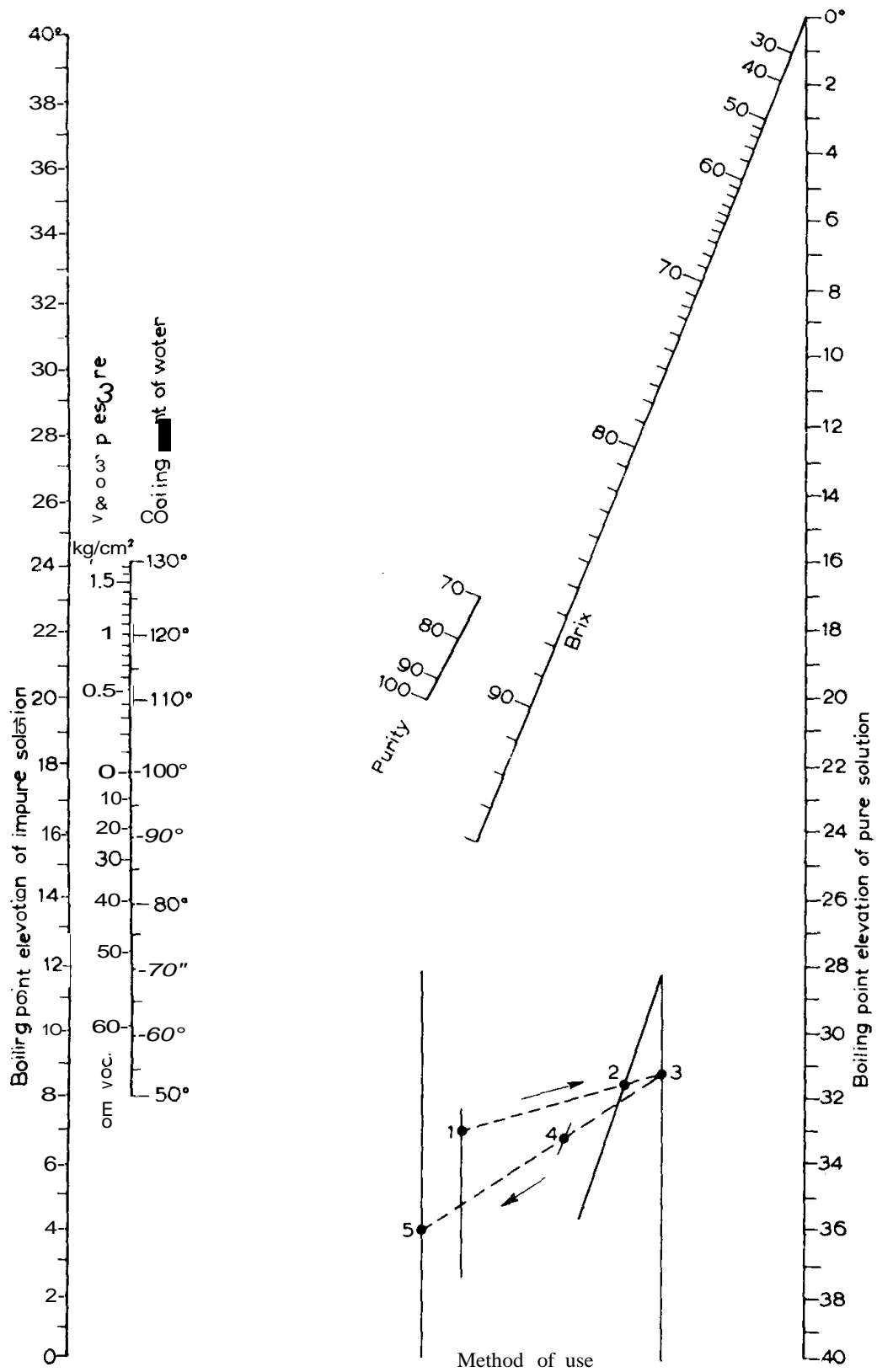


Fig. 32.1. Nomograph of Othmer and Silvis.

$$e = 0.025 B \frac{30 + B}{103.6 - B} \left(1 - \frac{0.54 h}{229 - h} \right)$$

$$\left(e = 0.045 B \frac{30 + B}{103.6 - B} \left(1 - \frac{1.37 h}{190.5 - h} \right) \right)$$

B = brix

h = vacuum, in cm (in.) of mercury.

(c) *Combined effects of brix and pressure.* Table 32.2 has been established for atmospheric pressure. Now the boiling-point elevation of sugar solutions varies with the pressure, but between fairly narrow limits.

The boiling-point elevation of a solution of any brix and purity under any given pressure may be obtained by the use of the nomograph of Othmer and Silvis⁶, which we reproduce in Fig. 32.1. The use of this nomograph gives, for atmospheric pressure, results differing slightly from the figures indicated in Table 32.2. There is no need to seek any greater precision.

Hydrostatic pressure. When a pressure P is exerted on the surface of a liquid, the pressure to which the molecules of the liquid are subjected at a certain depth in the liquid is equal to P increased by the pressure of liquid corresponding to the depth.

Since the boiling point increases with the pressure, if the temperature of the liquid corresponds to that necessary to produce boiling at the surface, this boiling would cease in the liquid layers situated at a certain depth.

However, in a vessel of a multiple effect evaporator, we shall assume that the transmission of heat is sufficiently rapid for the juice to boil throughout the full length of the tubes. If it were otherwise, in fact, the juice would not climb up the tubes. Moreover, since the juice arrives in the vessel at a temperature higher than that ruling in that vessel (since it comes from the preceding one), there is no need for any of the heating surface to be used to bring it up to boiling point; it boils immediately it makes contact with the heating surface (and in fact, even before making this first contact, owing to self-evaporation).

Under these conditions it may be assumed that the layer of juice which is situated at the level of the bottom tube plate will be subject to a hydrostatic pressure equal to the pressure of juice which lies above it. If, for example, the hydrostatic level of the juice in the vessel corresponds to one-third of the height of the tubes, this layer will boil at a temperature corresponding to the vapour pressure in the vessel increased by the hydrostatic pressure to which it is subjected.

In practice the matter is not quite as simple as this, owing to the state of agitation of the juice in the vessel; but it is reasonable to consider that this hypothesis, applied to the various depths of juice covering the tubular heating surface, will give a correct idea of the mean temperature of the juice.

This comment is of importance for the calculation of multiple effects. If all the juice were at the temperature corresponding to the pressure obtaining in the vapour space of the vessel, the temperature drop between heating vapour and juice would be obtained by simply deduc-

TABLE 32.3

BOILING-POINT ELEVATION e' AT THE BOTTOM OF A LIQUID COLUMN,
RELATIVE TO BOILING POINT AT THE SURFACE T (IN °C)

H (cm)	Brix	d	P (kg/cm ²)	T							
				120	110	100	90	80	70	60	50
10	0	1.00	0.0100	0.2	0.2	0.3	0.4	0.5	0.7	1.0	1.5
	10	1.04	0.0104	0.2	0.2	0.3	0.4	0.5	0.7	1.1	1.6
	20	1.08	0.0108	0.2	0.2	0.3	0.4	0.5	0.8	1.1	1.7
	30	1.13	0.0113	0.2	0.2	0.3	0.4	0.6	0.8	1.2	1.8
	40	1.18	0.0118	0.2	0.2	0.3	0.4	0.6	0.8	1.2	1.9
	50	1.23	0.0123	0.2	0.2	0.3	0.4	0.6	0.9	1.3	1.9
	60	1.29	0.0129	0.2	0.3	0.3	0.5	0.7	0.9	1.3	2.0
	70	1.35	0.0135	0.2	0.3	0.4	0.5	0.7	1.0	1.4	2.1
	<i>MC</i>	1.47	0.0147	0.2	0.3	0.4	0.5	0.7	1.0	1.5	2.3
20	0	1.00	0.0200	0.3	0.4	0.5	0.7	1.0	1.4	2.0	3.0
	10	1.04	0.0208	0.3	0.4	0.6	0.8	1.0	1.5	2.1	3.1
	20	1.08	0.0216	0.3	0.4	0.6	0.8	1.1	1.5	2.2	3.2
	30	1.13	0.0226	0.3	0.5	0.6	0.8	1.1	1.6	2.3	3.4
	40	1.18	0.0236	0.4	0.5	0.6	0.9	1.2	1.7	2.4	3.5
	50	1.23	0.0246	0.4	0.5	0.7	0.9	1.2	1.7	2.5	3.7
	60	1.29	0.0258	0.4	0.5	0.7	0.9	1.3	1.8	2.6	3.8
	70	1.35	0.0270	0.4	0.5	0.7	1.0	1.3	1.9	2.7	4.0
	<i>MC</i>	1.47	0.0294	0.5	0.6	0.8	1.1	1.5	2.1	3.0	4.3
30	0	1.00	0.0300	0.5	0.6	0.8	1.1	1.5	2.1	3.0	4.4
	10	1.04	0.0312	0.5	0.6	0.9	1.1	1.6	2.2	3.1	4.6
	20	1.08	0.0324	0.5	0.7	0.9	1.2	1.6	2.3	3.2	4.7
	30	1.13	0.0339	0.5	0.7	0.9	1.2	1.7	2.4	3.4	4.9
	40	1.18	0.0354	0.5	0.7	0.9	1.3	1.8	2.5	3.5	5.1
	50	1.23	0.0369	0.6	0.7	1.0	1.3	1.8	2.6	3.7	5.3
	60	1.29	0.0387	0.6	0.8	1.0	1.4	1.9	2.7	3.8	5.5
	70	1.35	0.0405	0.6	0.8	1.1	1.5	2.0	2.8	4.0	5.7
	<i>MC</i>	1.47	0.0441	0.7	0.9	1.2	1.6	2.2	3.0	4.3	6.2
40	0	1.00	0.0400	0.6	0.8	1.1	1.4	2.0	2.8	3.9	5.7
	10	1.04	0.0416	0.6	0.8	1.1	1.5	2.1	2.9	4.1	5.9
	20	1.08	0.0432	0.7	0.9	1.1	1.6	2.1	3.0	4.2	6.1
	30	1.13	0.0452	0.7	0.9	1.2	1.6	2.2	3.1	4.4	6.3
	40	1.18	0.0472	0.7	0.9	1.3	1.7	2.3	3.2	4.6	6.6
	50	1.23	0.0492	0.8	1.0	1.3	1.8	2.4	3.4	4.8	6.8
	60	1.29	0.0516	0.8	1.0	1.4	1.8	2.5	3.5	5.0	7.1
	70	1.35	0.0540	0.8	1.1	1.4	1.9	2.6	3.7	5.2	7.4
	<i>MC</i>	1.47	0.0588	0.9	1.2	1.6	2.1	2.9	4.0	5.6	8.0
50	0	1.00	0.0500	0.8	1.0	1.3	1.8	2.5	3.4	4.8	6.9
	10	1.04	0.0520	0.8	1.0	1.4	1.9	2.5	3.5	5.0	7.2
	20	1.08	0.0540	0.8	1.1	1.4	1.9	2.6	3.7	5.2	7.4
	30	1.13	0.0565	0.9	1.1	1.5	2.0	2.8	3.8	5.4	7.7
	40	1.18	0.0590	0.9	1.2	1.6	2.1	2.9	4.0	5.6	8.0
	50	1.23	0.0615	0.9	1.2	1.6	2.2	3.0	4.1	5.8	8.3
	60	1.29	0.0645	1.0	1.3	1.7	2.3	3.1	4.3	6.1	8.6
	70	1.35	0.0675	1.0	1.3	1.8	2.4	3.3	4.5	6.3	8.9
	<i>MC</i>	1.47	0.0735	1.1	1.5	1.9	2.6	3.5	4.9	6.8	9.6

TABLE 32.3 (continued)

<i>H</i> (cm)	<i>Brix</i>	<i>d</i>	<i>P</i> (kg/cm ²)	<i>f</i>							
				120	110	100	90	80	70	60	50
60	0	1.00	0.0600	0.9	1.2	1.6	2.1	2.9	4.0	5.7	8.1
	10	1.04	0.0624	1.0	1.2	1.6	2.2	3.0	4.2	5.9	8.4
	20	1.08	0.0648	1.0	1.3	1.7	2.3	3.1	4.4	6.1	8.6
	30	1.13	0.0678	1.0	1.3	1.8	2.4	3.3	4.5	6.4	9.0
	40	1.18	0.0708	1.1	1.4	1.9	2.5	3.4	4.7	6.6	9.3
	50	1.23	0.0738	1.1	1.5	1.9	2.6	3.6	4.9	6.9	9.6
	60	1.29	0.0774	1.2	1.5	2.0	2.7	3.7	5.1	7.2	10.0
	70	1.35	0.0810	1.2	1.6	2.1	2.9	3.9	5.3	7.4	10.4
	MC	1.47	0.0882	1.4	1.7	2.3	3.1	4.2	5.8	8.0	11.1
70	0	1.00	0.0700	1.1	1.4	1.8	2.5	3.4	4.7	6.5	9.2
	10	1.04	0.0728	1.1	1.4	1.9	2.6	3.5	4.8	6.8	9.5
	20	1.08	0.0756	1.2	1.5	2.0	2.7	3.6	5.0	7.0	9.8
	30	1.13	0.0790	1.2	1.6	2.1	2.8	3.8	5.2	7.3	10.2
	40	1.18	0.0826	1.3	1.6	2.2	2.9	3.9	5.4	7.6	10.6
	50	1.23	0.0861	1.3	1.7	2.3	3.0	4.1	5.6	7.8	10.9
	60	1.29	0.0903	1.4	1.8	2.4	3.2	4.3	5.9	8.2	11.4
	70	1.35	0.0945	1.4	1.9	2.5	3.3	4.5	6.1	8.5	11.8
	MC	1.47	0.1029	1.6	2.0	2.7	3.6	4.8	6.6	9.1	12.6
80	0	1.00	0.0800	1.2	1.6	2.1	2.8	3.8	5.3	7.4	10.3
	10	1.04	0.0832	1.3	1.7	2.2	2.9	4.0	5.5	7.6	10.6
	20	1.08	0.0864	1.3	1.7	2.3	3.0	4.1	5.7	7.9	11.0
	30	1.13	0.0904	1.4	1.8	2.4	3.2	4.3	5.9	8.2	11.4
	40	1.18	0.0944	1.4	1.9	2.4	3.3	4.5	6.1	8.5	11.7
	50	1.23	0.0984	1.5	1.9	2.6	3.4	4.7	6.4	8.8	12.2
	60	1.29	0.1032	1.6	2.0	2.7	3.6	4.9	6.6	9.2	12.6
	70	1.35	0.1080	1.6	2.1	2.8	3.7	5.1	6.9	9.5	13.1
	MC	1.47	0.1176	1.8	2.3	3.0	4.0	5.5	7.4	10.2	14.0
90	0	1.00	0.0900	1.4	1.8	2.3	3.1	4.3	5.9	8.2	11.3
	10	1.04	0.0936	1.4	1.9	2.4	3.3	4.4	6.1	8.4	11.7
	20	1.08	0.0972	1.5	1.9	2.5	3.4	4.6	6.3	8.7	12.0
	30	1.13	0.1017	1.5	2.0	2.6	3.5	4.8	6.6	9.0	12.5
	40	1.18	0.1062	1.6	2.1	2.8	3.7	5.0	6.8	9.4	12.9
	50	1.23	0.1107	1.7	2.2	2.9	3.8	5.2	7.1	9.7	13.3
	60	1.29	0.1161	1.8	2.3	3.0	4.0	5.4	7.4	10.1	13.8
	70	1.35	0.1215	1.8	2.4	3.1	4.2	5.7	7.7	10.5	14.3
	MC	1.47	0.1323	2.0	2.6	3.4	4.5	6.1	8.3	11.2	15.3
100	0	1.00	0.1000	1.5	2.0	2.6	3.5	4.7	6.5	8.9	12.3
	10	1.04	0.1040	1.6	2.0	2.7	3.6	4.9	6.7	9.2	12.7
	20	1.08	0.1080	1.6	2.1	2.8	3.7	5.1	6.9	9.5	13.5
	30	1.13	0.1130	1.7	2.2	2.9	3.9	5.3	7.2	9.9	13.5
	40	1.18	0.1180	1.8	2.3	3.0	4.1	5.5	7.5	10.2	14.0
	50	1.23	0.1230	1.9	2.4	3.2	4.2	5.7	7.7	10.6	14.5
	60	1.29	0.1290	2.0	2.5	3.3	4.4	6.0	8.1	11.0	15.0
	70	1.35	0.1350	2.0	2.6	3.5	4.6	6.2	8.4	11.4	15.5
	MC	1.47	0.1470	2.2	2.9	3.8	5.0	6.7	9.1	12.2	16.5
200	MC	1.47	0.2940	4.3	5.5	7.0	9.2	12.2	15.9	20.7	26.6

ting from the apparent temperature drop the boiling-point elevation corresponding to the vapour pressure and to the brix of the juice. But since the juice in contact with the heating surface boils at different temperatures, according to the depth, ranging from that at the surface to that corresponding to the level of the lower tube plate, the temperature drop will therefore be obtained by deducting from the apparent drop: (1) the boiling-point elevation e corresponding to the pressure in the vapour space and to the brix; (2) the supplementary elevation e' corresponding to the mean hydrostatic pressure.

If, for example, the hydrostatic level of the juice is at one-third of the height of the tubes, it may be assumed that the mean hydrostatic pressure will be that corresponding to one-sixth of the height of the tubes. In modern evaporators with long tubes, this consequence of the hydrostatic pressure is far from being negligible.

Table 32.3 gives the boiling-point elevation e' of the lower layers of sugar solutions of various concentrations, as a function of the boiling point T at the surface corresponding to the pressure obtaining above the liquid. In this Table, all figures are given in metric units:

H = depth of the liquid layer under consideration, in cm

d = density of the solution

P = hydrostatic pressure which is added to the pressure obtaining above the liquid, and is expressed in kg/cm^2 (for p.s.i., multiply by 14.22)

MC = heavy massecuite.

Properties of steam. In evaporator calculations, the properties of saturated steam are involved at every step. We shall find in Table 41.1 (p. 1034):

(1) The boiling point t of water corresponding to each pressure

(2) The sensible heat q contained in water which has attained that temperature

(3) The latent heat or heat of vaporisation r of water, for this temperature, *i.e.* the quantity of heat units which must be supplied to the water to change it into vapour at the pressure and temperature given.

For the temperatures which are met with in evaporation work, these two quantities of heat, q and r , may be obtained with sufficient precision by the formulae of Regnault:

$$q = t \quad (q = t - 32 \quad (t \geq 32^\circ\text{F})) \quad (32.5)$$

$$r = 607 - 0.7t \quad (r = 1.093 - 0.7(t - 32)) \quad (32.6)$$

$$\lambda = q + r = 607 + 0.3t \quad (\lambda = 1.093 + 0.3(t - 32)) \quad (32.7)$$

q = sensible heat supplied to unit weight of water to raise it from freezing point to t° , expressed in kcal/kg (B.T.U./lb.)

r = latent heat of vaporisation of water at t° , *i.e.* the quantity of heat to change unit weight of water into steam at the same temperature, in kcal/kg (B.T.U./lb.)

X = total heat to be supplied to unit weight of water, starting at 0°C (32°F), to change it to steam at t° , expressed in kcal/kg (B.T.U./lb.).

We would comment that these three equations adopt for their datum point the temperature of 0°C (32°F).

Strictly speaking, the specific heat of water increases slightly with temperature. At 100°C

(212°F), it is not 1 but 1.013. It is not necessary, however, to complicate the preceding formulae by taking this into account.

Object of evaporation

The clarification process has given a clear juice. This consists of sugar dissolved in water, together with certain impurities. Now that we have removed the impurities as far as possible, it remains to remove the water. This is the object of evaporation.

However, in proportion as we extract water from the juice, the sugar will become concentrated. It will then approach the point of saturation, that is, the point at which crystals will begin to appear in the liquid.

The concentration is pushed to its maximum until there is left for the mother liquor only the space remaining free between the crystals. The name given to the mixture so obtained of solid crystals and viscous mother liquor is "massecuite". Such a mass obviously cannot be handled like a juice or a liquid syrup. Hence the concentration process is separated into two phases:

(a) Evaporation so called, which proceeds from clear juice to syrup, and during which we are dealing only with a liquid material

(b) The sugar boiling, which commences just before the stage where crystals appear in the syrup, and which proceeds up to the maximum concentration.

The impurities obviously remain in the mother liquor, as also does part of the sucrose. It remains then to separate the sugar crystals from the mother liquor and to strive to extract from the latter as much as possible of the sugar which it retains. We shall study these operations under sugar boiling and centrifuging.

Boundary between evaporation and sugar boiling. The point at which cane juice begins to crystallise is in the neighbourhood of 78 – 80° brix (41 – 42° Baumé). Theoretically, therefore, the evaporation could be carried on up to 72 or 75° brix. In practice, the sugar boilers require a syrup capable of dissolving further crystals, so that they may dissolve false grain which may form at the beginning of a charge; this condition is indispensable for obtaining good results at the pan stage.

For this reason a brix figure of 72° is never exceeded. The evaporators are designed and operated in such a way that the brix of the syrup lies:

For raw sugar manufacture: between 60 and 70° brix (32 – 37° Be)

For white sugar manufacture: between 50 and 60° brix (27 – 32° Be).

Quantity of water to be evaporated

Let:

E = weight of water to be evaporated % cane, in kg (lb.)

J = weight of juice obtained % cane, in kg (lb.)

S = weight of syrup % cane, leaving the multiple effects, in kg (lb.)

B_j = brix of juice

B_s = brix of syrup.

Since the weight of dissolved material is the same before and after evaporation, we may equate these quantities as follows:

$$J B_j = S B_s$$

whence:

$$S = J \frac{B_j}{B_s} \quad (32.8)$$

and:

$$E = J - S = J \left(1 - \frac{B_j}{B_s} \right) \quad (32.9)$$

if $J = 100\%$ cane, $B_j = 13$, $B_s = 65$, we have:

$$E = 100 \left(1 - \frac{13}{65} \right) = 80\% \text{ cane}$$

We see also here the importance of evaporation in the sugar factory: it is the operation which eliminates by far the greatest weight of material: 80% of the weight of cane. By reason of the high latent heat of vaporisation of water, this will also obviously be the operation which involves the greatest number of heat units.

Multiple-Effect Evaporation

PRINCIPLE OF MULTIPLE EFFECTS

The greatest and most striking advance in the history of sugar manufacture was no doubt the development of multiple-effect evaporation, conceived about 1830 by Norbert Rillieux, an American of French origin, and introduced by him in Louisiana in 1844.

Evaporation in open pots over a fire had already been abandoned, and evaporation of juice by heating with steam had been commenced. Rillieux's idea was the following: since steam is used for heating juice to evaporate the water which it contains, why not utilise in the same way the vapour so furnished by the juice in order to heat a further portion of juice, or to finish the evaporation already commenced by ordinary steam?

One obstacle presents itself immediately: with steam at 110°C (230°F) (pressure of 0.4 kg/cm^2 (6 p.s.i.)), juice at atmospheric pressure may be heated and evaporated. The vapour of juice so boiling at atmospheric pressure is at 100°C (212°F). But with vapour at that temperature it is not possible to boil juice at the same temperature; a temperature difference is necessary between the heating fluid and the fluid to be heated.

Rillieux resolved this difficulty by putting the vessels following the first under vacuum, water or juice boiling at 90°C (194°F) under 23 cm (9 in.) of vacuum, at 80°C (176°F) under

40 cm (16 in.) of vacuum, at 70°C (158°F) under 52 cm (20 in.), etc.; it thus became possible to create the necessary temperature difference and to utilise the vapour arising from the juice in the first vessel to heat the juice in the second vessel, the vapour produced by the second to heat the third, and so on (Fig. 32.2).

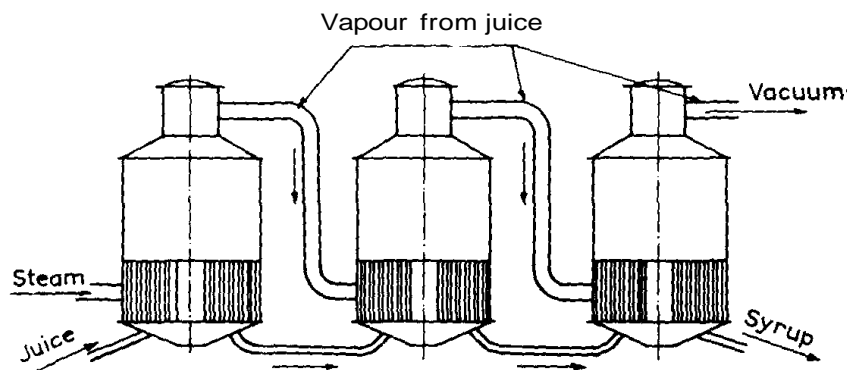


Fig. 32.2. Multiple-effect principle.

This solution has the disadvantage of requiring an installation to create the necessary vacuum. But boiling under vacuum presents two great advantages:

(a) It increases the total difference in temperature between steam and juice by a quantity equal to the drop in boiling point of the juice between the pressure of the first and that of the last vessel.

(b) It permits evaporation to be carried out at temperatures proportionately less dangerous, from the point of view of inversion and of coloration of the juice, as the juice becomes more concentrated and more viscous.

Upper limit. There is a certain critical temperature above which the sugar in the juice tends to caramelize, causing at the same time a loss of sucrose and a coloration which will persist to the final crystals of sugar.

In Czechoslovakia⁷, and for beet juices, the critical temperature has been stated as 130°C (266°F). Quillard⁸ considers that the first experiments of Kestner in 1906 demonstrated that juices could be submitted for a short time to a temperature of 129°C (264°F) without producing any decomposition. For cane juices, Deerr⁹ gives a temperature of 127°C (260°F); Tromp¹⁰ indicates 130°C (266°F) as the maximum temperature for steam for heating, or approximately 125°C (257°F) for the juice.

Honig¹¹ recommends 118°C (244°F), on condition that the juice does not remain more than two minutes at this temperature; but he is thinking then particularly of colour development in the juice.

The limits acceptable for beet sugar juice cannot be used for cane, since beet juice is more stable to temperature and less liable to deteriorate than cane juice.

For normal acidities of the juices circulating in a multiple effect (pH = 6.5 — 7.2), the loss of sucrose by inversion scarcely exceeds 0.1% per hour at 110°C (230°F). Above this, the loss increases rapidly. So that it may remain negligible in an ordinary multiple effect where the juice remains several minutes, or in a Kestner, where it spends 1–2 minutes, it is ad-

visible not to exceed, as the temperature of the juice in the first vessel, the figures gives in Table 32.4.

In the case of falling-film evaporation, where the passage of the juice is very rapid, temperatures may be allowed up to 130°C (266°F).

TABLE 32.4

MAXIMUM TEMPERATURES IN MULTIPLE EFFECTS

Ordinary multiple effect	120°C (1.0 kg/cm ²) = 248°F (14.3 p.s.i.)
Kestner	125°C (1.3 kg/cm ²) = 257°F (19.3 p.s.i.)
This corresponds to steam temperatures in the calandria of the first vessel of approximately:	
Ordinary multiple effect	125 – 130°C (1.3 - 1.7 kg/cm ²) = 257 – 266°F (19.3 – 24.6 p.s.i.)
Kestner	130 – 135°C (1.7-2.2 kg/cm ²) = 266 – 275°F (24.6 – 30.9 p.s.i.)

Lower limit. The lower limit of the temperature range withing which a multiple effect operates is the boiling point corresponding to the vacuum obtained in the last vessel of the set.

In the sugar factory, a vacuum of 66 – 68 cm (26 – 27 in.) is readily obtained, which would correspond to a boiling temperature of 47 – 52°C (117 – 125°F). But for various reasons which we shall see later (pp. 541, 610 and 649), it is of advantage to maintain a vacuum of only 63 – 66 cm (25 – 26 inches), or 56 – 52°C (133 - 125°F). We shall normally reckon on 55°C or 64 cm of vacuum, and it will be this temperature and this vacuum which we shall recommend should be sought at this point.

Overall temperature range. Adopting the latter value, we see that the extreme range of temperatures within which evaporation is carried out is 70°C (= 125 - 55°C). This extreme range is restricted generally to 55 or 60°C (= 115 or 110 - 55°C).

Syrup temperature. The syrup enters the last vessel generally at about 40° brix and leaves it at 60 or 65° brix. It has a mean brix value then of 50 – 55. The boiling-point elevation corresponding to this brix and this vacuum is approximately 2°C (4°F) (*cf.* Fig. 32.1).

To these 2°C, the boiling-point elevation due to hydrostatic pressure should be added. Hence a temperature of 55°C (131°F) in the vapour space of the last vessel corresponds to 58 – 60°C (136- 140°F) as the mean temperature of the syrup in this vessel.

CONSTRUCTION OF MULTIPLE EFFECTS

Evaporation by heating with steam in a single vessel is called evaporation in single effect.

If the juice from the vapour in this vessel is taken for heating a second vessel we have a double effect. In the same way, with three vessels, we have a triple effect. And so on in turn: a quadruple, a quintuple, a sextuple effect. There are thus as many evaporator vessels as effects.

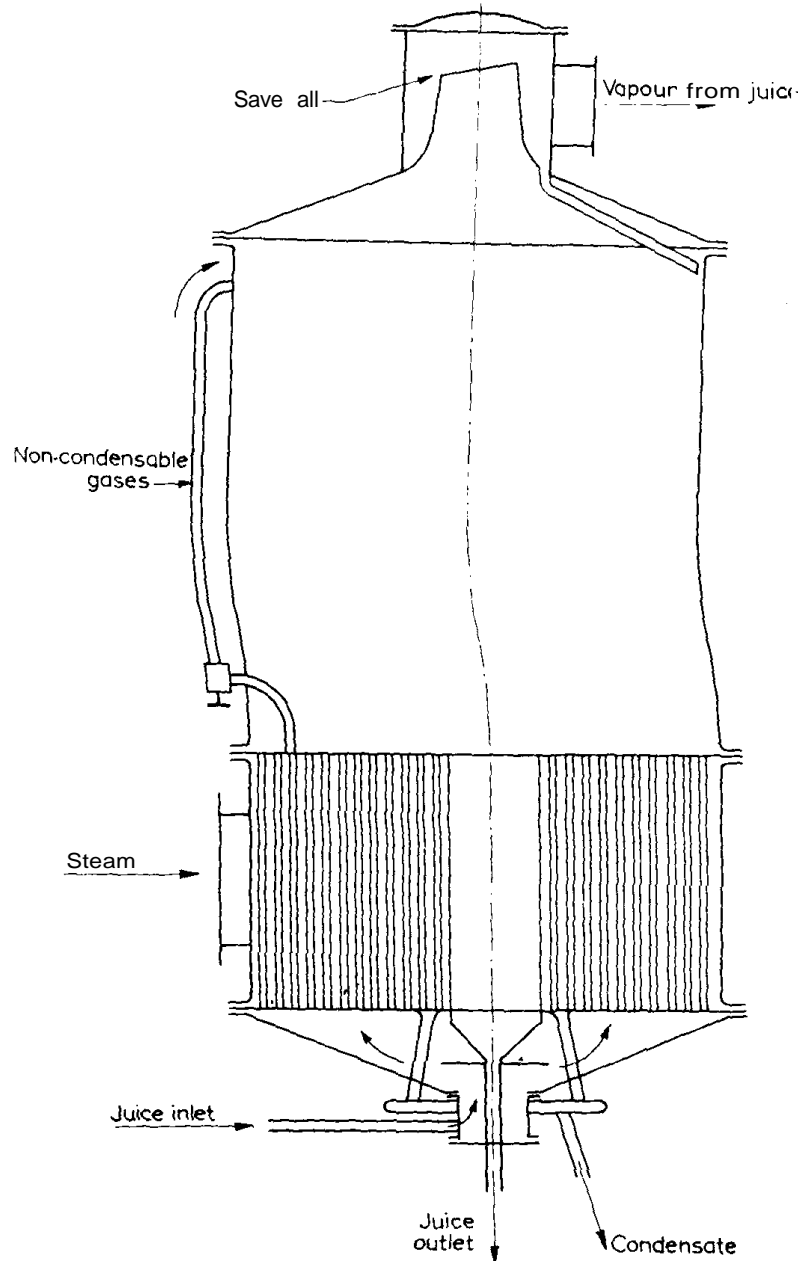


Fig. 32.3. Evaporation vessel.

The commonest evaporators encountered in practice are triple, quadruple and quintuple effects.

The evaporator vessel

The standard multiple-effect evaporator consists of a vertical cylinder, built onto the tubular calandria across which the heat exchange takes place (Fig. 32.3).

This cylindrical body terminates at the top in a "save-all", the object of which is to separate the liquid droplets which may be entrained with the vapour from the juice.

Previously the evaporator bodies were always fabricated in cast iron. More recently,

fabrication in steel plate is becoming more and more common. This permits of evaporator bodies which are less brittle, lighter and less expensive.

Height of the vessel. On examining a sectional view of an evaporator (Fig. 32.3), one is immediately struck by the large space lost above the tubular calandria. This space represents the greater part of the volume taken up by the equipment. However, it does not serve for juice, the level of which should not go above the upper tube plate. Its object is to diminish risk of entrainment of droplets of juice projected by the boiling. There is some reason for the customary saying "the best entrainment separator is a high vessel".

Tromp¹² quotes an American view that the height of the cylindrical body, above the calandria, should be 1.5 times to twice the length of the tubes. It is wise to specify at least twice; moreover, Tromp¹³ later recommended 2\ times. In Europe, a minimum of 3.6 - 4.0 m (12 - 13 ft.) is adopted.

Diameter. The diameter of the vessel also has an important effect on entrainment. According to the experiments of Loschge¹⁴, the area for evaporation, *i.e.* the horizontal section of the vessel, should provide 1 m²/400 m³/h (1 sq.ft./1,300 cu.ft./h) of steam produced, if dry vapour is to be obtained. The velocity of vapour leaving the liquid surface would then be approximately 10 cm/s (0.3 ft./s).

This indicates that it would be necessary to provide a larger vessel as the specific volume of the vapour produced is higher, that is, when the pressure is lower.

In present designs, this rule is not followed, as it would lead to prohibitive figures for the diameters, and thus vapour velocities in the vessels reach 3.6—4.6 m/s (12—15 ft./s) in the latter vessels. Hence the necessity of providing entrainment separators or save-alls at the vapour outlets from the evaporator vessels. In any case, it is considered¹⁵ that entrainment rapidly becomes excessive from the moment when the vapour velocity in the vessel reaches 6 m/s (20 ft./s).

Sight glasses. In order to observe the working of the evaporator, and to see what is happening inside it, the evaporator is furnished with windows or sight glasses formed of thick glass enclosed between the wall of the evaporator and a brass frame, and bolted on with a soft joint between.

French manufacturers sometimes provided rectangular sight glasses superposed in a vertical line situated on the front face of the evaporator body. They forgot that a man has two eyes placed on the horizontal and not on the vertical line. Circular sight glasses, which are more convenient, are now general.

The sight glasses should be placed on two opposite vertical lines so that what is happening in the vessel is seen against a window and not against a dark background. The best arrangement is to install a lateral window at 45° from the front of the vessel in addition to sight glasses at front and back; outside this extra window will be placed a powerful light with a reflector which will light up the interior of the vessel, without dazzling the observer.

The calandria

The bore of the holes provided in the tube plates to take the tubes should be about 0.75 mm

($\frac{1}{32}$ in.) greater than the exterior diameter of the tubes.

Vertical baffles are often placed in the calandria, with the object of compelling the steam to follow a certain path. Unfortunately it is impossible to remove or replace these metal baffles, which are subject to corrosion. When they are destroyed or damaged, the steam does not follow the path originally intended for it, and the position of the incondensable-gas withdrawal pipes generally becomes unsuitable for the new steam path.

Centre well. The calandria is generally designed with a wide tube or centre well, the object of which is to return to the bottom the juice which has been projected over the top tube plate. This centre well is often utilised to collect the concentrated juice in order to remove it from one vessel to the following vessel.

The diameter of the centre well varies from $\frac{1}{4}$ to $\frac{1}{8}$ of the interior diameter of the vessel. Certain manufacturers replace the centre well by a lateral well, on annular downtake, or by a series of downtakes of small diameter distributed over the calandria.

Tubes. The tubes of the calandria were of steel or of brass. Tubes of brass or stainless steel have a much longer life.

The best brass for the tubes of a multiple effect has the composition Cu = 70%, Zn = 30%, or preferably Cu = 70%, Zn = 29%, Sn = 1%, or Cu = 64%, Zn = 36%, the latter being known under the name of alloy U-Z 36.

If the proportion of copper falls below 60%, the metal becomes subject to attack by the incondensable gases.

Brass tubes must be annealed at their ends, otherwise they will become brittle after expanding, and susceptible to split lengthwise. Tubes of stainless steel are the best, but are expensive.

Stainless steel for tubes is generally 430 (17% Cr), or AI SI 304 (18% Cr + 8% Ni).

Length of tubes. The length of the tubes in standard multiple effects varied originally from 1.2 to 1.5 m (4-5 ft.). However, manufacturers have increased the length of the tubes as far as 4.6 m (15 ft.). The evaporation rate does not gain anything from this, but for a given evaporation duty one obtains in this way vessels which take up less space and are more economical. The lengths most generally used today range from 2.25 to 4 m (7- 13 ft.).

The length of the tubes should be about 7 mm ($\frac{1}{4}$ in.) greater than the exterior distance between the tube plates. They will then project about 3 mm ($\frac{1}{8}$ in.) outside the tube plates.

Choice of lengths of tubes. Previously, multiple effects often consisted of similar vessels with identical tubes. However, for economy, vessels are sometimes adopted using tubes of the same diameter, but of lengths decreasing from the first to the last vessel, with 15–20 cm (6–8 in.) difference between one vessel and the next.

The tubes, in practice, wear or corrode mostly near their ends close to the tube plates. In this way, when they reach the stage where they have to be replaced they may then be removed from all the vessels and a length of 75 - 100 mm (3–4 in.) cut off each end of the tubes, which may then be replaced in the following vessel in each case. The only new tubes to be purchased then are those for the first vessel.

The longest tubes are kept for the first vessel because this is generally the one which requires the largest heating surface, and the one in which the juice is most fluid and in the best condition to climb up the tubes.

Diameter of tubes. The tubes of multiple effects have an interior diameter varying from 27 to 50 mm (1 – 2 in.). Their thickness varies from 1.5 to 2.5 mm (0.06 – 0.10 in.) for steel tubes and from 1.5 to 2.0 mm (0.06 – 0.08 in.) for brass tubes. The commonest dimensions are as follows:

<i>Steel:</i>	27 X 30	30 x 35	31 x 35	34 x 38	39.5 x 44.5	45 x 50	mm
<i>Brass:</i>	27 X 30	28.8 x 32	32 x 35	34.8 x 38	41.8 x 45	46 x 50	mm
<i>Stainless steel:</i>			32.6 x 35	35 x 38.1 35.6 x 38.1		47.5 x 50.8	mm

with a tendency to standardise stainless-steel tubes for cane sugar at 35.6 X 38.1 mm (1.4 X 1.5 in.); or, in terms of outside diameters for both steel and brass tubes, 32, 35, 45, 50 mm ($1\frac{1}{4}$, $1\frac{3}{8}$, $1\frac{3}{4}$, 2 in.) o.d.

Tubes of the smallest diameter (30 mm o.d.) are seldom employed, however, except in the beet industry. Those of 32 x 35 mm ($1\frac{3}{8}$ in. o.d.) are often replaced by those of 33 x 36 mm ($1\frac{7}{16}$ in. o.d.), which is a standard dimension in America and also in certain European countries.

In the beet industry multiple effects are often used with tubes of 27 x 30 mm ($1\frac{1}{4}$ in. o.d.) in all vessels from the first to the last. In the cane sugar industry, small diameters for the last vessels are avoided. Certain practical men refuse to go below 50 mm (2 in.) o.d. (which, moreover, is the maximum size) for the last two vessels. However, this dimension, previously standard, no longer corresponds to actual practice, and we would recommend the adoption of brass tubes of 34.8 x 38 mm ($1\frac{7}{16}$ in. o.d.).

With steel tubes the tendency is to employ mainly 34 x 38 mm.

Narrower tubes give theoretically a better coefficient of heat transfer, because the mean distance of particles of juice from the heating surface is smaller. They permit also of the installation of a larger heating surface in a vessel of a given diameter. On the other hand, they are more difficult to clean by hand or by mechanical cleaner, because it is necessary to pass the cleaning tool through a greater number of tubes. When small tubes are employed, it is of advantage to clean the vessels by means of acid rather than mechanical descalers.

Claassen¹⁶ estimates that tubes of small diameter work better at low evaporation rates, whereas larger tubes, offering more space to the mounting bubbles, work better at high rates.

The choice of diameter of tubes for a multiple effect, however, is not of prime importance.

What is important, in our opinion, is to adopt for the whole factory a standard external diameter of tubes, 34.8 x 38 mm ($1\frac{1}{2}$ in. o.d.), for example, and to use this for the heaters as well as the multiple effect.

Interdependence of diameter and length of tubes. Moreover, diameter and length of tubes are not independent of each other. The tubes actually permit of a better climbing film of juice as they become narrower, since the climbing-film effect depends on the relationship between heating surface of the tube and its interior cross-section.

This consideration limits the length permissible for each diameter. Table 32.5 gives the order of size of these dimensions.

TABLE 32.5

MAXIMUM LENGTH OF EVAPORATOR TUBES FOR DIFFERENT DIAMETERS

<i>Diameter (mm)</i>	<i>Maximum length</i>	
	<i>(m)</i>	<i>(ft.)</i>
50 (2 in. o.d.)	2.5	8
38	3.5	11
35 (1 $\frac{3}{8}$ in o.d.)	4.0	13
30	4.50	15

In the cane sugar factory it would be wise to keep definitely below these maxima.

However, it is desirable to take into account the evaporation rates and the brix of the juice, in other words, of the place of the vessel in the set. From this point of view, it is desirable to supplement the preceding table by Table 32.6, considering only the present standard diameter of 38 mm, and replacing the maximal length of tube by the assumed optimal length, taking into account all factors, including cost.

TABLE 32.6

LENGTHS RECOMMENDED FOR TUBES OF THE MULTIPLE EFFECT

<i>Tube length (optimal)</i>				
<i>1st vessel</i>	<i>2nd vessel</i>	<i>3rd vessel</i>	<i>4th vessel</i>	<i>5th vessel</i>
4.0 m (13 ft.)	3.5 m (11 ft.)	3.0 m (10 ft.)	2.5 m (8 ft.)	2.25 m (7.5 ft.)

By optimal length, we understand the best compromise between cost and floor space involved for a vessel on one hand, and the evaporation rate which can be obtained in good average regular operation, on the other hand. The evaporation rate in fact tends to fall as the length of tubes increases, for a given design of vessel and a given tube diameter.

Actually this optimal length decreases with the place of the vessel in the set, since the film effect is more marked, as the juice is less viscous and the vapour at higher temperature; hence the figures in Table 32.6.

Arrangement of the tubes. The layout of the tubes in the tube plates generally follows a staggered arrangement (Fig. 32.4).

This arrangement permits the greatest number of tubes to be accommodated per unit area of the plates, for a given distance between the tubes.

The tubes are thus arranged in straight lines in three different directions at 120° to one another (Fig. 32.5).

Since this arrangement is that which permits the maximum density of tubes, what relation-

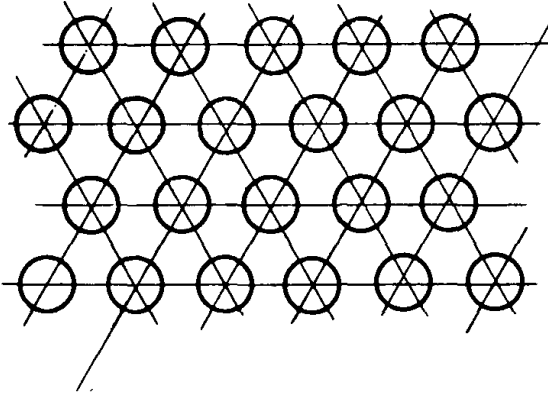


Fig. 32.4. Arrangement of tubes.

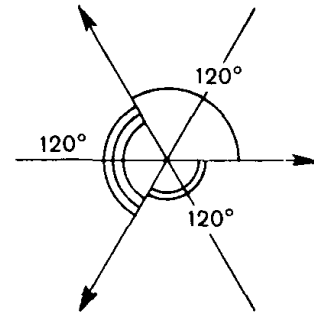


Fig. 32.5. Layout for a staggered arrangement.

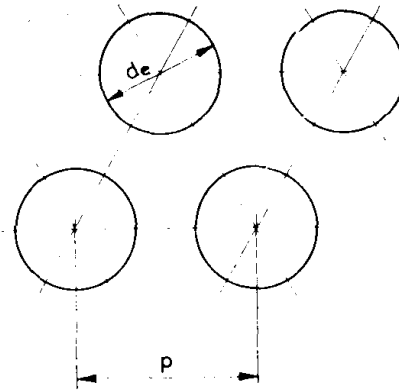


Fig. 32.6. Tubes in staggered arrangement.

ship should be observed between the exterior diameter d_e of the tubes and the distance p between the tubes, centre to centre?

This distance p between the centres of two adjacent tubes is called the "pitch" of the tubes. If we use s_e for the exterior cross-section of a tube and a for the corresponding area of the tubed portion of the tube plate, we see readily that we have (Fig. 32.6):

$$\frac{s_e}{a} = \frac{\frac{\pi d_e^2}{4}}{\frac{\sqrt{3}}{2} \cdot p^2} = \frac{\pi}{2\sqrt{3}} \cdot \frac{d_e^2}{p^2} \quad (32.10)$$

If we designate by k_e the ratio of the area of the holes to the total area of the plane, we shall then have:

$$\frac{p^2}{d_e^2} = \frac{\pi}{2k_e \sqrt{3}} = \frac{0.907}{k_e} \quad (32.11)$$

hence:

$$p = 0.952 \frac{d_e}{\sqrt{k_e}} \quad (32.12)$$

The coefficient k_e increases slightly with the diameter of the tubes. It varies from 0.45 to 0.55 in modern vessels. If we adopt 0.5 we should have:

$$p = 1.35 d_e \quad (32.13)$$

The minimum width of the strip of metal separating two neighbouring holes is then approximately one-third of the diameter of the tubes. This width is termed the "ligament".

Area of juice passage. The coefficient k_e corresponds to the external area of the tubes, and consequently to the area of the holes in the plate. If we are interested in the area of the passage offered to the juice circulating through the tubes, it is necessary to consider the internal cross-section of the tubes. We have then:

$$k_i = \frac{s_i}{\sigma} = k_e \frac{s_i}{s_e} \quad (32.14)$$

k_i = ratio of the cross-section of juice passage in the tubes to the corresponding area of the plate, in the portions which are entirely used for tubes

s_i = interior cross-section of one tube.

The ratio s_i/s_e has a value 0.878 for tubes of 35.6 x 38 mm.

However, these coefficients k_e and k_i apply only to the portions of the tube plate which carry the tubes. If it is desired to compare the total cross-section of juice passages of all the tubes, s_i , to the total surface Σ of the tube plate, it is necessary to modify this latter figure by a second coefficient α which takes into account the space lost by the tie rods and the steam lanes, and the space lost around the border of the calandria, and also by a third coefficient β which takes into account the centre well. We have then:

$$s_i = k_i \alpha \beta \Sigma \quad (32.15)$$

α is of the order of 0.98

β depends entirely on the centre well: $\beta \approx 1 - D^2/\Delta^2$

D = diameter of centre well

Δ = interior diameter of the vessel.

The total coefficient $k = k_i \alpha \beta$, which varies from 0.33 to 0.38 (mean 0.36), measures the ratio between the cross-section of passages offered to the juice by the small tubes and the total cross-section of the vessel.

CIRCULATION OF STEAM AND VAPOUR

The steam or vapour is admitted to the calandria by one, two or more inlets. Generally, only one vapour inlet is used in vessels of less than 3 m (10 ft.) diameter; two are provided for vessels of 3 m or more diameter, sometimes four for very large vessels, in such a way as to supply effectively all the tubes; some would otherwise be too far away from a single entry.

In the case of a very high calandria, each steam entry may be divided between two openings, placed at $\frac{1}{4}$ and $\frac{1}{2}$ of the height of the tubes.

At the point furthest from the inlet or the inlets, the pipes for withdrawal of incondensable gases are located.

In order to facilitate access of vapour to the furthest portions of the calandria, steam lanes are often provided between the tubes; these are obtained by simply leaving out rows of tubes over part of their length (Fig. 32.7).

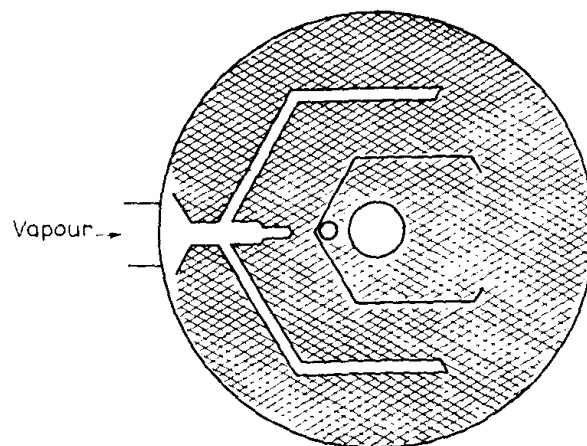


Fig. 32.7. Example of vapour lanes.

Calculation of vapour pipes

From considerations of economy on the one hand, and of pressure drops and risks of entrainment on the other hand, the vapour pipes are calculated so as to give the approximate vapour velocities shown in Table 32.7:

TABLE 32.7

VAPOUR VELOCITIES RECOMMENDED, IN m/s (ft./s)

	<i>Triple</i>		<i>Quadruple</i>		<i>Quintuple</i>	
	(m/s)	(ft./s)	(m/s)	(ft./s)	(m/s)	(ft./s)
Exhaust steam to 1st effect	25-30	80-100	25-30	80-100	25-30	80-100
Vapour from 1st effect	30-35	100-115	30-35	100-115	30-35	100-115
Vapour from 2nd effect	40-45	130-150	35-40	115-130	30-35	100-115
Vapour from 3rd effect	50-60	165-200	40-45	130-150	35-40	115-130
Vapour from 4th effect			50-60	165-200	40-45	130-150
Vapour from 5th effect					50-60	165-200

Vapour velocities from the last effect going to the condenser may be higher than 60 m/s (200 ft./s) on condition that they do not exceed 75 m/s (250 ft./s) and that all precautions are taken against risks of entrainment.

Once these velocities have been chosen, the calculation is carried out as follows:

(a) Take the vapour output by weight to be allowed for, say P . Increase this by 10% so as to allow for possible variations: say P' .

(b) Take from the table for saturated steam (Table 41.1, p. 1034) the specific weight d corresponding to the vapour temperature. From this calculate the volume of vapour to be passed per hour, say Q . Hence the volume per second, say q .

(c) From the velocity V which has been fixed, the cross-section S of the pipe is calculated. Hence its minimum diameter D , which in rounding off to the nearest even size is naturally chosen on the large side.

Example. To calculate the vapour pipe from the body of the first vessel to the calandria of the second. Given:

Weight of vapour to be supplied to the second vessel:	$P = 20,000$ kg/h (44,150 lb./h)
Temperature of this vapour:	? = 103°C (217.5°F)
Vapour velocity chosen:	$V = 25$ m/s (82 ft./s)

We have:

$$P' = 1.1 p = 22,000 \text{ kg/h (48,550 lb./h)}$$

The specific volume at 103°C is 1.515 m³/kg (24.2 cu.ft./lb.). The volume of vapour to be handled is therefore:

$$Q = 22,000 \times 1.515 = 33,330 \text{ m}^3/\text{h}$$

or:

$$q = 33,330/3,600 = 9.258 \text{ m}^3/\text{s (327 cu.ft./s)}$$

To obtain a velocity of 25 m/s, the cross-section required will be:

$$S = \pi D^2/4 = 9258/250 = 37 \text{ dm}^2 \text{ (3.98 sq.ft.)}$$

Hence:

$$D^2 = 37 \times 4/\pi = 47.15 \text{ dm}^2 \text{ (5 sq.ft.)}$$

and:

$$D = \sqrt{47.15} = 6.87 \text{ dm (2.24 ft.)}$$

We shall take:

$$D = 700 \text{ mm (27.5 in.)}$$

CONDENSATES

The heat transmitted from the vapour to the juice across the tubular calandria corresponds to the latent heat of vaporisation of the steam. All the steam used for heating is therefore transformed into condensate in the calandria. Thus it is necessary to evacuate or remove a weight of such condensate equal to the weight of steam used.

This removal of condensate is effected by drains placed at the lower part of the calandria (Fig. 32.8). These drains should be distributed in the lower tube plate so as to provide one drain for approximately each 3 m² (30 sq.ft.) of cross-section of the vessel.

The internal cross-section of these drainage pipes should be designed so as to give a velocity of flow of the condensate of approximately 0.5–0.6 m/s (2 ft./s). In the pipe T which is connected to the drains and the collecting pipe, this velocity could be increased to 0.8 m/s (2.5 ft./s).

The drains should be connected to the calandria at a by a small cone the object of which is to increase the diameter of the pipe at the point which determines the flow through the drain.

Extraction of condensates

The extraction of these condensates from the calandria may be made in various ways. According to the pressure ruling in the calandria we shall distinguish:

	<i>Method of extraction</i>
(a) Calandrias under pressure	Steam trap
	Steam trap
	Pump
(b) Calandrias under pressure or under vacuum	Grain
	Flash vessel

A. Steam trap. When the calandria is under a pressure above atmospheric, which is the general case for the first vessel, the condensate may be allowed to flow out by gravity. However, since it is necessary that the pipe should not become empty and allow steam to escape, it is necessary to insert a device for separation and control, known as a "steam trap".

This is a small vessel through which the condensate passes, which is fitted with a float actuating an outlet valve which does not open until the water attains a certain level in the vessel. When the water is flowing continuously, the float progressively opens the valve in proportion as the water level rises, and reaches a position of equilibrium corresponding to the rate of flow.

Thermal steam traps are now used, functioning by the action of the very slight difference

in temperature between the condensate entering and the vessel in which it is received (Figs. 32.9 and 32.10).

B. Condensate pump. With the aid of a pump, condensate may be extracted from a calandria under any vacuum provided the three following conditions are fulfilled:

(7) It is necessary that the pump should be capable of delivering to a height equal to the geometric height of the delivery increased by the difference in pressure between atmosphere and the calandria concerned.

(2) Since some leakage is bound to occur at the pump, it is necessary to provide it with a small pressure-equalisation pipe putting the admission valve of the body of the pump in communication with the top of the calandria. Otherwise (a) air, entering through leaks in the pump, would be entrained with the water in the suction pipe and would hinder the flow of water to the pump; (b) vapour formed in the pump from the hot water would accumulate and form a vapour lock.

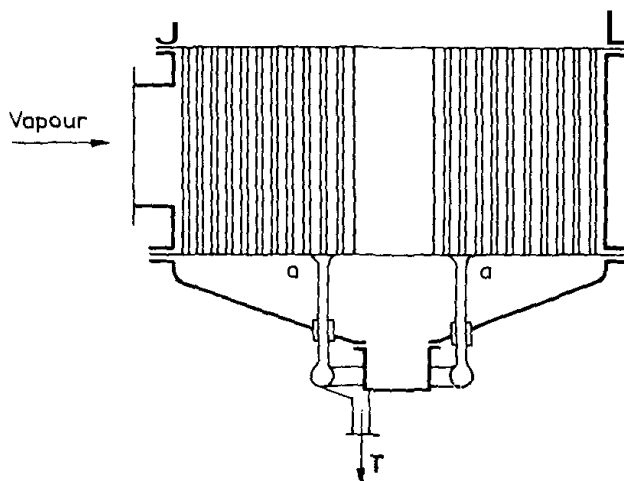


Fig. 32.8. Drainage of condensates.

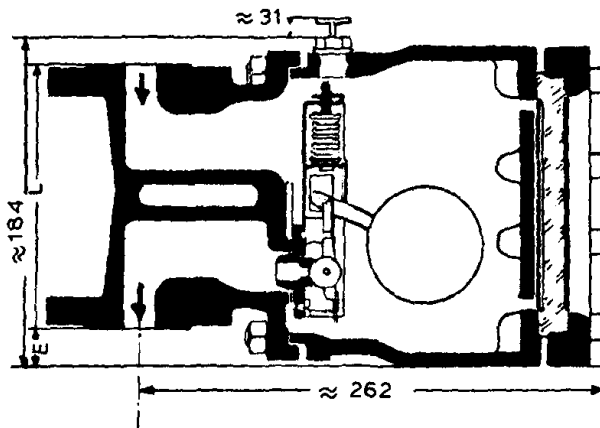


Fig. 32.9. Vertical steam trap with float (Gestra Una TR 11).

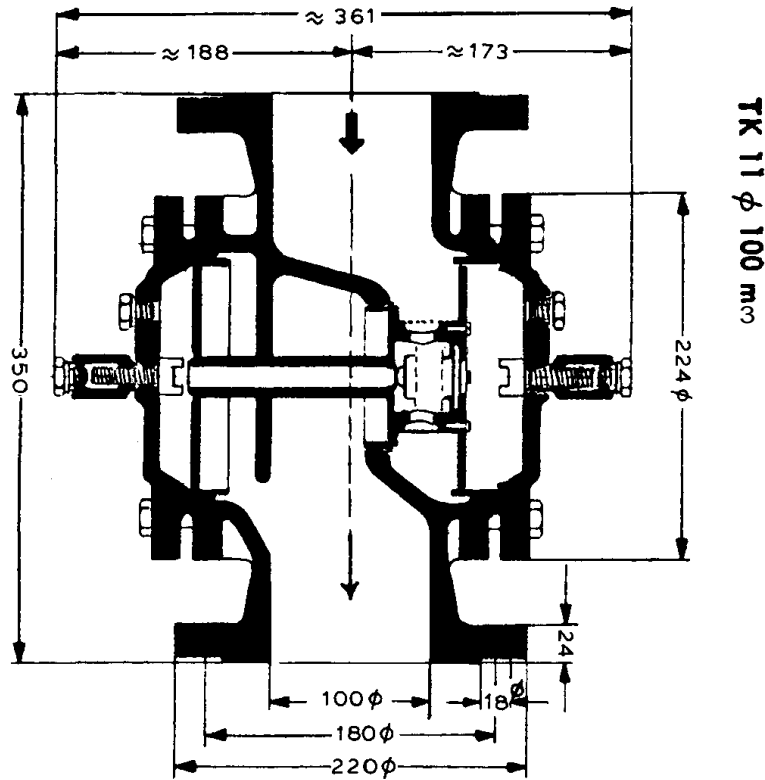


Fig. 32.10. Horizontal thermal steam trap (Gestra TR 11).

(3) As for steam traps, it is necessary to provide a pump three or four times greater in capacity than would seem theoretically necessary.

Types of pump. Centrifugal pumps are now the only type used. It is of advantage to use pumps specially designed for working with suction under vacuum, which have the glands in communication with the delivery side of the pump. This avoids or reduces entry of air, which otherwise would inevitably occur at these points.

Condensate suction pipes. The suction pipes taking the condensates from the calandria to the pump should fulfil the following conditions:

- (a) They should be as short as possible, to avoid retarding, by an unnecessary loss of heat, a flow which is already difficult.
- (b) For the same reason they should have as few bends as possible. Bends which cannot be avoided should be as open as possible and of the highest practicable radius.
- (c) They should have as few joints as possible to reduce risks of air leakage.
- (d) They should have a cross-section corresponding to the condition already indicated (p. 516: Velocity of water at most equal to 0.8 m/s).
- (e) The horizontal portions of the pipes should drain towards the pump.

The pressure-equalisation pipes should fulfil the same conditions except for their area, which should be about 0.04 of the cross-section of the suction pipe. That is, their diameter should be about one-fifth of the diameter of the water pipe.

C. Siphon. The two methods of extraction of condensates from calandrias under vacuum, which we have just discussed, do not permit of circulation of the condensates from one vessel to another. The two methods which we will now discuss have on the contrary the object of allowing the condensates to be circulated from vessel to vessel.

Flash. This circulation offers a valuable advantage from the point of view of heat economy.

If we extract condensate directly from each vessel it takes with it all its sensible heat content. If on the other hand we arrange that the water may be passed from one vessel to the following calandria, this water, which in the preceding calandria is at the maximum temperature compatible with the liquid state at the pressure in this calandria, will then pass suddenly to a lower pressure. It thus produces a spontaneous evaporation or "flash", which takes place almost explosively by reason of the physical incompatibility of the conditions of pressure and temperature. The quantity of water evaporated will correspond exactly to the quantity of heat given up.

Thus a small quantity of vapour has been recovered, which will then join with the vapour coming from the juice in the preceding vessel and so will increase the available heat in the next calandria. Further, if the condensate is thus circulated from one calandria to the next throughout the set, the quantities will increase, so increasing the amount of steam available.

Gain due to circulation of condensates. The benefit resulting in this way from the flash produced by circulation of condensates from calandria to calandria is given by the calculation on p. 561, for a quadruple effect without vapour bleeding. The evaporation obtained from unit weight of steam introduced to the calandria of the first effect is given in Table 32.8.

TABLE 32.8

QUADRUPLE-EFFECT GAIN WITH CIRCULATION

	<i>Evaporation</i> kg/kg steam		
	<i>Without circulation</i>	<i>With circulation</i>	<i>Gain %</i>
1st effect	0.923	0.923	0
2nd effect	0.958	0.972	2
3rd effect	1.002	1.047	4.5
4th effect	1.040	1.162	11.2
	3.923	4.104	4.6

Siphons. The difficulty encountered in passing condensate from one calandria into the following one was due to the difference in pressure between these two enclosures.

This difficulty has been overcome by connecting the two calandrias by medium of a U-tube, or "invert siphon" (Fig. 32.11). In a way, the difference existing between the pressures in the two calandrias is automatically balanced by the difference in level H which is established between the two columns of liquid in the legs of the U-tube.

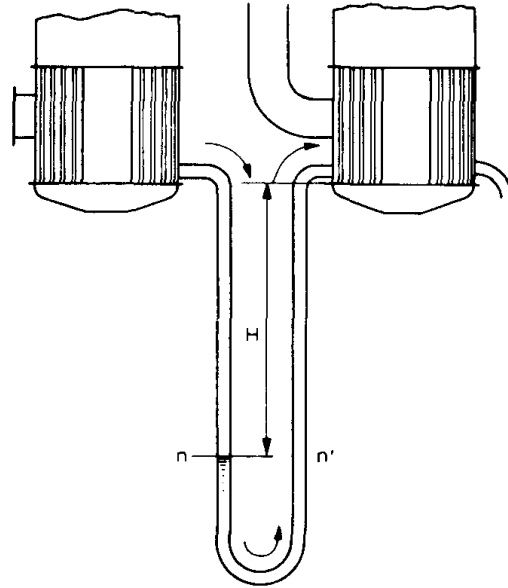


Fig. 32.11. Siphon.

Length. Theoretically, if there is 0.3 kg/cm^2 (4 p.s.i.) difference in pressure between the two calandrias, a siphon will be required with a length of about 3 m (9 ft.).

In practice it is necessary to allow a margin of safety of at least 50%, and 100% if possible:

(a) By reason of fluctuations in the pressures: if the siphon were too short, even by a very small amount, for the difference in pressure existing at a given instant, the upstream pressure would push the liquid level past the bend of the U, and the siphon would fail immediately.

(b) On account of flash: the water rising in the second branch of the U, as soon as it has passed the level n' (Fig. 32.11), is subject to a pressure lower than that of the preceding vessel. Since it has just condensed, it is practically at the temperature of ebullition corresponding to that pressure. It will therefore begin to boil, and more and more vigorously as it mounts in the tube, owing to decreasing pressure. Consequently, it is not a column of water which balances the difference in pressure but a column formed of a mixture of water and vapour, of density much lower than 1.

(This statement appears unscientific and contrary to the hydrostatic law relative to communicating vessels. It is obvious, however, that this law is not applicable when bubbles predominate. The bubbles will finally destroy the liquid column. To convince oneself of this it is only necessary to compare the levels of juice in the interior of an evaporator and in its gauge glass (*cf.* p. 613), "climbing-film effect").

Diameter. For the same reason, and to avoid a climbing-film effect in the rising branch of the siphon, it is necessary to use a large diameter for this branch.

The diameters of the descending and ascending branches of the siphon will be chosen so that the velocity of flow of the water, assumed to be free of vapour bubbles, will not exceed:

Descending branch:	1.2 m/s (4 ft./s)
Rising branch:	0.6 m/s (2 ft./s)

If these maxima are observed, a margin of 50% in the length of the siphon will be sufficient. If it is installed, as is often found, of the same diameter for both branches, corresponding to a velocity of 1–1.5 m/s (3–5 ft./s), it will be necessary to increase the safety margin to 100%, and there will still be numerous causes of trouble. Since it is the height of the siphon which generally presents difficulty in installation and support of the siphon, it is of great advantage to choose the former solution.

D. Flash pots. Siphons are economical and very simple, but they have the disadvantage of requiring, generally, a greater height than that available between the bottom of the calandrias and the floor level. Multiple effects are generally located on the first-floor level, that is, at a height of 4–4.5 m (13–15 ft.) above ground level, and the bottom of the calandrias is situated appreciably lower.

It is preferable to install flash pots or expansion vessels (Fig. 32.12, 32.13). These are closed vessels placed below the evaporators, in which flash or self-evaporation takes place. Each of these is connected to:

- (a) The previous flash pot, which delivers to it condensate which has already undergone flash
- (b) The calandria from which it receives condensates
- (c) The top of the following calandria, which fixes the pressure in the flash pot and to which it supplies the flash vapour proceeding from these two sources of condensate
- (d) The following flash pot, to which it delivers its condensates in turn.

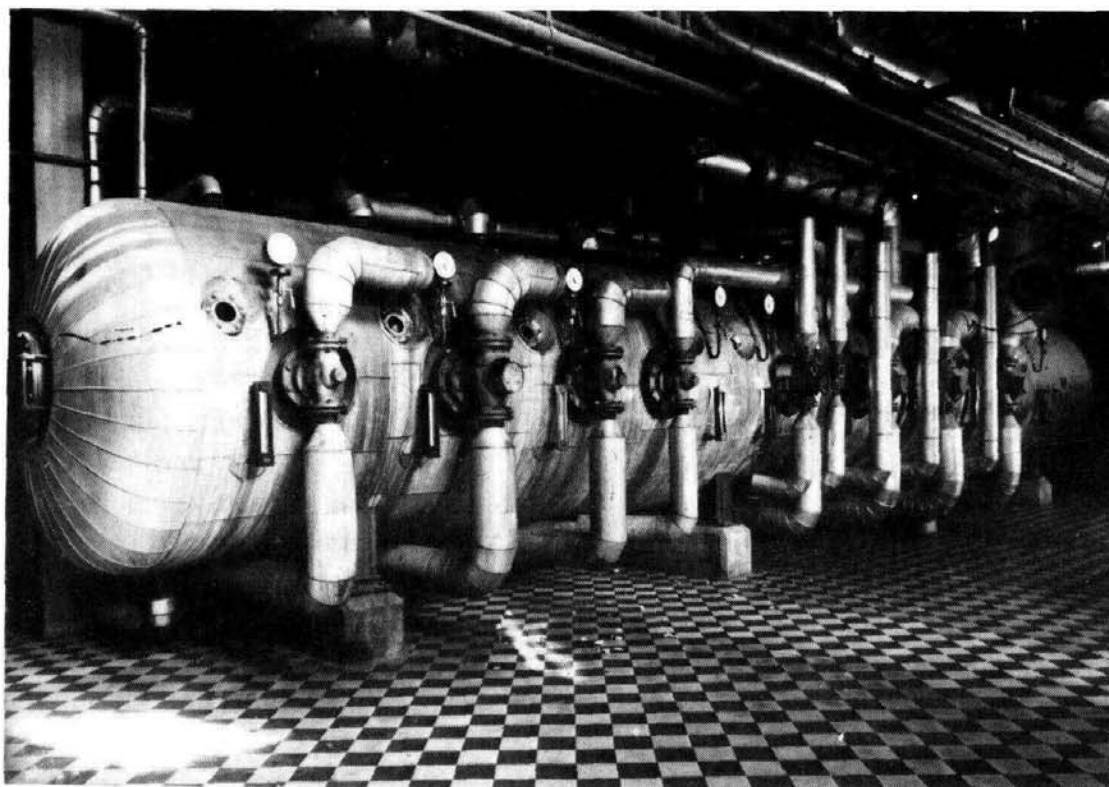
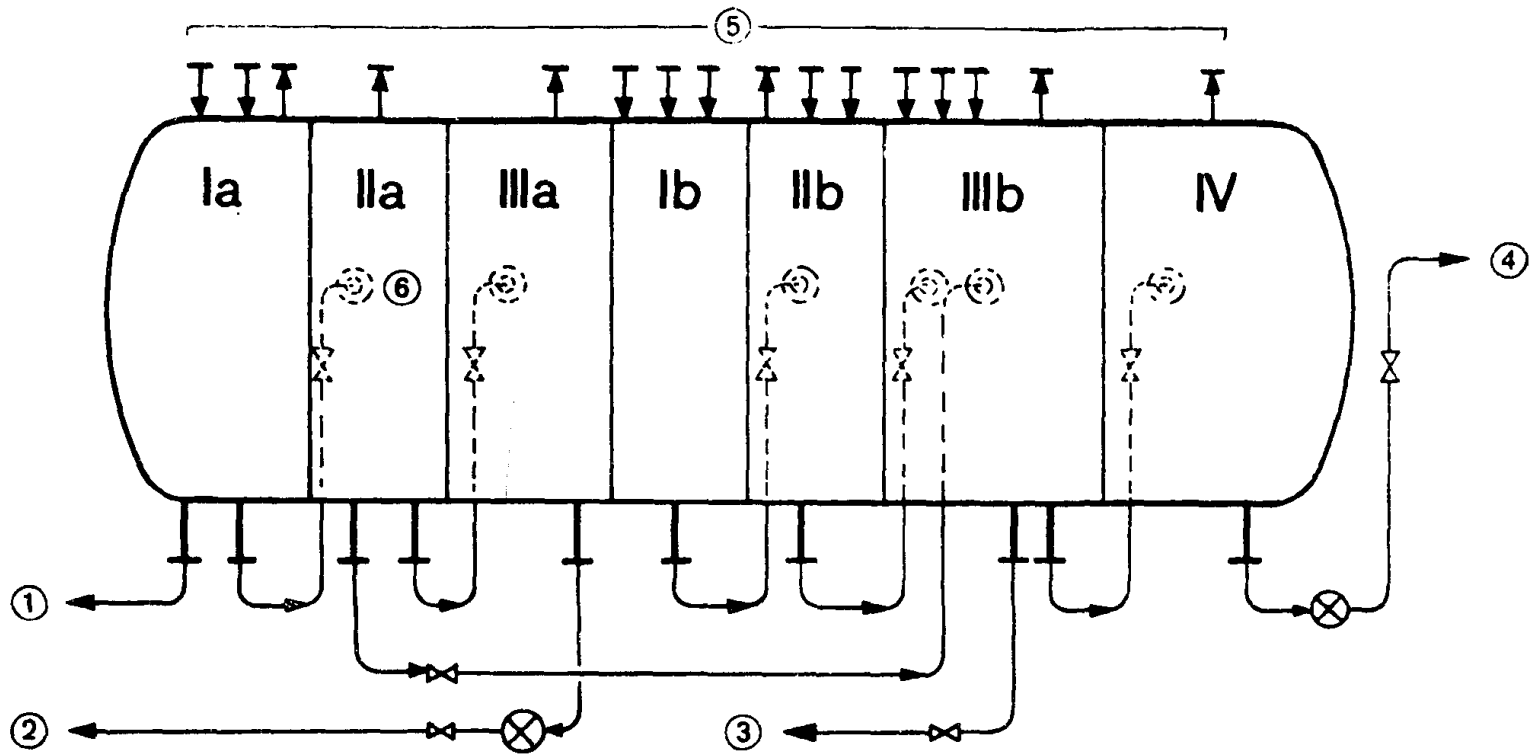


Fig. 32.12. Flash pots.



Condensate Flash Tank

- ⊕ injection water for desuperheater
- ⊕ boiler feed water
- ⊕ emergency outlet
- ⊕ to hot water line
- ⊕ condensate feed and flash vapor lines
- ⊕ float valves

- a condensate from sugar of evaporator stages I and II
- b condensate of evaporator stages III and IV as well as of vapor-heated preheaters

Fig. 32.13. Condensate flash tank (B.M.A.).

It is of advantage to provide these flash pots with sight glasses in two opposing sides of the vessel, which permit an observer to see what is happening inside.

The siphon between the calandria and the flash pot may be omitted if a float-operated regulating device is inserted in the line to regulate the flow.

When a general condensate receiver is used, **Webre**¹⁷ very wisely recommends that the various condensate pipes should descend vertically near the vessel, and enter it at the bottom by means of a T-piece and an elbow, provided with a valve. The lower end of the T will then discharge into a pipe going to the drain, also provided with a valve. Thus, at any moment, the condensate from any vessel may be separated and sent to the drain until the leak detected can be attended to.

The water level in the various compartments of these vessels is controlled by regulators: automatic butterfly valves of the Maisoneilan, Schumberger, or Camflex type. This water is evacuated by centrifugal pumps. If the vessels are just below the evaporators, they may be of single-stage type. If there is a large difference in level, for example, evaporators on a high stage and condensate vessels at ground level, multistage pumps will be used which handle a greater range of pressure; the suction should always be positive (net positive suction head), but can be smaller. These pumps are obviously more expensive. Generally, three pumps are used, the respective roles of which will be discussed later (p. 524).

Control of condensates. It happens at times that a leak arises in a tube of a heater or of the multiple effects, or at a joint in a tube plate. If it is the juice which is under the lower pressure (the commonest case):

(a) If the leak occurs in the upper part of the calandria, the vapour will pass directly from the calandria into the vapour space of the evaporator vessel. This will not be very serious. In certain cases this leak could even be considered as a supplementary outlet for incondensable gases.

(b) If the leak occurs in the lower part of the calandria, condensate will pass into the juice, increasing the evaporation work required, or diluting the syrup.

In order to avoid such leaks, which are difficult to detect, it is necessary to test the calandrias frequently.

If it is the vapour which is at a lower pressure (this is generally the case with juice heaters), the juice will penetrate into the steam space, and will become mixed with the condensate.

In the same way abnormal entrainment will introduce juice into the condensates.

These two latter cases are much more serious, because the water which is thus polluted serves as boiler feed.

When the quantity of juice introduced into the condensates in this way is appreciable, it may be detected by a characteristic odour. But by this time it may have already done some damage. It is also advisable to test the condensates frequently and promptly, by the reaction with α -naphthol. As soon as a trace of the characteristic violet ring is detected, the apparatus responsible should be identified and isolated and, if possible, should be shut down immediately, or else the condensate from it should be sent to the drain until the leak can be repaired.

Automatic control. A sample is taken continuously, receives the required quantity of a-

naphthol, and passes through a colorimeter which sets off an alarm as soon as a violet colour is shown. This control should be carried out on the condensates as well as on the hot water going to the cooling tower, since entrainment is more likely to occur in the last vessel than in the preceding ones.

Composition of condensates

Contrary to what some people imagine, the condensates from evaporation of cane juices are not pure distilled water. Evaporation indeed constitutes a type of distillation and, with the water, there distil over various volatile components of the juice, the boiling points of which are in the neighbourhood of that of water, or lower. Honig¹⁸ indicates in particular: oxygen, CO₂, SO₂, NH₃, organic acids, aldehydes, methyl and other alcohols. The pH, measured at 25°C, varies from 5.5 to 9.

Temperature of condensates

It is generally assumed that the condensates leaving a calandria are at the temperature of the vapour used in that calandria. Actually, the condensates cool down slightly in running along the tubes, and their temperature may be given approximately by¹⁹:

$$t_c = t_v - 0.4 (t_v - t) \quad (32.16)$$

t_c = temperature of the condensates leaving the calandria

t_v = temperature of the vapour used for heating the calandria

t = temperature of the juice in the vessel.

Utilisation of the various condensates

When the condensates furnished by the different vessels are kept separate they are generally utilised as follows:

Condensate from the 1st vessel:	Boiler feed
Condensate from the 2nd vessel:	Make-up for boiler feed
Condensate from later vessels:	{ Imbibition Washing filter-cake, dilution of molasses, etc.

Condensates from live or exhaust steam, that is, those coming from defecators, from pans, and from juice heaters working on exhaust steam, are sent with the condensates from the first effect to a feed-water tank supplying under a positive head the feed pump for the boilers.

The condensates from the second effect are sent to a neighbouring compartment of this tank which delivers into the main tank by a float-operated valve. In this way, one sends to the boilers only the proportion of second-effect Condensate which is strictly necessary. The excess overflows into the tank containing condensate from the later vessels.

These precautions have the object of avoiding as far as possible all risk of introducing sugar to the boilers by polluted condensates, and are based on the following considerations:

(a) The boiler water describes a closed cycle: feed-water tank — boilers — steam → engines or turbines — exhaust main — calandrias — feed-water tank. It should therefore

have no need of make-up water. But there are losses in this circuit: steam to the derrick, steam used for cleaning, traps, leaks, etc. It is these losses which make it necessary to make up with water taken from outside this circuit.

(b) It is for this reason that recourse is had to condensates from the second vessel, formed by condensation of vapour from the juice of the first vessel, and consequently originating in the cane itself, and not from the piping bringing fresh water supply to the factory.

(c) The second vessel is chosen in preference to the later ones because it furnishes water at higher temperatures, and especially because the risk of entrainment increases from the first to the last vessel.

(d) On the other hand, the second vessel should normally be ample to furnish all the make-up necessary. If a thermo-compressor is fitted to the first effect, it will be sufficient to produce the make-up, and the condensates from the second effect may be used elsewhere.

Conclusion. In the general case the best disposition of condensates from the multiple effect is the following:

(a) Profit from self-evaporation by installing flash pots.

(b) Take the condensates from the first effect to the first flash pot and so to the feed-water tank. Their temperature is then slightly below that indicated by the pressure gauge of the second effect. Hence less escape of steam. This flash pot is not connected to the second.

(c) Take from the second flash pot, if necessary, the make-up required for the boiler feed-water tank.

(d) Circulate all the other condensates from one pot to the next.

(e) Extract the condensates from the last pot, and send them to the hot-water tank.

Thus all the condensates may be sent to a single tank, with several compartments, communicating with the last effect. The piping is simpler, the saving in heat remains the same, but the last vessel itself must be capable of absorbing an excess of vapour which is not negligible.

INCONDENSABLE GASES

In setting a multiple effect in operation, it is necessary to raise the desired vacuum in the various vessels.

On the other hand, in normal operation, the vapour which arrives in each calandria introduces with it air and foreign gases. These originate from:

(a) The gases dissolved in the juice, which are released on boiling. With beet products, ammonia is given off. With cane it is mainly air.

(b) Air brought in by the exhaust steam. In general it contains very little. (This air is introduced by the make-up water which carries air in solution).

(c) Air which enters by leaks at the joints of the calandrias and of the vessels at valves, sight glasses, etc. This is by far the largest quantity.

It follows that:

(1) These foreign gases or "incondensables" consist almost entirely of air.

(2) The quantity of these incondensable gases is very small for the vessels heated by ex-

haust steam (air *(b)*), that is, in the multiple effects for the first vessel, but much more important for the vessels heated by vapour from juice (air *(a)*) such as the second vessel, and particularly for the vessels under vacuum (air *(a)* + air *(c)*).

If they are not removed as quickly as they enter, these gases would accumulate in the calandria, and would quickly stop all evaporation; in a matter of minutes for a third or a fourth effect, for example.

A small proportion of air is sufficient to cause the vapour temperature in the calandria to fall below that of the juice which it is supposed to heat. The difference between these two temperatures is actually only a few degrees. Now the pressure of a mixture of air and steam is equal to the sum of the partial pressures of the air and the steam; and the temperature of the vapour or steam is that corresponding to its partial pressure.

The partial pressure of the gases being proportional to the relative volume which they occupy, it is necessary in order to decrease the partial pressure of the air, to remove the gases as quickly and as completely as possible. Hence the necessity of provision for removing such gases from the calandria. Hence the equal importance of a good distribution of the gas-withdrawal pipes.

Gas withdrawal

For calandrias heated by exhaust steam, such as that of the first effect, and those of vacuum pans, the steam pressure is higher than atmospheric pressure, and the quantity of incondensable gas is also very small. It will suffice to provide a very small pipe leading from the calandria, at the point furthest from the steam entry and discharging to atmosphere.

This will be regulated in such a way as to allow a small wisp of steam to escape continuously (Fig. 32.14)

For calandrias heated by vapour from the juice, it will be necessary to install several withdrawal pipes, at a certain number of points judiciously chosen. These points should be

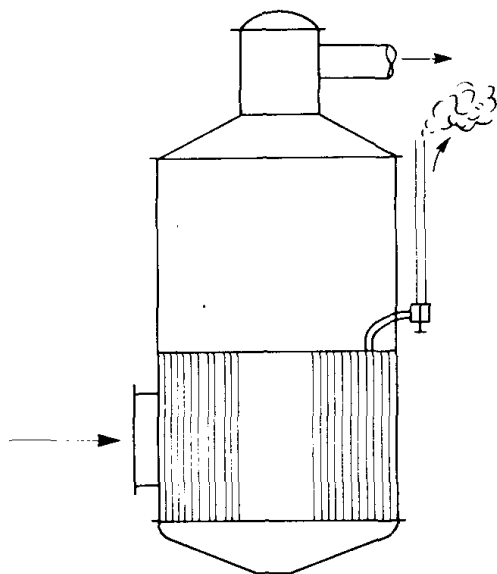


Fig. 32.14. Incondensable-gas pipe for first vessel.

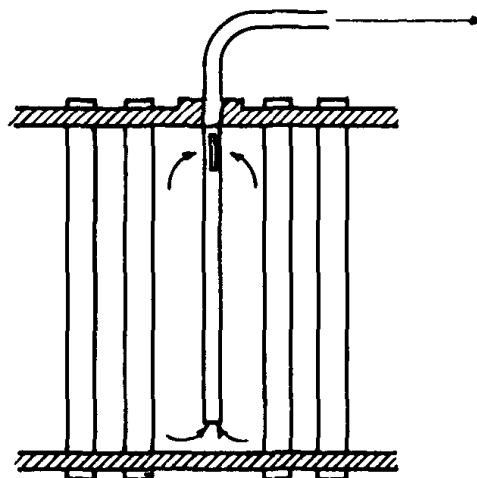


Fig. 32.15. Withdrawal of incondensables.

located in the places which are most difficult for the vapour to reach, and where it is ascertained or supposed that air pockets will form.

Webre and Robinson²⁰ indicate a simple method of detecting pockets of air in a calandria: this is to watch through the sight glasses of the evaporator vessel. Active zones will be marked by a violent ebullition, since the tubes situated in air pockets will not project juice, or will project it only slightly. They give also (ref. 20, pp. 134 – 139) the location of the air pockets according to the method of circulation of the vapour in the calandria and the arrangement of the latter.

Since air is heavier than steam, it may accumulate at the bottom of the calandria. Tests have in fact shown²¹ that the proportion of incondensable gases was greater at the lower part of the calandria. To avoid this, some of the gas-withdrawal pipes, which are generally connected to the top tube plate, are prolonged inside the calandria to about 10 cm (4 in.) above the bottom tube plate. The margin of 10 cm has the object of preventing condensate from being sucked into the tube when, for any reason, the condensate does not drain away readily (Fig. 32.15).

Since these tubes may corrode, and since inspection and replacement of them is difficult, the present trend is to replace them by cocks simply attached to the exterior of the calandria, at positions furthest from the vapour entry.

To extract the air, a pressure difference is necessary. In vessels which are under vacuum, the gas pipes must therefore discharge into the vapour space of the vessel, and are passed to the exterior of the vessel, so that a regulating valve can be placed on the line and located handy to the operator (Fig. 32.16).

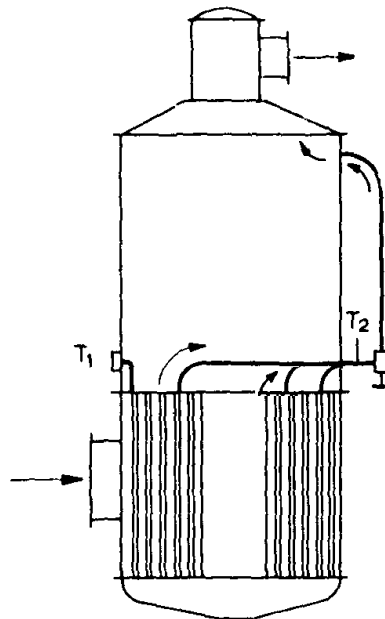


Fig. 32.16. Incondensable-gas pipe for vessel under vacuum.

Certain manufacturers take the gas pipes from all vessels directly to the condenser. This system has the disadvantages of:

(/) Rendering regulation more delicate, by increasing the pressure drop actuating flow of the gases for the first vessels, which have least need of it

(2) Losing completely the small excess of vapour which is necessarily allowed to escape with the gases.

However, it avoids the disadvantage of accumulating the incondensable gases all the way from the second to the last effect.

Dimensions of gas pipes. It must not be forgotten that the incondensable-gas pipes must remove quantities of air which, in the general case, increase from vessel to vessel.

For a quadruple of about 1,000 m² (10,800 sq.ft.) heating surface with the first vessel evacuating to the atmosphere, and the later ones from calandria to vapour space, Claassen²² recommends the following diameters for incondensable-gas pipes:

1st vessel:	10	mm	(3 in.) (to atmosphere)
2nd vessel:	13-20	mm	($\frac{1}{2}$ - $\frac{3}{4}$ in.)
3rd vessel:	20-25	mm	($\frac{3}{4}$ - 1 in.)
4th vessel:	28	mm	(1 $\frac{1}{8}$ in.)

These pipes obviously should be provided with valves which can be conveniently regulated.

Settings of valves. Air and corrosive gases should never be allowed to accumulate in the calandrias. Since there is little disadvantage in short-circuiting a few pounds of vapour from one vessel to the next, it is preferable to leave the gas valves a little too wide open rather than not enough. As these are generally valves with handwheels, their opening is generally expressed in number of turns of the handwheel. Overlooking the case of the first vessel which is regulated independently and easily seen, it is generally found that valve openings varying from about half a turn at the second vessel to 2 or 3 turns for the last are satisfactory.

Smith²³, in Hawaii, gives an interesting method of arriving at a good setting for these valves. Two thermometers are placed, one (T_1) on the calandria, the other (T_2) on the incondensable-gas pipe which leads from it, between the calandria and the valve (Fig. 32.16). If necessary, T_1 could be replaced by the thermometer in the vapour space of the preceding vessel. By reason of what we have said above (p. 526) on the partial pressures and the temperature of the mixture of air and vapour, it will be seen that since all the air passes the thermometer T_2 , this thermometer will indicate a temperature lower than that given by T_1 . If the valve is opened wide, a higher and higher proportion of vapour will pass, and the temperature T_2 will approach closer and closer to T_1 . However, experience shows that a good regulation corresponds to a difference of about 3°C (5°F) between T_1 and T_2 . Paturau²⁴ considers that 1.5°C (3°F) should be sufficient.

The same procedure may be employed with a vacuum pan, but Smith indicates in this case 3 - 6°C (5 - 10°F) as the optimal difference.

According to Claassen²⁵, good elimination of incondensables requires that there should be withdrawn with them a volume of vapour at least 30 times greater than that of the incondensables to be eliminated. Honig²⁶ estimates that a volume 50 times greater must be withdrawn.

JUICE CIRCULATION

Juice feed

There are several methods of feeding juice to the successive vessels:

- (1) From the bottom of one vessel to the top of the calandria of the following one
- (2) From bottom to bottom
- (3) From top to bottom with Chapman circulation
- (4) By lateral or annular downtake
- (5) From bottom to bottom with level regulator.

(1) From bottom of one vessel to top of the following. This system is now of historical interest only, as it has been completely abandoned. It had the disadvantages that:

(a) The juice was introduced above the tubes, whereas ebullition requires that it should rise in the tubes.

(b) Part of the juice arriving could descend by the centre well and pass close to the juice outlet leading to the following vessel, without traversing the tubes.

Now the heat-transfer coefficient depends on the brix of the juice, and decreases as the brix increases. It is therefore desirable that the average brix of the juice in the vessel should be as low as possible. This is the case when the average brix is equal to the mean between the brix of the juice entering and the brix of the juice leaving. Now, if the juice tends to short-circuit the tubes and pass directly through the vessel, the portion of the juice in the vessel which boils in the tubes will become concentrated to a brix higher than that of the juice leaving, and this vessel will then function under unfavourable conditions.

(2) From bottom to bottom. In this system (Fig. 32.17) the juice pipe is straight and short and enters the bottom of the evaporator vessel; the juice is distributed by various methods, for example, a conical flange, or a ring pipe with holes or slots, which direct the juice towards the tubes on the periphery.

The juice is thus forced by the pressure, and particularly by the flash of the juice, towards

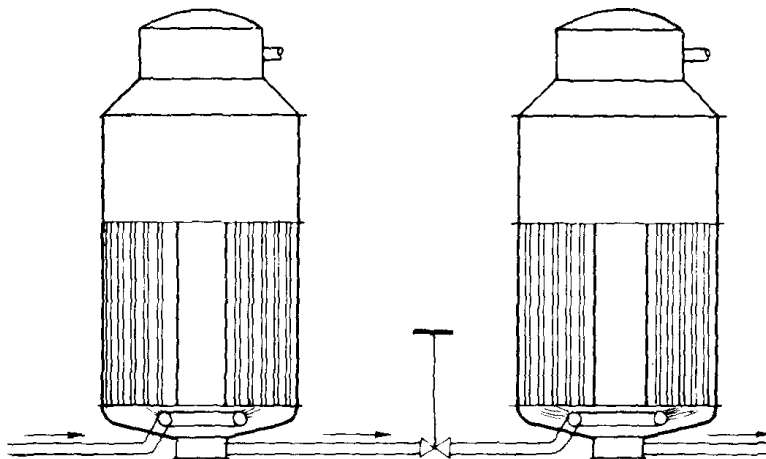


Fig. 32.17. Juice circulation from bottom to bottom.

the tubes and in the direction opposite to that of the juice outlet. Normally it should then rise at least once through the tubes, before descending again by the centre well.

Dimensions of juice pipes. It is the difference in pressure which maintains the flow of the juice from one vessel to the next. In order to avoid inclusion of vapour bubbles with the juice as far as possible, the juice outlet pipe is connected to a small sump in the middle of the bottom plate of the vessel and thus situated under the centre well. The cross-section of the connecting pipe will be chosen so that the juice velocity, assumed free of vapour bubbles, does not exceed:

Triple effect:	1.5 m/s (5 ft./s)
Quadruple effect:	1.2 m/s (4 ft./s)
Quintuple effect:	0.9 m/s (3 ft./s)
Sextuple effect:	0.75 m/s (2.5 ft./s)

The connecting and regulating valve should be as close as possible to the following vessel.

The connection may be made by means of an inverted siphon, but even in this case it is still necessary to provide a regulating valve at some part of the pipe. It may be sufficient to take these siphons down only 4 m (13 ft.) below the calandria (for a quadruple effect), but it is safer to make them longer, if possible, as much as 6 m (20 ft.) (*cf.* p. 519). In the case of Chapman circulation (see below), the height of the siphon may be reckoned from the upper tube plate.

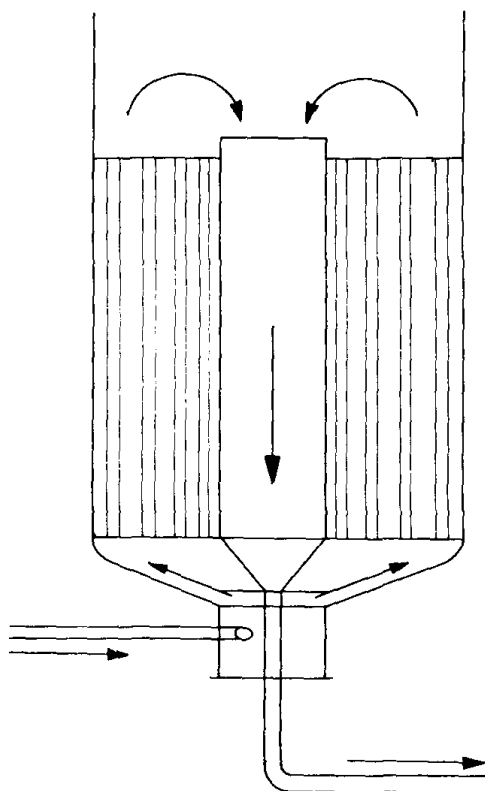


Fig. 32.18. Chapman circulation.

(3) **Chapman circulation.** This system (Fig. 32.18) consists simply of closing off the lower opening of the centre well by a funnel which forms the outlet point for the juice passing to the following vessel. Thus the only juice which leaves this vessel is that which flows down into the centre well. Consequently, if it is arranged that the entering juice is directed towards the bottom of the tubes near the periphery of the calandria, there is a good chance that it will pass several times through the tubes before reaching the centre well.

This system has given excellent results. Webre²⁷ reports that it has given gains in capacity

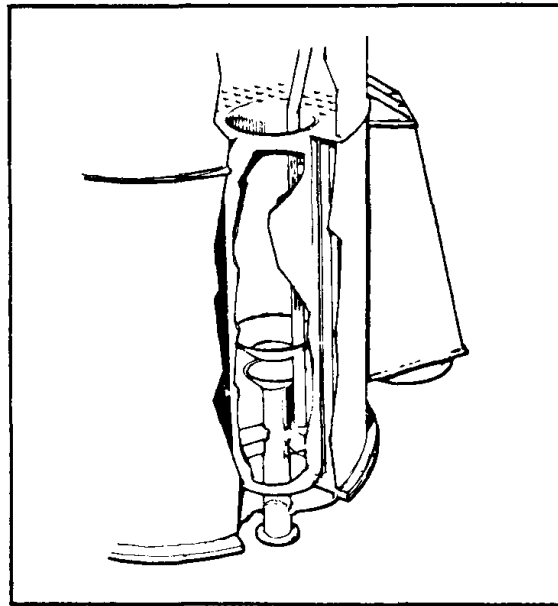


Fig. 32.19. Lateral outlet. Circular outlet.

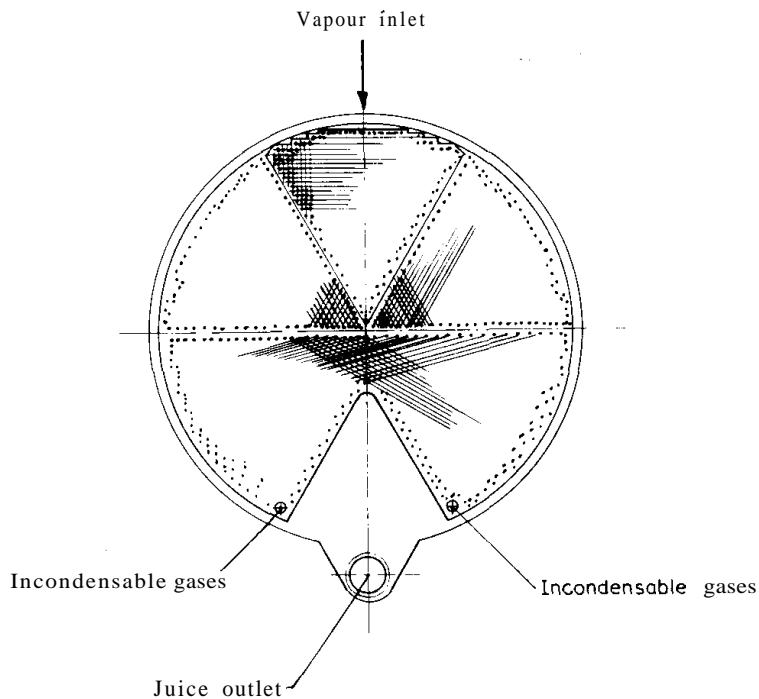


Fig. 32.20. Calandria with lateral outlet.

of the order of 25 – 50%, though these figures are much higher than those experienced by the author. The system necessitates only very slight modifications, which are readily made.

(4) **Lateral or annular downtake.** Some manufacturers have adopted the same principle, but place the **downtake** for the concentrated juice not in the centre but at the periphery of the calandria. The downtake may be circular (Fig. 32.19) or triangular (Fig. 32.20). In the latter case it takes the form of a sector of the circle of the calandria, but with a width only half or two-thirds of the radius of the calandria. This lateral downtake presents the advantage that the juice, if distributed at the bottom on the side opposite the downtake, has a large number of tubes to traverse before reaching the downtake.

Other manufacturers, of which Fives Cail- Babcock is one, prefer an annular downtake (Fig. 32.21). The vessel is then made in two parts: the calandria of diameter D , and the shell, of diameter $D + 2e$ (e being the width of the annular downtake). The downtake thus consists of the annular space between calandria and shell. The steam is admitted from above in the centre of the calandria; the juice has to pass through the tubes between the steam inlet and the downtake before entering the latter. Control of the rate of flow is achieved by means of

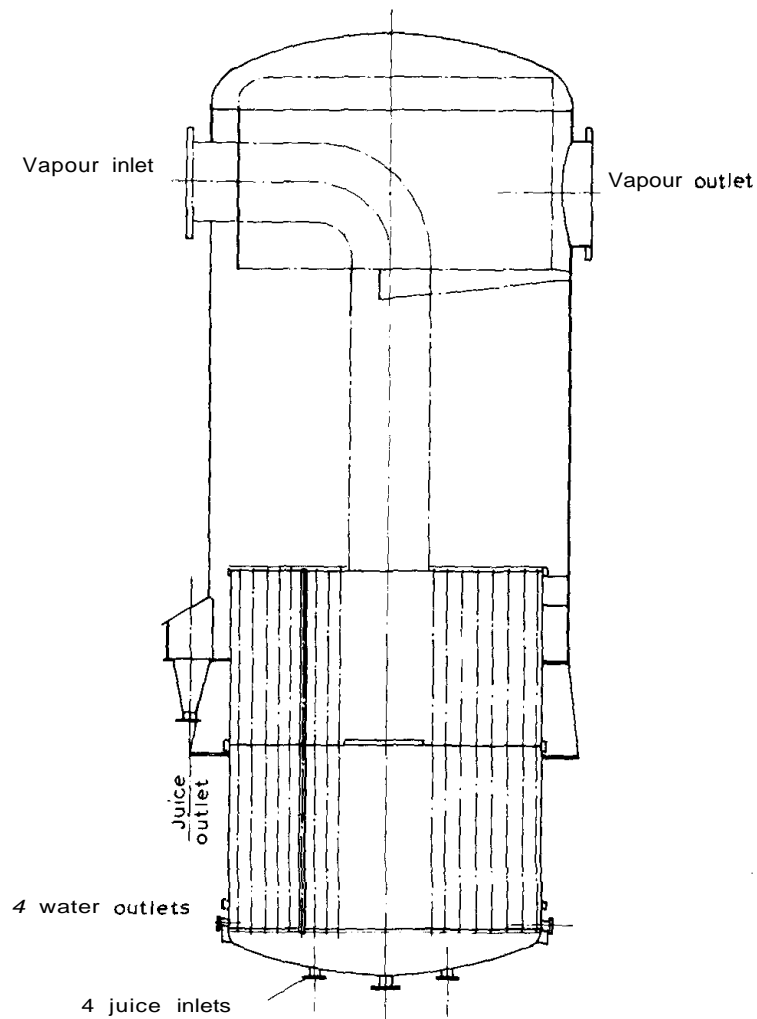


Fig. 32.21. Vessel with annular outlet.

a level regulator, which maintains the juice level close to the mid-height of the downtake, the control acting on the amount fed into the vessel. This system has the advantage of maintaining a high flow of juice into the annular downtake and assures a certain flexibility in the regulation of the juice level. The control becomes to some extent automatic by adapting the gushing of juice from the tubes according to the juice flow. This is an advantage common to the Chapman system and the downtake systems. The regulator is of torsion-tube and integral-action type, functioning by reducing the flow of juice: for example, Maisoneilan, Camflex or Minitork models.

(5) By juice-level regulator. Originally, the valves on the connecting juice pipes between effects were regulated by an attendant who maintained the desired levels in the various vessels. He is now relieved of this duty by placing this valve under the control of a juice-level regulator (Figs. 32.22 and 32.23).

It is indeed important to maintain the juice level at its optimal value, as the efficiency of the evaporator falls rapidly below or above that value. Now the operator is not capable of keeping the level within such narrow limits as does the automatic regulator, which keeps the level practically constant; and this control renders the presence of the attendant unnecessary.

If the valve on the pipe leading from one vessel to the following one is of a type which does not open fully, it should be placed at the lowest point of the pipe, as it is the valve that causes the major part of the loss of head: it will thus act where the difference of pressure is the maximum possible: generally 25 - 36% for vessels with central downtake.

Juice level

We have just been discussing the value of maintaining an optimum level for the juice in the evaporator vessels. What is this optimum level?

If the juice level is too low, the boiling juice cannot reach the top of the tubes. There will then be no circulation of juice in the calandria. If the level is too high, the tubes will be drowned or fully submerged: there will then be no climbing film.

Experiments by Kerr have given a graph (Fig. 32.24) giving the change in evaporation rate or in coefficient of heat transfer as a function of the hydrostatic level of the juice in the calandria. This graph gives a very definite maximum at the point where the juice level reaches 35% of the length of the tube, measured from the bottom. It may be considered that heat transmission is at a maximum for juice levels between 30 and 40% of the height of the tube.

It is often stated simply that the optimum level to be maintained corresponds to one-third of the height of the tubes; in general, 25 — 36% for vessels with central downtake.

Influence of length of tubes. However, this graph applies only for the old types of evaporator, for which it was established. With modern vessels with long tubes, approaching the Kestner type of evaporator, this optimum tends to decrease, and drops to 15 – 20% in the Kestner with very long tubes (7 m).

Influence of tube diameter. Hausbrand and Hirsch²⁸ have observed that the optimum level depends on the diameter of the calandria tubes. For tubes of small diameter, it will be lower (for example, 25%) than for tubes of larger diameter (for example, 30%), on account of the greater chance of formation of slugs of vapour in the small tubes.

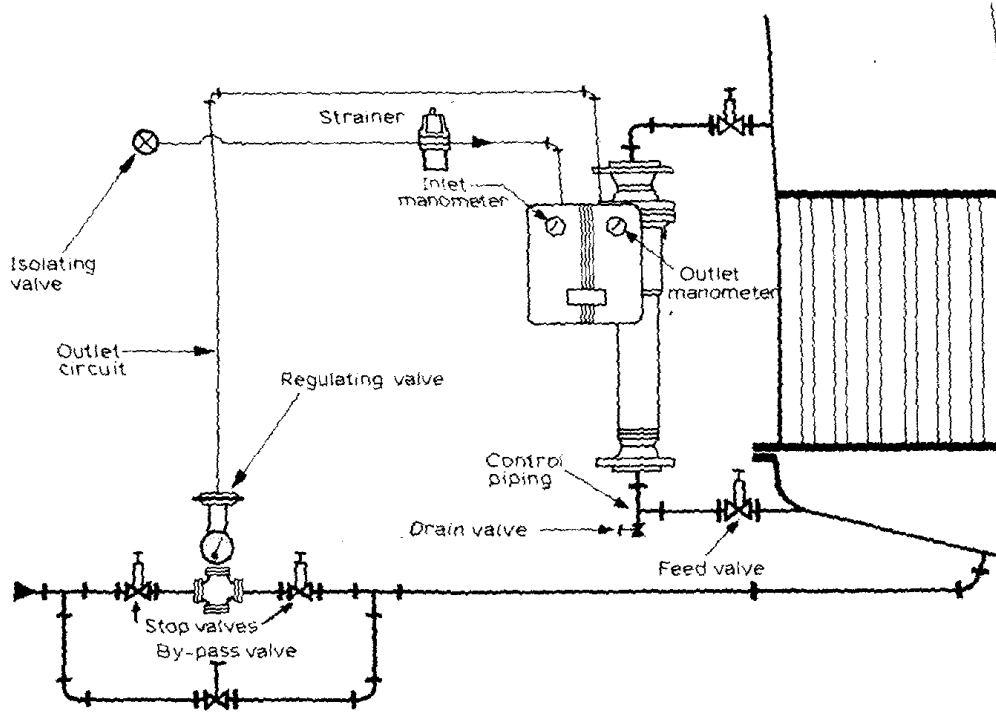


Fig. 32.22. Juice-level control in multiple effects.

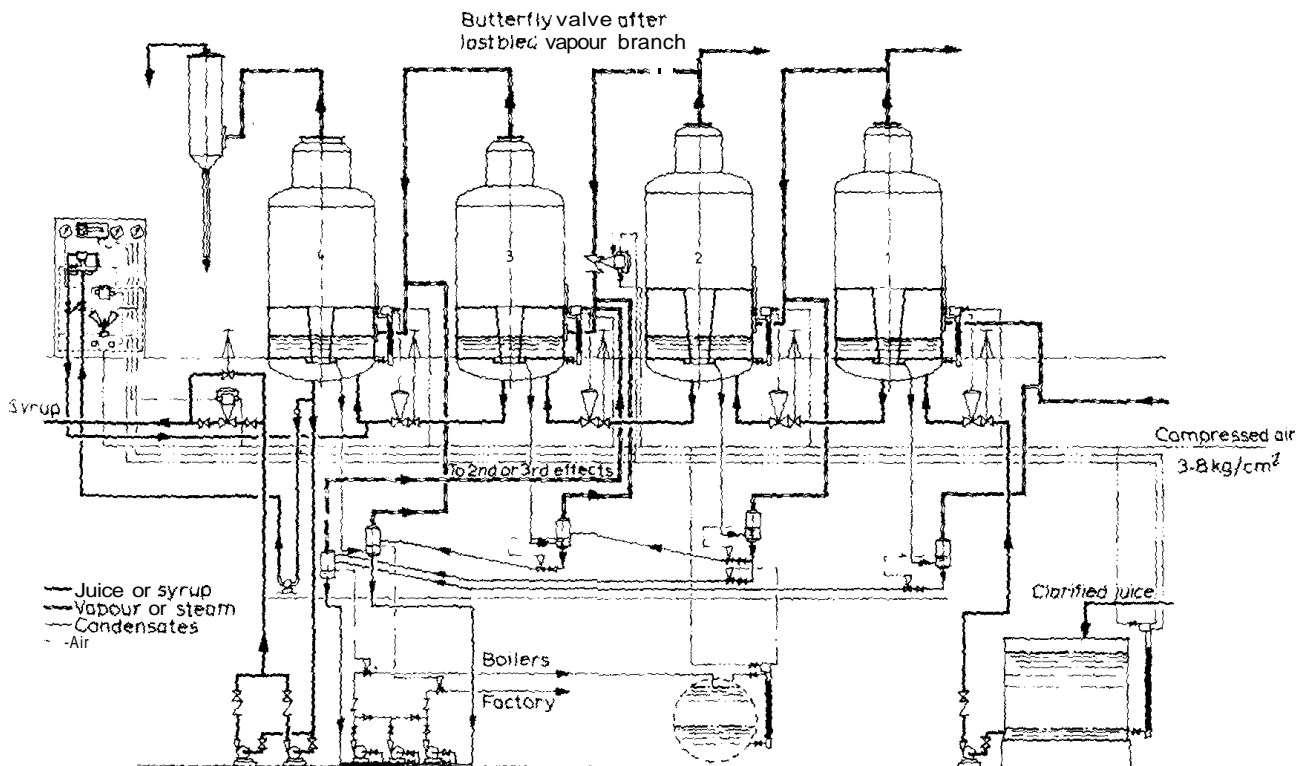


Fig. 32.23. Juice-level control (Fives Cail - Babcock).

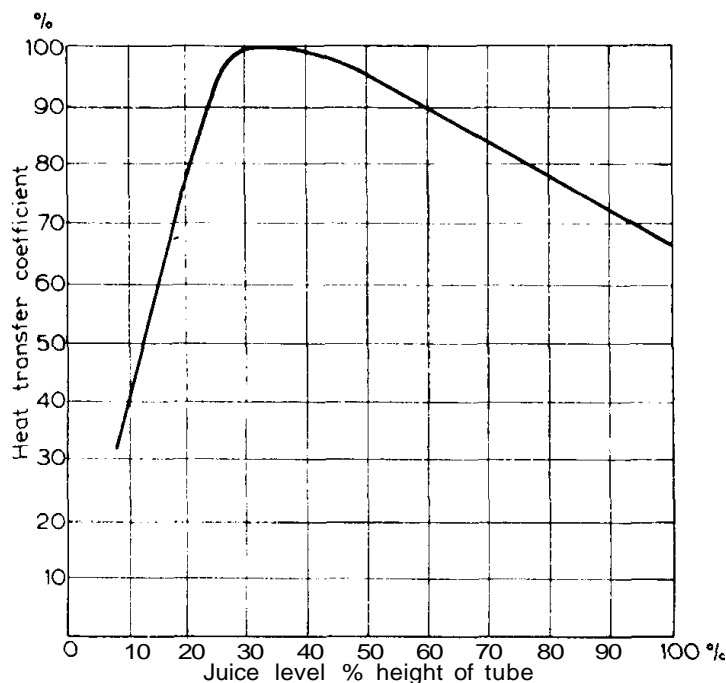


Fig. 32.24. Variation of heat-transfer coefficient as a function of juice level in the calandria.

Influence of position in the set. In Formosa²⁹ it has been ascertained that the optimum level is as follows:

- 40% for the 1st vessel
- 25% for the 2nd vessel
- 20% for 3rd and 4th vessels

Influence of scale. The optimal level is lower in a vessel with clean tubes. As the tubes accumulate scale, it is advisable to raise the level, for example, from 15 to 25% or from 20 to 30% of the tube length, during the course of the week.

Thus the optimum level varies between 15 and 40% of the height of the tubes. The true criterion which should guide the operator is the ebullition of the juice in the interior of the vessel. The operator will know that he is in the neighbourhood of the optimum level when he sees the juice spouting from the tubes, over the whole surface of the tube plate, and wetting it without submerging it. It should fall around each tube and immediately descend in some of the neighbouring tubes, which throw it up afresh.

Control of brix of syrup

For proper operation of the pans and good exhaustion of the molasses, it is important to have a syrup of uniform and constant brix. Otherwise, the sugar boilers will become confused and the settings of automatic pan controllers will have to be altered very frequently.

There are some excellent types of control equipment available (Fig. 32.25) for ensuring a uniform brix of syrup. The control is obtained by recycling, or on the other hand short-circuiting of syrup between the last two effects. It presupposes an evaporator set of capacity

somewhat greater than strictly necessary, otherwise the crushing rate of the factory would be somewhat decreased, or the brix of syrup would fall below the set value.

Such a controller is generally an indicator-controller of brix, actuated by the differential pressure registered between the bottom and top of a column of syrup about 3 m (10 ft.) in height, and converting the variation in pressure into the movement of a butterfly valve regulating the flow of juice or syrup.

Removal of the syrup. The syrup is extracted from the last vessel by a pump.

In the case of a centrifugal pump, the narrow passages of the rotors are liable to become blocked by deposits. If this trouble is encountered, a useful method consists of installing the syrup pump and the condensate pump in such a way that they can be interchanged by the simple operation of valves. By alternating the pumps, the deposit formed while pumping



Fig. 32.25. Cabinet for brix and evaporator control (Babcock Atlantique).

syrops is removed. This system obviously cannot be used when these condensates are being used as boiler feed-water.

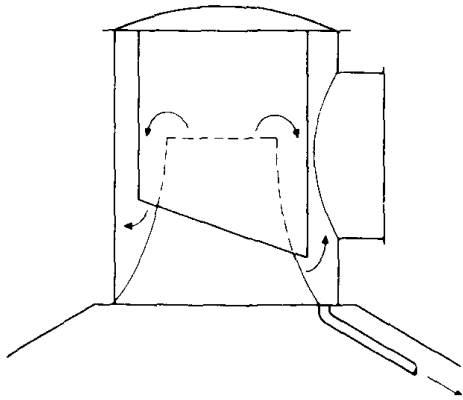


Fig. 32.26. Standard entrainment separator.

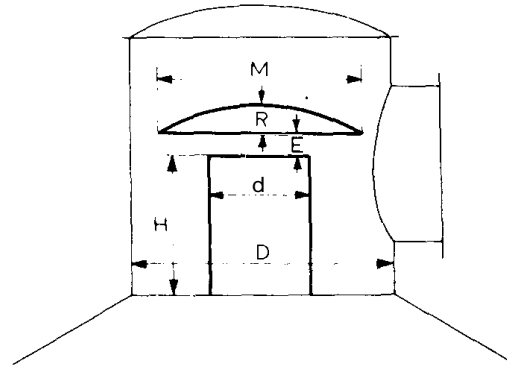


Fig. 32.27. Umbrella-type separator.

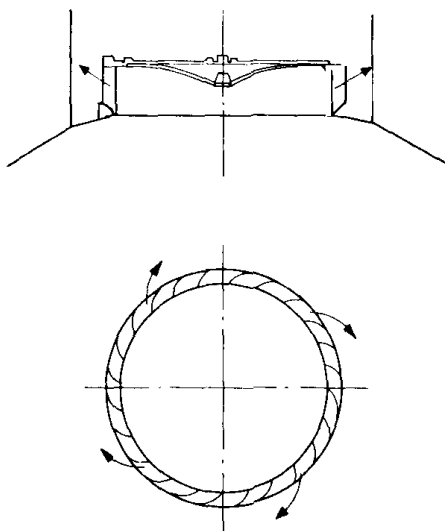


Fig. 32.28. Centrifugal-type separator.

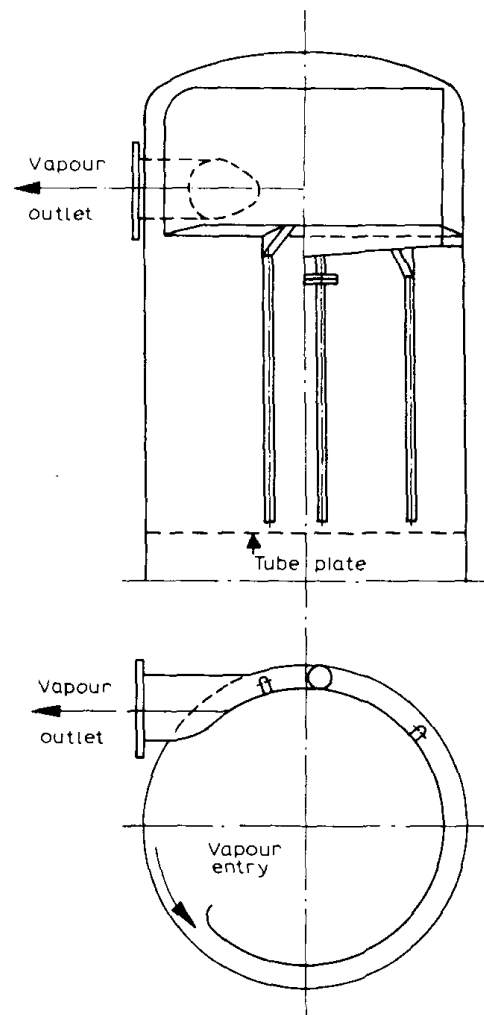


Fig. 32.29. S.G.C. entrainment separator.

ENTRAINMENT

During evaporation, the ebullition projects into the vapour space of the vessel fine droplets of juice, some of which are actually minute bubbles of vapour enclosed in a film of juice, like soap bubbles. These droplets thus are very light, and the current of vapour will easily entrain them and carry them towards the following vessel or towards the condenser.

The danger of entrainment is greater as the vacuum becomes greater. It is therefore mainly in the last vessel that entrainment is most to be feared.

The risk similarly increases very rapidly as the size of the droplets decreases. When they are relatively large, a slight increase in the height of the vessel is sufficient to stop them. If they occur in the form of a mist or fog, it is much more difficult. Thus it is important to avoid as far as possible the formation of this mist.

To avoid losses by entrainment, it is indispensable to furnish the evaporator vessels with a device for separating the drops of juice. It is generally placed at the top of the vessel, and is termed a "save-all", or "entrainment separator". It is based on the same principles as analogous apparatus designed for removing solid or liquid particles from gases: abrupt changes of velocity or of direction, centrifugal force, impact on a wall or some obstacle, etc.

We show in Figs. 32.26 — 29 four different types of separators.

In the umbrella type of "save-all", also known as the Heckmann, the proportions recommended in Queensland by Halpin³⁰ are as follows:

$$\begin{array}{ll} D = 1 & R = 0.036 \\ d = 0.29 & E = 0.06 \\ M = 0.36 & H = 0.43 \end{array}$$

In the S.G.C. entrainment separator of Fives Cail—Babcock (Fig. 32.29), the diagram corresponds to medium-sized evaporator vessels, for which 1 vapour outlet passage is sufficient. It is important, for efficient separation, that the passage should be narrow. Also, for evaporator bodies of more than 2,500 m² (27,000 sq.ft.), 2 concentric vapour passages are provided, side by side, each receiving half the total vapour. The two passages join at the outlet. The juice or syrup recovered by such separators is returned to the vessel by small pipes which deliver the liquid close to the wall of the vessel, or preferably into the main body of liquid. It is advisable to provide two or three such pipes.

Entrainment separators

On account of the vacuum, the most serious entrainments are those produced in the last vessel. In order to catch these before they reach the condenser, an entrainment separator is inserted in the vapour pipe going to the condenser. This is generally (Fig. 32.30) a cylindrical vessel forming a simple enlargement of the vapour pipe, and containing grilles formed from tubes of copper or steel, split longitudinally in two, placed side by side (Fig. 32.31) and arranged vertically, with the hollow side facing the vapour stream. These tubes are about 25 – 40 mm (1 - 1½ in.) in diameter, and are separated by an interval equal to about half their width. Hence it is necessary that the cross-section of the "save-all" should be at least three times and preferably four times that of the vapour pipe in which it is inserted. The entrained

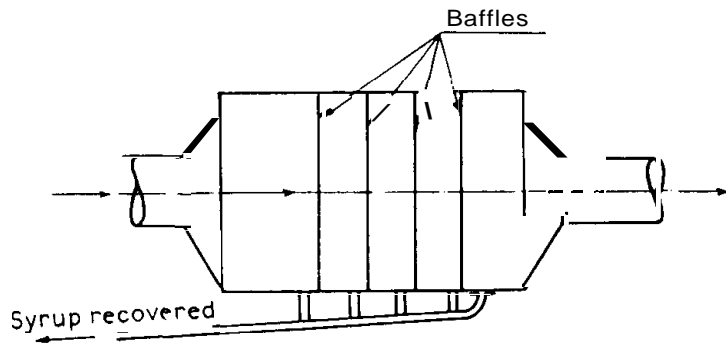


Fig. 32.30. Entrainment separator.

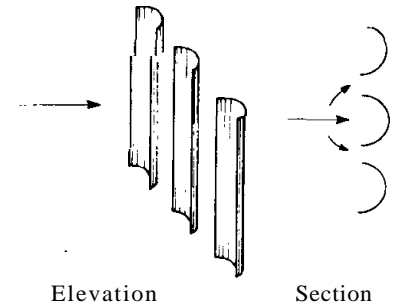


Fig. 32.31. Split tubes for grilles.

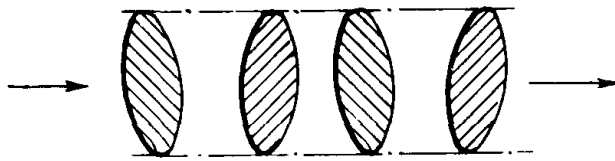


Fig. 32.32. Arrangement of successive grilles.

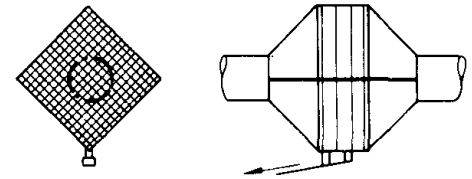


Fig. 32.33. "Ace of diamonds" separator.

droplets strike against the hollow side of the tubes, adhere to them, and descend the length of the tube. The syrup recovered is collected at the bottom of the vessel and drains to a recuperator bottle.

Since one such grille has an efficiency much lower than 100%, several grilles are placed one after the other, giving the tubes a slight inclination of several degrees from the vertical, alternating to the left and to the right with successive grilles (Fig. 32.32), so that they slope in opposite directions. At least four grilles are necessary, preferably five.

To supplement the action of these large and rather cumbersome entrainment separators, an additional unit is often added, on the individual vacuum pipes of the vacuum pans and, if necessary, of the multiple effects. This separator is based on the same principle, but is relatively flat in the direction of the pipe length; it is set up diagonally or as an "Ace of diamonds" (Fig. 32.33) and is of square section, with tubes inclined to the right and left at 45° .

Several other types of entrainment separator are also used.

Mesh eliminators. It is possible to assist the entrainment separators and to reduce their task by interposing at the top of the evaporator vessel a "mattress", 10–15 cm (4–6 in.) thick, formed of fine threads of stainless steel knitted or felted together, somewhat after the style of a Liebermann cloth (p. 759), so as to obtain a layer of the desired thickness. This mattress is compressed and held between two supporting grilles formed of hoop iron and fixed at the edges. It catches the entrained droplets, which accumulate, join together, increase in size and fall back into the evaporator in the form of drops of sufficient size to prevent being re-entrained.

These mesh separators would have an efficiency³¹ of approx. 85%. They involve an additional pressure drop of approx. 7–10 mm ($\frac{1}{4}$ – $\frac{1}{2}$ in.) of water, say 0.5–0.8 mm (0.02–0.03 in.) of mercury, which is thus very slight. It is estimated that they would pay for themselves

in three years, but they foul up dreadfully in a single season: they are fragile and very difficult to clean effectively. It is essential to install soda and acid sprays above them in order to assure chemical cleaning which will be frequent and as effective as possible. Otherwise, they quickly become blocked and ineffective, and can even obstruct the operation of the multiple effects. The suppliers of such mesh separators are, in France, Knit and Tissmetal, and Koch, in the U.S.A.

Recovery bottle. The syrup recovered in the entrainment separator is passed to a small chamber fitted with sight glasses, so that one can see the liquid flowing into its interior. This is called the recuperator bottle or recovery bottle (Fig. 32.34).

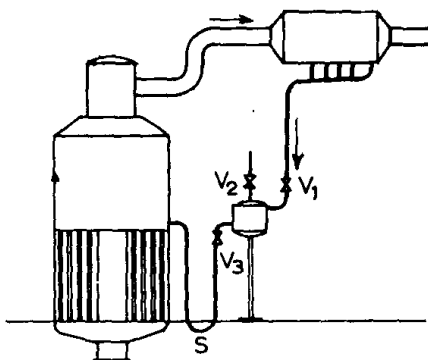


Fig. 32.34. Recovery bottle.

This bottle is connected to the last vessel. In operating, it is allowed to fill; when it is full, the syrup inlet valve v_1 is closed, the valve v_3 connecting to the vessel is opened, also the vent cock v_2 communicating with the atmosphere. The syrup is then drawn into the vessel. Immediately valves v_2 and v_3 are closed, v_1 is reopened, and the bottle is allowed to fill again.

This method has the serious disadvantage that the operators frequently forget to empty the bottle. Thus the syrup recovered is lost, as if the separator were not there.

It is preferable to interpose a small syphon S between the bottle and the last vessel, and to leave v_1 and v_3 always open; or the valves may be omitted. The syrup will then flow continuously into the vessel by gravity.

Since the pressure drop between the last evaporator vessel and the entrainment separator is very small, a very short syphon will be sufficient, of about 90—120 cm (3—4 ft.).

Magnitude of entrainment

Deerr³² estimates that losses by entrainment may attain 3% of the juice in a multiple effect without any entrainment separators, and that they can be reduced to less than 0.1% if these devices are properly installed.

Prinsen Geerligs³³ cites a number of factories where undetermined losses had been high, and where several improvements in the entrainment separators allowed these losses to be reduced in % of sugar in cane:

From:	3.59%	4.55%	2.45%	3.39%	3.12%
To:	2.65%	2.65%	1.57%	1.17%	0.78%

respectively.

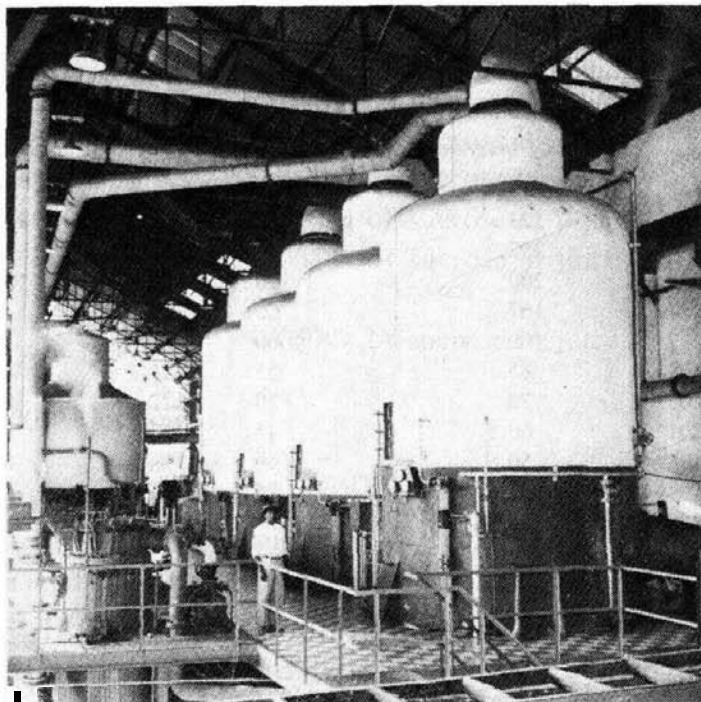


Fig. 32.35. Quadruple effect, H.I.W. design.

Precautions to be taken against entrainment

The best methods of reducing entrainment are:

(a) Avoid increasing the vacuum to an unnecessarily high figure. Be content with 60–65 cm (24–26 in.), values, which moreover offer certain advantages from other points of view (*cf.* pp. 506, 610, 649).

(b) Arrange the juice entry at the bottom of the vessel and not above the upper tube plate, where it would provoke splashing, and might even break up the juice to a fine mist.

(c) Avoid forcing the multiple effects above their normal capacity (*cf.* p. 508).

(*cf.*) Avoid allowing the juice level to rise appreciably above the optimum level.

(e) Provide the vessels with “save-alls” and the vapour pipe with an efficient entrainment separator.

(f) But it must not be forgotten that the best save-all is a high vessel; the body of the vessel should have above the upper tube plate, at least twice and preferably two and a half times the height of the latter.

INVERSION LOSSES

When the juice is carried to a high temperature, its sugar content undergoes decomposition by inversion, which is more marked at high temperatures and higher acidity. As far as temperature is concerned, it is above 100°C (212°F) that inversion increases rapidly; it becomes prohibitive above 125–130°C (260–265°F). Taking the rate of inversion at 100°C as 100, Honig³⁴ gives the figures of Table 32.9.

TABLE 32.9

TEMPERATURE AND INVERSION RATE

Temperature (°C)		Rate of inversion	Temperature (°C)		Rate of inversion
(°C)	(°F)		(°C)	(°F)	
120	248	523	80	176	15.7
110	230	257	75	167	8.5
107	225	200	70	158	4.9
105	221	167	65	149	2.9
100	212	100	60	140	1.7
98	208	85	55	131	1.0
96	205	72	50	122	0.47
94	201	60.5	45	115	0.25
92	198	50.6	40	104	0.13
90	194	41.5	25	77	0.02
85	185	24.5			

As far as acidity is concerned, inversion increases mainly below pH = 6.5.

In Table 32.10 we quote the table of Stadler which gives the quantities of sucrose inverted at common temperatures and pH values.

Values may readily be extrapolated from this table by remembering that columns of the same decimal of pH may be deduced one from the other by a simple displacement of the decimal point by one digit. For example, at 100°C, the percentage of sucrose inverted per hour will be:

at pH = 5.0, 2.12%	at pH = 4.8, 3.37%
at pH = 6.0, 0.21%	at pH = 5.8, 0.34%
at pH = 7.0, 0.021%	at pH = 6.8, 0.034%

TABLE 32.10

PERCENTAGE OF SUCROSE INVERTED PER HOUR (STADLER)

Temperature		pH					
(°C)	(°F)	6.0	6.2	6.4	6.6	6.8	7.0
50	122	0.0010	0.00063	0.0004	0.00025	0.00016	0.00010
60	140	0.0035	0.0022	0.0014	0.00088	0.00056	0.00035
70	158	0.011	0.007	0.0044	0.0026	0.0018	0.0011
80	176	0.033	0.021	0.013	0.0083	0.0052	0.0033
85	185	0.053	0.034	0.022	0.013	0.0084	0.0053
90	194	0.089	0.056	0.035	0.022	0.014	0.0089
95	203	0.14	0.088	0.055	0.035	0.022	0.014
100	212	0.21	0.13	0.084	0.053	0.034	0.021
105	221	0.35	0.22	0.14	0.088	0.056	0.035
110	230	0.54	0.34	0.22	0.14	0.086	0.054
120	249	1.1	0.70	0.44	0.28	0.18	0.11

In the same way, for a pH of 7.2, the figure will be 0.013, since for pH of 6.2, it is 0.13.

For a standard quadruple effect, with the first vessel heated by steam at 112°C (234°F), Claassen³⁵ evaluates losses of sucrose by inversion at:

1st vessel:	0.020%	of sucrose in juice
2nd vessel:	0.015%	of sucrose in juice
3rd vessel:	0.010%	of sucrose in juice
4th vessel:	0.005%	of sucrose in juice

	0.050%	of sucrose in juice

or approximately 0.007% on cane.

Control of inversion losses

The general method employed in the sugar factory to detect inversion losses in the course of evaporation consists of following the glucose ratio of the juice. This term means the ratio of reducing sugars to sucrose (or to pol) in the juice. The ratio should decrease slightly between clarified juice and syrup. Honig³⁶ estimates the normal difference at about 4% (3.12 compared with 3, for example). If this quotient increases, or remains stationary, it is probable that an inversion loss is occurring during the evaporation. This must then be attributed either to too high a temperature in the first vessel, or to too long a time in the vessel (generally due to short-circuit of part of the juice from entry to exit), or to too low a pH value.

The normal pH drop between clear juice and syrup, according to Honig, is 0.3, and should not exceed 0.5. For example, the clarified juice may be at 6.9 and the syrup at 6.5.

Normal inversion losses in the course of the evaporation, again according to Honig, should not exceed 0.2% of sucrose as a maximum.

It is, moreover, of value to follow the glucose ratio throughout the manufacture. It must be remembered, indeed, that every loss of glucose shows up as a loss of sucrose, since, in the molasses, one replaces the other more or less completely. Roughly, we may reckon that four parts of reducing sugars release one part of sucrose. Now, while sucrose is destroyed at high temperature and low pH, glucose is destroyed at high temperature and high pH. A compromise must accordingly be adopted for the pH value, and it is equally desirable to avoid remaining too long above a pH of 8.5 as to avoid going below pH 6.5, particularly at high temperatures.

HEAT LOSSES

As in juice heaters, utilisation of the heat supplied in the steam to a vessel of the multiple effects would not be 100%. Part of the heat always escapes, by radiation and convection, to the ambient atmosphere.

Tromp³⁷ estimates the heat loss in the several vessels of a multiple effect of which the vessels themselves are lagged, but not the interconnecting pipes, at 3 - 10% of the heat supplied to the vessel.

Deerr³⁸ estimates the loss by radiation and convection in the case of a quadruple, at ap-

proximately 40 kg (90 lb.) of steam per tonne of cane. Since the actual loss corresponds only to the steam supplied to the first vessel, this means:

(1) 10 kg (22 lb.) steam/tonne cane if the quadruple is of ample capacity, and concentrates all the syrup to the desired degree

(2) 40 kg (90 lb.) steam/tonne cane if the quadruple is of insufficient capacity, because then this loss will decrease with the concentration of the syrup, and must be compensated fully by a corresponding consumption of steam in single effect in the vacuum pans.

Generally, multiple effects are lagged with wooden slats enclosing the cylindrical parts of the vessels, of the "save-all" and of the pipes. The bottom, the top dome, the top of the "save-all" and the bends of the pipes are left bare. In this case, one may estimate the losses at 5% of the heat exchanged in each vessel.

If the vessel is completely lagged with an insulation in the form of a powder or a paste, the loss may be estimated at about 2%.

Sandera³⁷ has found a loss of 83 kcal/m²/h (30 B.T.U./sq.ft./h) of exposed surface of an evaporator, covered with an insulating layer of 60 mm (2½ in.) thickness, for a temperature drop of 52°C (94°F) between the vapour from the juice and the ambient air. This represents a loss of close to 1.6 kcal/m²/°C/h (0.33 B.T.U./sq.ft./°F/h).

It is recommended³⁹ that all surfaces should be painted with aluminium paint; this is sufficient to decrease the losses by 10—20%, whether the equipment and pipes are lagged or not. Anti-rust aluminium paint is available.

TABLE 32.11

HEAT LOSSES FROM A MULTIPLE EFFECT IN PER CENT OF STEAM SUPPLIED TO 1ST EFFECT

	<i>Bare vessels</i>	<i>Partially lagged</i>	<i>Completely lagged</i>
Double effect	1.06	0.46	0.26
Triple effect	4.20	2.07	1.05
Quadruple effect	9.80	5.00	2.70

However, what is interesting is to estimate the loss of heat in relation to the steam supplied to the first vessel. Following Webre⁴⁰, Kerr gives the estimates shown in Table 32.11 for the loss of heat suffered by a multiple effect, expressed in per cent of the steam supplied to the first vessel.

We shall retain the value of 5% for a quadruple effect partially lagged, or 2.7% if fully lagged.

Influence of place of the vessel in the set

We may assume as a first approximation that the loss of heat from a vessel is proportional to the difference in temperature between the vessel and the ambient air. In a multiple effect, the loss of heat will therefore be greater for the first vessel than for the second, for the second than for the third, and so on.

On the other hand, a loss of heat in the first vessel decreases the evaporation in the second (since it is the vapour which comes from the first which heats the second), then in the third,

and so on. In other words, the heat losses suffered by the first vessel, between the vapour space and the following calandria, affect the second vessel and have their repercussions on the following vessels. They are then multiplied by $(n - 1)$, n being the number of effects. The losses of the second vessel are multiplied by $(n - 2)$, and so on. The losses from the last vessel are of no consequence, since its vapour goes to the condenser. We shall take for example a quadruple with temperatures as follows:

		<i>Difference in temperature above ambient air of 30°C (86°F)</i>
Exhaust steam	= 112°C (234°F)	
Vapour from 1st vessel	= 104°C (219°F)	74°C (133°F)
Vapour from 2nd vessel	= 94°C (201°F)	64°C (115°F)
Vapour from 3rd vessel	= 80°C (176°F)	50°C (90°F)
Vapour from 4th vessel	= 55°C (131°F)	25°C (45°F)

The heat losses from the first vessel will therefore be 1.5 times as much as for the third if they are of equal surface and of comparable insulation.

On the other hand, the loss from the first vessel has its effect finally three times, the loss from the second twice, the loss from the third once only.

The losses by radiation and convection in each vessel will have approximately the following relative importance:

1st vessel:	$74 \times 3 = 222$,	or close to 9
2nd vessel:	$64 \times 2 = 128$,	or close to 5
3rd vessel:	$50 \times 1 = 50$,	or close to 2
4th vessel:	$25 \times 0 = 0$,	or close to 0

This shows immediately to what extent it is important to insulate the first vessel, and to take particular care with this insulation.

Conversely, one could well dispense with lagging for the last vessel. However, for aesthetic reasons, it is given the same exterior covering as the preceding vessels, but of minimal thickness.

INCRUSTATIONS

In operation, the tubes of a multiple effect become fouled in two ways:

- (1) On the outside, a deposit of oil carried by the steam is formed
- (2) Inside, incrustations or scale derived from the juice are deposited.

r deposit of oil

This deposit, originating in the oil entrained with the exhaust steam, obviously occurs only in the first vessel. It hardly exists except in the case of factories which still use reciprocating

steam engines, particularly if these are lubricated by atomisation of oil at the steam entry. To avoid this brownish or blackish deposit of dirty oil, or to reduce it to a very great extent, it is of great advantage to interpose an oil separator in the path of the steam between the exhaust main and the first effect.

This deposit is detrimental to heat transmission in the vessel. It may be removed during the slack season by introducing into the calandria:

(a) A very dilute acid, for example HCl of 0.25%

(b) Molasses diluted to 20° brix⁴¹ which is left to ferment slowly, and which is emptied out drop by drop, regulating the rate so that emptying the calandria takes about one month

(c) Water covered with a layer about 7 mm ($\frac{1}{4}$ in.) thick of kerosene, which again will be run out drop by drop so that the petrol will slowly clean the full height of the tubes

(d) Turpentine: the calandria is half filled with water⁴², and oil of turpentine is introduced so as to obtain a layer of 10 cm (4 in.) (or in the ratio of 2.0 kg per 1,000 dm² (40 lb. per 1,000 sq.ft.) of heating surface); the liquid level is then made to rise and descend 10 or 15 times so that the turpentine scours the full length of the tubes

(e) Caustic soda, which may be boiled in place of juice: this procedure has given excellent results in India.

(2) Scale

The most troublesome deposit is formed by incrustations inside the tube.

Origin. These originate from:

(a) Material in suspension in the juice, poorly separated by a defective defecation or filtration; these materials deposit mainly in the first vessel

(b) Non-sugars in solution, which become insoluble as the juice becomes concentrated. These deposits form mostly in the later vessels.

Quantities. We may reckon on an average of 200 - 800 g of scale on a dry basis per m² of heating surface, say approximately 500 g/m², or 10-50 g/t.c. (0.1 lb/sq.ft., or 2-10 lb./100 t.c.).

Distribution. The distribution of these deposits in the tubes depends mainly on the speed of circulation or the state of agitation of the liquid; the deposits form where the liquid is stagnant and where the speed of circulation is low.

Composition. Scales consist principally of:

(a) Calcium salts: phosphate, sulphate, oxalate, and carbonate of calcium; the phosphates decrease while the sulphates increase from the first to the final vessel⁴³.

(b) Metallic oxides: oxides of magnesium, aluminium and iron

(c) Silica; this forms the greater part of the deposit in the last vessel.

In addition, sulphites are found in factories using sulphitation. They may amount to 30% of the scale when the acidity of the juice reduces the pH below 6.7. They are reduced by carrying out the defecation in a slightly alkaline medium (pH = 7.2), and by preventing the temperature from dropping between sulphitation and evaporation.

In defecation factories, the proportion of calcium phosphate may in turn exceed one-third.

In carbonatation factories, it is calcium oxalate which predominates. However, it is always present, although oxalic acid is not found in the juices; *Prinsen Geerligs*⁴⁴ has shown that it was due to oxidation of glycollic acid in the juice at high temperature.

Sulphate scales are the hardest. Silicious scales are sometimes porous: these are softer, but may occur as hard and dense scale.

Thermal conductivity. Tests made in Queensland⁴⁵ have given as mean thermal conductivity for scales from evaporator vessels a value of 0.45 kcal/m²/°C/h/m (0.3 B.T.U./sq.ft./°F/h/ft.).

Distribution of scale. Scales are much thicker at the lower part of the tubes, where the juice is relatively stagnant, than at the top, where it is in active movement. Further, in checking the cleaning of a vessel, one must not be content to inspect the top of the tubes; it is necessary to place a lamp at the bottom of the tube and to inspect the tube from above. This will give a better indication of the condition of the bottom part of the tube.

Further, one should not trust to the appearance of the tube. We have known a case of one factory which cleaned the multiple effects with a mechanical cleaner. The tubes were polished and brilliant, but the evaporator gave very poor performance. On closer investigation it was found that the tubes contained an inner ring of very hard scale, which had become polished like agate. When tubes are replaced, it is advisable, therefore, occasionally to cut a tube in two longitudinally, and to inspect closely this section of the tube.

Relative proportions in the vessels. Quillard⁴⁶, for beet factories, estimates that scale reduces the heat transmission in the different vessels in proportions which increase with their place in the set. To take this into account, he proposes the following coefficients of increase, for a quadruple, to be applied to the respective heating surfaces of the vessels:

Pre-evaporator:	10% = 1.10
1st vessel:	25% = 1.25
2nd vessel:	30% = 1.30
3rd vessel:	35% = 1.35
4th vessel:	40% = 1.40

These proportions would represent equally well the relative importance of scale in the various effects for the cane sugar factory.

In Cuba it has been found on an average that in a quadruple effect, the amount of scale is respectively 32, 14, 13, and 41% of the total weight of scale on a dry basis.

Influence of evaporation rate. Apart from avoiding use of excess lime in the clarification, or using magnesia to replace lime (*cf.* p. 429), there is little that can be done to avoid scale formation. It is well known, however, that scale is deposited all the more rapidly and abundantly when the evaporation rate is sluggish. It is important therefore to make every effort to maintain continuous operation and a high evaporation rate; hence the advantage

sometimes obtained by Kestner-type evaporators (*cf.* p. 613). For the same reason, it is advisable to keep to the minimum the volume of juice in transit in the evaporator (*cf.* p. 533).

Cleaning of evaporators

Incrustations form rapidly and it is these which force a weekly stop, more or less prolonged, for cleaning the evaporators.

Certain factories, where the clarification is very good and the multiple effect generously designed, have to clean only once in fourteen days. We know a case of one factory possessing a Kestner of 450 m² (4,850 sq.ft.), as the first vessel without vapour bleeding, followed by a second vessel of 350 m² (3,775 sq.ft.), which is cleaned only in the slack season. This is a considerable simplification and economy, which unfortunately one would not be able to guarantee *a priori*.

Cleaning is effected in two ways:

- (a) Mechanically, by scrapers or rotary cleaners
- (b) By chemical means, by pumping soda and/or acid.

(a) Mechanical cleaning. Mechanical cleaning is effected with the aid of electric descalers. These consist of a small frame (Fig. 32.36) carrying an electric motor which may be plugged in to a power point provided close to the evaporators. This motor has a flexible shaft some 6 m (20 ft.) in length, enclosed in a protective sheath, and terminating in a tool with serrated rollers (Fig. 32.37), which it drives at high speed. These rollers are loose on their axes so that centrifugal force presses them against the wall of the tube, which permits a given tool to clean tubes of slightly differing diameters.

Generally two passages of the tool are sufficient, from top to bottom and from bottom to top, for each tube. If the tube is nearly clean, such treatment is not desirable, as it would remove some of the metal itself. It is desirable to carry out the cleaning after filling the vessel with water up to the top of the tubes.

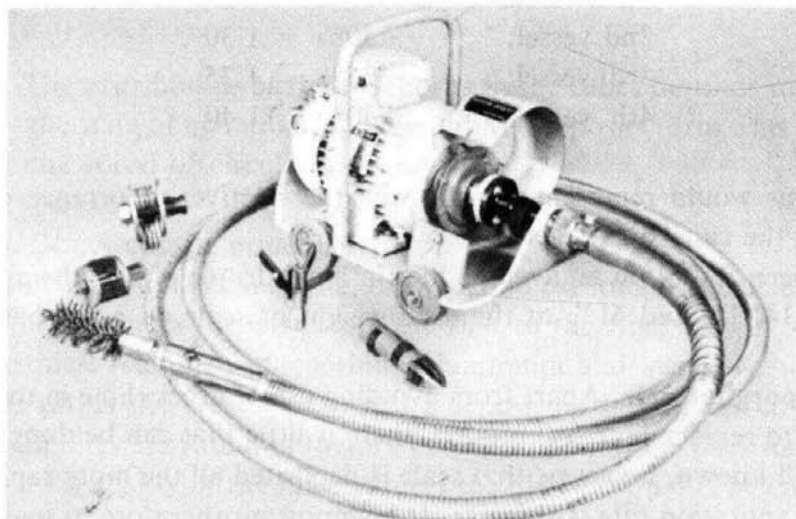


Fig. 32.36. Mechanical descaler (motor and flexible shaft) (Nordon).

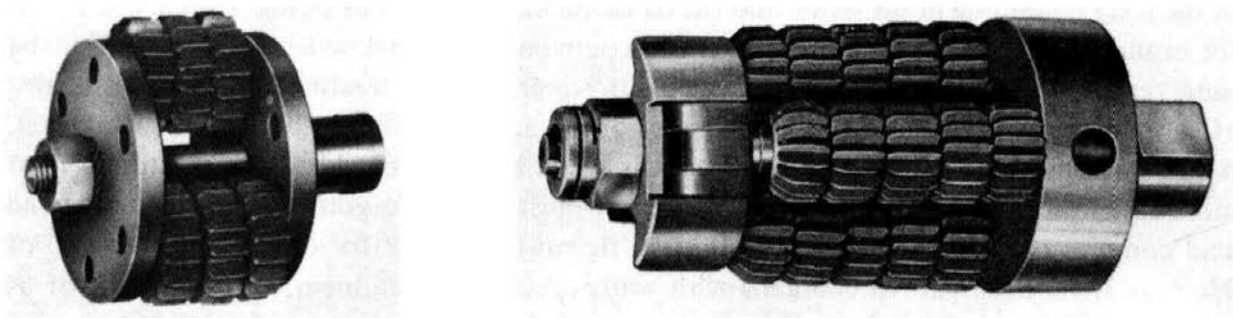


Fig. 32.37. Descaling tools.

It is possible for one man to clean about 50- 100 tubes, of the usual length of 1.5 – 2 m (5 – 7 ft.), per hour. However, in the case of hard and adherent incrustations, it is recommended that the tool be passed up and down the tubes at a speed of the order of 75 cm/min (30 in./min), which reduces the rate to about 12- 15 tubes/man/h.

The optimum speed of rotation of the tool is of the order of 3,500-4,500 r.p.m. for normal tube diameters.

Disadvantages. Mechanical cleaning is a disagreeable job, and expensive in manpower. It wears the tubes, small particles of metal being detached from them.

Voltage. French regulations forbid the use of power at more than 24 V in a closed metal vessel where men are working. It is therefore necessary to provide, in the neighbourhood of the multiple effects, a small transformer or rotary converter, and equipment operating at low voltage.

Descalers operated by compressed air may also be used; these require a relatively powerful compressor, but function very well.

(b) Chemical cleaning. This process consists of boiling a solution of NaOH or of HCl in the vessel. The methods vary greatly: practically always soda is employed, sometimes soda and acid in turn.

Soda dissolves mainly the oxalates, sulphates and the silicates. Hydrochloric acid dissolves especially the carbonates, sulphites and the phosphates.

In some cases soda at 30 or 32° Baumé is employed. In India⁴⁷, the strength is taken up to 40° Be. Others⁴⁸ use it at 27° Be (= 50° brix). Others go as low as 8 or 9° Be, and even 5°.

For HCl, a strength of 2% is never exceeded. A solution of 1%, 0.5% or 0.25% is used.

In Java⁴⁵ the practice was to boil first with a solution of 2 - 5% of caustic soda, followed by one of 0.25% of HCl, and it was stated that more concentrated solutions did not give any better results. However, in Australia⁵⁰, it has been found that the time necessary to obtain a satisfactory result depended on the concentration: one hour at 2%, 1 $\frac{3}{4}$ hours at 1.5%, 3 hours at 1%; below 1% the action became very weak.

In the U.S.A. (Delden), soda of 4% concentration (6° Be) is generally employed.

Furthermore, the acid strength drops⁵¹ very rapidly when boiling with HCl, on account

of the soda remaining in the scale, and the consequent reactions (dropping from 2.7 to 1.6%, for example, in the first 10 min); and it is recommended to commence with 2.5%. For the same reason it is necessary to boil carefully with water after the treatment with soda and also after treatment with acid. If the tubes are of brass, action on the metal is slight; if of steel, it will be considerable and in both cases it is recommended to add to the solution 5% of molasses, which forms an excellent inhibitor. It must not be forgotten that the commercial acid contains only 30% of HCl; it is thus sufficient to dilute it for example in the ratio of 250 1 to 3,000 1 (50 gal. to 600 gal.) with water. After the treatment, the scale falls of its own accord, though sometimes it is necessary to assist this by means of a jet of water.

The HCl could be replaced by "Versene", diluted to 3.5%; the results would be quite good, but the cost of the material and of its regeneration are prohibitive.

Fives Lille – Cail recommend treatment with soda, at 30 – 32° Be, for three hours, and with HCl for 1 hour.

Spengler⁵² recommends the opposite order:

- (1) Passage of hydrochloric acid (1% solution)
- (2) Washing with water
- (3) Passing caustic soda (8% solution).

Sometimes other products are added. In Trinidad⁵³ NaCl is added in the ratio of 10 g/l (1 lb./10 gal.) to a solution of soda of 50° brix (27° Be). In Louisiana⁵⁴, it is estimated that the best cleaning is obtained by a mixture of 60 – 75% of NaOH and 40-25% of Na₂CO₃. The consumption per 10 tonnes of cane is 0.5 – 1.0 kg (1.2 – 2.2 lb.) of this mixture, and 1.0-1.5 kg (2.2-3.3 lb.) of HCl.

Honig⁵⁵ recommends replacing the HCl by sulphamic acid, which is easier to handle, less corrosive, and equally effective.

In Reunion, where scales are not generally too tenacious, it is often satisfactory to use one treatment with soda, and the annual consumption is based on only 1.2- 5 kg (2.5 - 10 lb.) per 100 t.c.

In Cuba, it is estimated that cleaning with soda alone is much superior to cleaning with acid followed by soda. The soda should then be of 30 – 40%, at a temperature of 120°C (248°F) for 3 hours' boiling. They advise 0.5 l/min of circulation of the solution per m² (1 gal./100 sq.ft.) of heating surface, and a total volume to be circulated of 2 l/m² (4 gal./100 sq.ft.). In certain factories, the distinction is made according to the place of the vessel: 5% soda in the first vessel, increased progressively to 20% in the final vessel.

It is difficult to recommend one method rather than another, since the nature of the scale varies greatly from one country to another. It is advisable to try different methods and to proceed cautiously. In the case where silicates in the last vessel are found particularly tenacious, the Louisiana mixture may be used, that is, the soda is replaced by a mixture of 70% soda and 30% sodium carbonate.

As regards the time of treatment, we may reckon on 3-4 hours for the treatment with soda⁵⁶ and about 1 hour for the acid treatment. With the rinsings and dead time, it may be necessary to be satisfied with a total of 5 hours.

It is preferable to carry out this washing by means of a spray, and not as a bath, in order to economise on solution. It is then distributed by a perforated pipe (Fig. 32.38).

Heating is effected generally by the calandria, by opening the steam valve slightly.

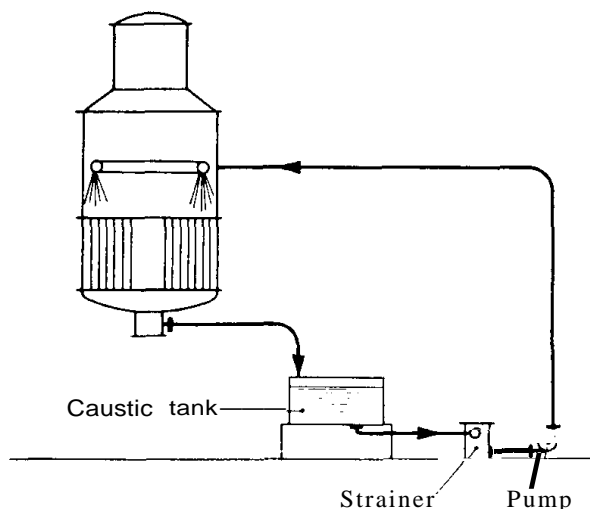


Fig. 32.38. Washing by sprays.

Tromp⁵⁷ prefers heating by a special coil, and recommends closing the vacuum valve and the valves for incondensable gases, and releasing through the top of the vessel the gases formed, in order to avoid any corrosion of the following calandria. He also advises never to enter with a naked lamp a vessel where acid has been boiling, since it may form explosive gases³⁷.

After rinsing with water, steam will always be opened into the calandria: the softer scales are then dried, cracked, and fall off in the last vessel. They are readily removed in the other three vessels by brushing.

Finally, the bottom of the vessel will be cleaned, removing the loose scale which has fallen to the bottom.

The solutions are kept in tanks (of cast iron for the hydrochloric acid). The soda solution is regenerated by adding further soda to bring it up to the desired concentration. It thus can serve about a dozen times. The acid, on the other hand, rapidly loses its efficacy and generally serves for one treatment only. Honig⁵⁸ recommends that its pH be controlled, to a value below 3 before use, and that it should not rise above 5.5 in the course of the cleaning. He also advises, for handling this solution, that piping of polyvinyl chloride should be used.

Inhibitors. The addition of 5% of molasses to the acid solution used reduces the loss of metal by acid attack in the proportion of 20 or 40 to 1 in the case of steel tubes⁵⁹. With brass tubes, the loss by corrosion is much lower. Some factories use formalin, in the ratio of 6l/100 kg (0.6 gal./100 lb.) of HCl.

Prevention of scaling

There are two processes which have the object of avoiding or more accurately decreasing the formation of scale. These are:

- (A) "Ionisation" apparatus
- (B) Addition products.

(A) **Ionisation apparatus.** These are two in number: one of Belgian fabrication, the "C.E.P.I."; the other of English manufacture, the "Superstat".

Both of these consist of a tube, placed in the limed-juice pipe passing to the heaters, and surrounded for example by a solenoid carrying an electric current. The molecules of mineral salts dissolved in the juice thus become "ionised", and would tend to remain in suspension rather than depositing on the heat-exchange surfaces. One of these two devices produces an electric field, the other a magnetic field.

These types of apparatus do not seem to have given very definite results, and are not widely used. It was recommended⁶⁰ to mount them in a vertical position and to clean them every year with a 5% solution of HCl to remove the particles of iron adhering to the magnet, the presence of which would reduce the efficiency of the apparatus.

(B) Addition products. There are used:

(a) **Tetraphosphoglucosate of lime.** This would avoid a large proportion of the deposits in the multiple effects. It is introduced at the outlet from the clarifier, dissolved in water in the ratio of 60 g/l (6 lb./10 gal.) and introduced in the ratio of 0.5 - 1 kg (1 - 2.2 lb.) per 100 t.c.⁶¹. Other dosages recommended are: in Cuba⁶², 100 kg (234 lb.) of tetraphosphoglucosate of lime is dissolved in 180 l (40 gal.) (0.58 kg/l) of hot water from the condenser, and of this (1) 80 cm³/t.c. is placed in the juice passing to the pre-evaporator; (2) the same quantity is passed into the juice entering the 4th vessel. This factory reports excellent results from this method.

At Laupahoe⁶³, 20 g (0.04 lb.) of tetraphosphoglucosate of lime is used per t.c. A solution is made up of 10.8 kg (24 lb.) in 455 l (100 gal.) of water (24 g/l), and after standing for 8 hours, the solution is added to the clarified juice by means of a small copper tube 3 mm (0.12 in.) in diameter, with a cock for regulating the rate of flow. Results have been excellent.

(b) The Wright Chemical Corp. supplies a similar product under the name of Wrikorg C. Knox⁶⁴ recommends the following proportions:

1st vessel:	115 g/100 t.c.	($\frac{1}{4}$ lb./100 t.c.)
2nd vessel:	225 g/100 t.c.	($\frac{1}{2}$ lb./100 t.c.)
3rd vessel:	225 g/100 t.c.	($\frac{1}{2}$ lb./100 t.c.)
4th vessel:	340 g/100 t.c.	(1 lb./100 t.c.)

	0.9 kg/100 t.c.	(2 lb./100 t.c.)

Often, however⁶⁵, this has not allowed chemical cleaning or mechanical descaling to be dispensed with.

(c) E. F. Drew and Co. manufacture a polyphosphate of soda, known as "APA-M", for the same purpose. We have heard of this only in beet factories⁶⁶, where a dosage of 8 p.p.m. is recommended.

(d) The most spectacular results are obtained with finely powdered magnesia: "Magox" "Cosmag", etc. (*cf.* p. 429).

OPERATION IN PARALLEL

When a factory is modified to increase its capacity, it is often convenient to place 2 or 3 evaporator vessels in parallel. For example, 2 small vessels may be connected so as to make one effect of the desired heating surface (Fig. 32.39).

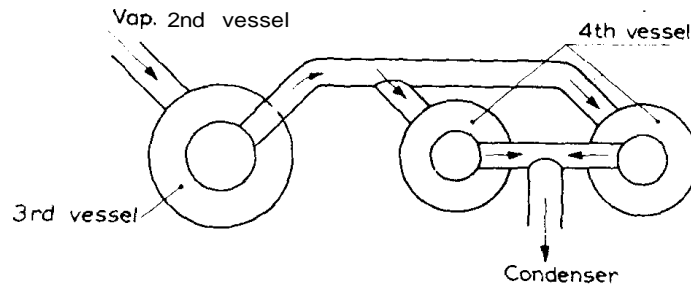


Fig. 32.39. Parallel arrangement of two vessels to make a fourth effect.

The preceding effect will then supply its vapour simultaneously to the 2 calandrias. The 2 vapour outlets will be recombined to pass to the following effect.

It is preferable in this case to adopt the following precautions:

(1) Vapour. Generally, the vapour is simply led into the 2 vessels from the preceding vessel by a Y-piece. However, Claassen¹ recommends that the vapour should preferably be passed across all the first calandria before entering the second. This permits of a better withdrawal of gases, a better distribution around the tubes, and a greater velocity of vapour. Hence an improvement in the heat-transfer coefficient.

It goes without saying that such a solution is possible only on condition that the vapour entry to the vessel can be made of the desired cross-section, and that the incondensable-gas withdrawal points can be located at the new dead-points which will now obtain.

(2) Juice. It is of great advantage, in the operation of 2 or 3 vessels in parallel, to pass the juice in series through the vessels, with circulation on the Chapman system, for example. In this way, a methodical concentration is obtained, and the optimum condition is approached which we have already indicated (p. 529), that is, to ensure that the brix of the juice in a vessel should be the mean between the brix of the juice entering and the brix of the juice leaving. This avoids the too rapid traverse of a vessel in short-circuit by part of the juice, while other parts of the vessel concentrate a juice which does not circulate.

Since there is no difference in pressure to make the juice pass from one vessel to the following one, it may be left to flow simply by gravity. It is necessary to allow for this by giving to the interconnecting juice pipe between the vessels a cross-section corresponding to a juice velocity of 50 cm/s (20 in./s) as a maximum.

If the 2 vessels placed in parallel have tubes of different lengths, the vessel with the longer tubes should preferably be placed ahead of the other. It is desirable then that the point on the tubes corresponding to the mean hydrostatic level of the juice (generally at $\frac{1}{3}$ the length of the tubes from the bottom) should be at the same level in the 2 vessels, or preferably slight-

ly higher in the first vessel. If this is not the case naturally, one of these vessels will be slightly raised.

One should not lose sight of the fact that flash takes place entirely in the former vessel. Since on the other hand the brix is lower in this vessel, it will furnish more vapour than the following vessel or vessels.

(3) Condensates. Since the calandrias of vessels in parallel are under the same pressure, their condensates may be recombined in the same vessel. However, it is preferable to evacuate them separately: in that way a better account can be kept of the respective evaporation of each vessel.

When these various precautions are observed, a multiple effect having its effects or some of its effects formed of 2 or more vessels in parallel, will certainly operate as well as one with single vessels.

Placing vessels in parallel is thus an excellent solution, which may be adopted whenever practicable.

VAPOUR BLEEDING

Originally, for greater simplicity, all vessels of a multiple effect were made of equal size, and each one supplied vapour simply to the following vessel.

However, even when the vessels are of equal size, it is possible to take a certain quantity of vapour from each of them, for use at the heaters. We shall see later how to determine the quantities of vapour which can thus be used (*cf.* p. 579).

This operation is most advantageous, from the point of view of heat economy. If in a heater or a vacuum pan, the exhaust steam used for heating is replaced by vapour from juice taken from one of the vessels of the multiple effect, an economy will be realised which is all the more important when the vapour is taken from a vessel further on in the set. If the vapour is taken from the last vessel, the saving is complete, since the vapour used would otherwise be lost, and moreover the load on the condenser is at the same time reduced (*cf.* p. 565).

Since the bled vapour is at a lower temperature than the exhaust steam, a larger heat-exchange surface will obviously be required and, further, it will not be able to raise the material to be heated to a temperature above its own. On the contrary, it is advisable to allow a certain margin between the temperature of the vapour used and the temperature of the juice leaving the heater. We have already indicated (Table 30.1, p. 457) the value to be given to this margin in practice.

The multiple effect thus offers a complete range of vapour temperatures, by means of which there may be arranged a scheme of heating cold juices which will permit them to be heated progressively to the boiling temperature necessary for defecation, while utilising at each stage the maximum amount of low-temperature vapours, and in consequence achieving the maximum economy.

We shall see later (p. 566) how this economy may be calculated.

VAPOUR RECOMPRESSION

Turbo-compressor

The turbo-compressor, sometimes described by the picturesque term "heat pump", is a centrifugal compressor which allows of increasing to a certain extent the pressure of a vapour and consequently its temperature.

In the sugar factory, this device permits an evaporator vessel to be heated by the vapour from its own juice. It is in fact sufficient to pass the latter vapour through the compressor in order to give it the excess heat necessary to be able to heat the juice from which the vapour is obtained.

Imagine, for example, an evaporator the vapour space of which is at 100°C (212°F) (Fig. 32.40). The juice will boil at about $100-100.5^{\circ}\text{C}$ ($212-213^{\circ}\text{F}$). Take this vapour from the juice at 100°C , which in consequence is at atmospheric pressure (1 kg/cm^2 ; 14.7 p.s.i.a.), and pass it through a compressor which will raise it to 1.3 kg/cm^2 (18.1 p.s.i.a.). The temperature corresponding to this new pressure is 106°C (223°F), and the vapour thus compressed and recirculated to the calandria will now be able to evaporate afresh the juice in the vessel at $100-100.5^{\circ}\text{C}$.

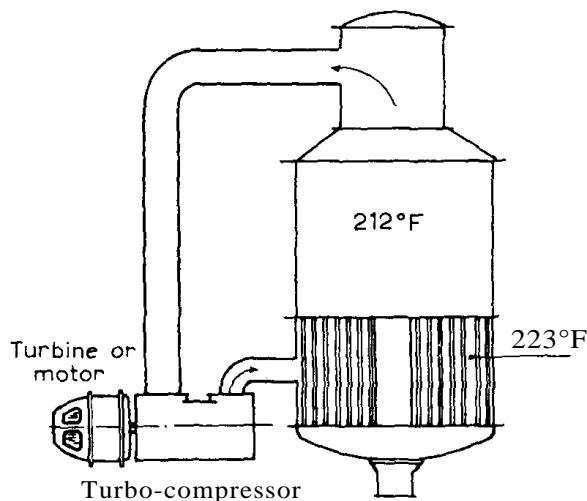


Fig. 32.40. Turbo-compressor.

The turbo-compressor is driven by a steam turbine or an electric motor (motor-compressor). The apparatus is obviously not of interest unless under the condition that the steam consumption of this turbine, or the extra consumption of the turbo-alternator supplying the current to the motor, is lower than the quantity of vapour produced by the evaporator. This applies for a vessel operating in single effect, such as a vacuum pan. However, it is even necessary, for comparison with a multiple effect, that the ratio between these two quantities should be approximately of the same order as the ratio between the heating steam and the total vapour produced from the juice in a triple or quadruple effect, that is to say, $1 : 3$ or $1 : 4$.

The turbo-compressor can attain an entrainment ratio (ratio of weight of vapour compressed to weight of steam consumed) of 5, whereas the alternative device, the thermo-

compressor, which we are about to study, hardly ever exceeds 2.5. It can function, for example, with actuating steam at 30 or 40 kg/cm² (425 - 570 p.s.i.), and take from the evaporator vessel vapour at 0.7 kg/cm² (10. p.s.i.) for re-injection into the calandria at 1.1 kg/cm² (15 p.s.i.). Its efficiency, moreover, is higher, as the difference between the initial pressure of vapour and the pressure of the compressed vapour is lower.

It has the advantage of operating without consumption of heat, but obviously requires consumption of energy. It may become of interest when there is high-pressure steam to spare, and it then bears comparison with a quintuple effect. However, it costs about five times as much as a thermo-compressor, and is not widely used. It may become a future solution.

Advantages and disadvantages. While it offers the advantage of taking up less space than an ordinary multiple effect, the turbo-compressor presents on the other hand the serious disadvantage of being a piece of machinery, in other words, of having moving parts, which are expensive, which wear, require lubrication, maintenance, replacement and renewal, and are subject to breakdowns and stoppages. In the beet sugar factory, it is stated that such incidents are rare.

Although the turbo-compressor is fascinating to study from the technical point of view, and probably has a great future, we shall leave it to one side for the moment, since it will probably not pass for some time into the domain of practice, at least in the cane sugar factory; and we shall study an apparatus the object of which is entirely analogous, which does not present the same disadvantages, and which has greater prospects of wider application.

Thermo-compressor

The thermo-compressor has exactly the same object as the turbo-compressor, that is, to raise the value of the vapour coming from the juice and to give it the increment of pressure and temperature necessary to permit it to raise to boiling point the juice from which it has been obtained.

We shall consider the same vessel as just now (Fig. 32.41), and substitute for the turbo-compressor a special nozzle T. If we supply this nozzle with steam at high pressure (10, 14, 18 or 25 kg/cm²; 150, 200, 250 or 350 p.s.i.), this steam will aspirate the vapour from the vessel and return it to the calandria while giving up to it part of its own heat energy. The mixture thus acquires a pressure and a temperature sufficient for boiling the juice in the vessel.

The main difference compared with the turbo-compressor is that the actuating steam is necessarily mixed in the calandria with the aspirated vapour, whereas with the turbo-compressor it is possible to separate the actuating steam and to use it otherwise.

It follows that in addition to the outlet for the recirculated vapour, it is necessary to provide for the vessel another outlet for the vapour from the juice, since the quantity of H.P. steam introduced, like the recirculated vapour, evaporates in its turn an equivalent quantity of water.

If, for example, 1 kg of H.P. steam at 25 kg/cm² can aspirate 3 kg of vapour at 100°C ($p = 1 \text{ kg/cm}^2$) and provide $3 + 1 = 4$ kg of steam at 106°C ($p = 1.3 \text{ kg/cm}^2$), these 4 kg of steam will evaporate about 4 kg of vapour from the juice, 3 kg of which will continue to recycle through the thermo-compressor, and 1 kg must be sent elsewhere, for example,

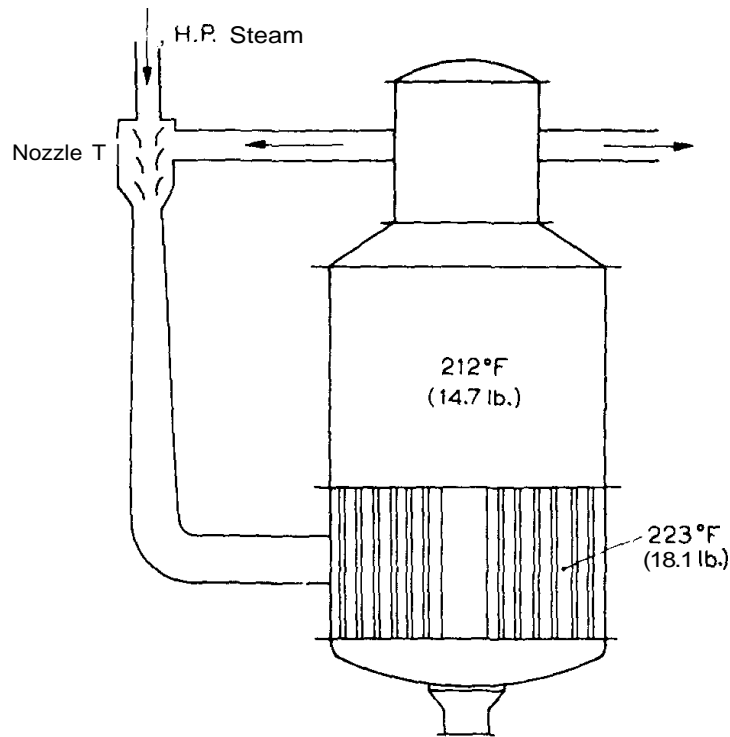


Fig. 32.41. Thermo-compressor.

to the 2nd vessel of a multiple effect of which the vessel considered would serve as the 1st effect. It will be noted in passing that a multiple effect installed in this way with a thermo-compressor attached to the 1st vessel would have a 1st vessel much larger than the following ones, since it would be capable of evaporating 4 times the quantity of vapour.

Actually, the proportion is never as high as this, since, for reasons which we shall see later (p. 603), it is desirable to introduce into the 1st vessel exhaust steam in excess of the quantities furnished by the thermo-compressor. A quantity of vapour from the juice equivalent to the quantity of exhaust steam introduced will obviously be passed entirely to vapour bleeding from the 1st vessel or to the 2nd vessel, together with the quantity of vapour corresponding to the high-pressure steam utilised.

The incondensable gases from a vessel provided with a thermo-compressor can thus be sent into the vapour space of the same vessel. (With the turbo-compressor of Fig. 32.40, this would not have been possible, since the vessel operates in closed circuit, and the gases would accumulate in the circuit.)

It is obviously preferable to evacuate the gases to the atmosphere, if the pressure existing in the calandria permits. Otherwise they would be sent, not into the vapour space, but directly into the vapour pipe going to the following vessel, to avoid risk of being left to recirculate in the closed circuit.

Entrainment ratio. The term "entrainment ratio" of a thermo-compressor signifies the ratio between the weight of vapour aspirated and the weight of actuating steam. This ratio is designated by the letter μ :

$$\mu = \frac{\text{weight of vapour aspirated}}{\text{weight of actuating steam}} \quad (32.17)$$

The **entrainment ratio** is the most important figure in the performance of a thermo-compressor. It may be calculated by the formula of Truffault:

$$\begin{aligned}
 (\mu_0 + 1)^2 &= \frac{80}{t_m - t_0} [(\log P - \log p_m) - 0.01 (t_0 - 100)] \\
 \left((\mu_0 + 1)^2 &= \frac{144}{t_m - t_0} [(\log P - \log p_m) - 0.0056 (t_0 - 212)] \right) \quad (32.18)
 \end{aligned}$$

μ_0 = entrainment ratio with a new nozzle

t_m = temperature corresponding to the absolute pressure p_m of the mixture, *i.e.* of the vapour in the calandria, in °C (°F)

t_0 = temperature corresponding to the absolute pressure p_0 of the vapour from the juice in the vessel, in °C (°F)

P = absolute pressure of the actuating steam, in kg/cm² (p.s.i.a.).

In order to take into account wear of the nozzles, we shall take:

$$p = 0.95 \mu_0 \quad (32.19)$$

According to the efficiency of the nozzle under consideration, the coefficient 80 (144) in formula (32.18) could be modified.

These equations do not take into account the superheat of the actuating steam. They are derived from the analytical relationships which exist between the graphs which can be drawn on the entropy diagram to represent the changes of state of the fluids passing through the nozzle. Now these graphs are practically identical for the pressures given, and change very little with superheat of the steam. In other words, superheat has only a negligible influence, on account of the parallelism of the lines of equal pressure.

Practical values of entrainment ratio. Table 32.12 gives values of the entrainment ratio, derived from eqn. (32.18), for several common values of P , p_m and p_0 .

Sensitivity of the thermo-compressor. On the other hand, the degree of superheat has a very marked influence on the output of this apparatus, on account of its effect on the specific volume of the vapour.

Moreover, the variations in volume are very marked in the zone of recompression (100-110°C; 212-230°F), and all the more so as the aspirated vapour approaches 100°C,

It follows that the temperature of the vapour to be compressed should be determined very accurately before the purchase of the thermo-compressor. If the pressure proves to be different from that for which the equipment has been designed, eddies will be created in the aspiration chamber, which will absorb uselessly a great proportion of the kinetic energy available for recompression.

The thermo-compressor is thus a delicate piece of apparatus, the efficiency of which depends on close adherence to the conditions of operation for which it has been designed.

TABLE 32.12

VALUES OF ENTRAINMENT RATIO

<i>Calandria pressure</i>			<i>Vapour-space pressure</i>			<i>Entrainment ratio for H.P. steam at P (kg/cm²)</i>				
<i>p_m abs.</i> (kg/cm ²)	<i>p_m gauge</i> (kg/cm ²)	<i>t_m</i> (°C)	<i>p₀ abs.</i> (kg/cm ²)	<i>p₀ gauge</i> (kg/cm ²)	<i>t₀</i> (°C)	18	20	25	30	45
2.53	1.5	127	1.88	0.85	117	1.31	1.39	1.54	1.66	1.91
2.03	1.0	120	1.50	0.47	111	1.60	1.67	1.81	1.92	2.15
1.83	0.8	117	1.37	0.34	108	1.79	1.86	2.00	2.11	2.34
1.53	0.5	111	1.16	0.12	103	2.07	2.14	2.27	2.38	2.60
<i>(British units)</i>								<i>(p.s.i.a.)</i>		
<i>(p.s.i.a.)</i>	<i>(p.s.i.g.)</i>	<i>(°F)</i>	<i>(p.s.i.a.)</i>	<i>(p.s.i.g.)</i>	<i>(°F)</i>	250	300	350	400	640
34.7	20	259	26.8	12.1	244	1.51	1.64	1.74	1.83	2.12
29.7	15	250	22.4	7.7	234	1.64	1.76	1.86	1.93	2.20
24.7	10	239	18.2	3.5	223	1.80	1.90	1.99	2.06	2.31
19.7	5	226	15.3	0.6	214	2.56	2.67	2.77	2.85	3.13

Variation of entrainment ratio with difference in temperature. When the pressure p_m remains constant, and the pressure p_0 varies, the efficiency of the nozzle, which is a maximum for a given pressure p_0 , will fall.

Approximately, it may be stated that, when the drop in temperature varies by $\pm x\%$, the value of $(\mu_0 + 1)^2$ will fall by $x/2\%$ in relation to what it would have been if the nozzle had been designed for the temperature drop actually obtaining.

For example, if a nozzle has been designed for a temperature drop of 8°C , and if the corresponding entrainment ratio was 2.4: if this temperature drop increases to 9°C , μ will fall to 2.10 (instead of 2.20); if the temperature drop decreases to 7°C , μ will increase to 2.53 (instead of 2.64).

Limits of use. Theory shows, and practice still more, that in turbo-compression and in thermo-compression, a good efficiency is obtainable only under the following conditions:

(a) A small drop in temperature ($t_m \sim t_0$) between the compressed vapour and the vapour to be compressed, that is, this temperature drop should be not more than 10°C (18°F).

(b) Small elevation of boiling point (since this elevation increases the temperature difference); hence these operations are suitable only for low brix.

In other words, thermo-compression can be employed only in the first effect, or in a pre-evaporator.

Importance of efficiency. In an ordinary factory, the requirements of low-pressure steam for the boiling house are much greater than the quantities of exhaust steam available. Such a factory must reduce a large quantity of live steam to the exhaust pressure to complete the requirements of low-pressure steam. The expansion of this make-up steam is generally effected in a simple reducing valve, without obtaining any benefit from it.

The advantage of the thermo-compressor is that the compressor itself carries out this expansion. Instead of being done uselessly in a reducing valve, the expansion takes place in the thermo-compressor, and each kg of live steam so expanded produces, without cost, evaporation of μ kg of water from the juice. This gain is all the more valuable since it costs absolutely nothing.

The efficiency of the nozzle is then of secondary importance only; however bad it be, there remains still a definite gain.

Installation. In the general case, the thermo-compressor operates in conjunction with the 1st effect, with:

Pressure of the actuating steam P , the pressure of the high-pressure steam in the factory

Pressure of compressed vapour p_m , the pressure of the exhaust steam

Pressure of vapour before compression p_0 , the pressure of the 1st effect.

With a pre-evaporator:

The pressure P remains that of the high-pressure steam

The pressure p_m becomes a pressure intermediate between that of the exhaust steam and P , and generally $0.4\text{--}0.5 \text{ kg/cm}^2$ (6-7 p.s.i.) higher than the exhaust-steam pressure

The pressure p_0 becomes the exhaust-steam pressure.

Steam economy due to thermo-compression. The economy realised by the installation of a thermo-compressor corresponds, as we have just discussed, to the additional quantity of water evaporated due to it in the 1st vessel or the pre-evaporator.

However, this quantity is limited by the maximum quantity of live steam which can be introduced into the calandria of the 1st effect or into the exhaust-steam system without having an excess of exhaust steam, which would then be discharged above the roof by the safety valve.

In other words, a thermo-compressor can only be used to the extent of the make-up steam necessary for manufacture. If a factory is producing from its engines as much exhaust steam as it consumes in the boiling house, thermo-compression would offer no interest for it.

We shall see later (p. 603) the possibilities of thermo-compression as a function of the quantities of exhaust steam produced and consumed.

OPERATION OF MULTIPLE EFFECTS

Relative evaporations in the various effects

We shall consider a quadruple effect without vapour bleeding, functioning between 112°C (234°F), as the temperature of the exhaust steam, and 55°C (131°F), as the temperature corresponding to the vacuum obtainable, and receiving juice at 98°C (208°F). What would be the quantity of vapour supplied by each effect, per unit weight of exhaust steam supplied to the calandria of the 1st effect?

We shall calculate for two extreme cases:

- With separate extraction of condensates from each vessel
- With circulation of the condensates from the 1st to the last effect.

We shall assume that the juice entering the 1st effect is 5 times the weight of steam entering, and that the temperatures are those indicated in the diagram given (Fig. 32.42).

The calculations are given in metric units, for purposes of illustration, in the belief that the method of calculation can still be readily followed by engineers more conversant with the British system of units.

- Separate extraction of condensates from each vessel.** The quantity of vapour furnished

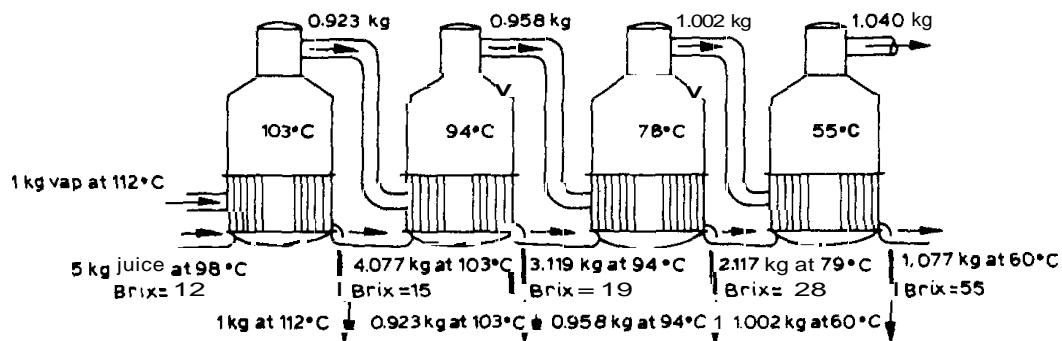


Fig. 32.42. Separate extraction of condensates from each vessel.

by each effect is obtained by writing a heat balance for each effect; that is, the quantity of heat entering the vessel is equal to the quantity of heat leaving it.

We shall assume that the quantities of heat lost by radiation and convection in each vessel are respectively 1.25, 1, 0.75, 0.50% of the heat exchanged in that vessel.

We shall assume further that the juice entering is at 12° brix, and note that its specific heat is calculated by eqn. (30.1).

For the 1st effect, for example, we have:

(1) *Heat entering:*

$$\begin{array}{rcl}
 1 \text{ kg of steam at } 112^\circ: \text{ total heat (Table 41.1)} & = & 643 \text{ kcal} \\
 5 \text{ kg of juice at } 98^\circ: 5 \times 98 \times 0.93 & = & 456 \text{ kcal} \\
 & & \hline
 & = & 1,099 \text{ kcal}
 \end{array}$$

(2) *Heat leaving:*

$$\begin{array}{rcl}
 \text{Loss of heat: } 1,099 \times 0.0125 & = & 14 \text{ kcal} \\
 \text{Condensate: } 1 \text{ kg at } 112^\circ & = & 112 \text{ kcal} \\
 \text{Vapour: } x \text{ kg at } 105^\circ & = & 640x \text{ kcal} \\
 \text{Juice: } (5 - x) \text{ kg at } 103^\circ & = & (5 - x) 103 \times 0.91
 \end{array}$$

We have then:

$$14 + 112 + 640x + (5 - x) 103 \times 0.91 = 1,099$$

whence:

$$x = 0.923 \text{ kg}$$

Proceeding in the same way from one vessel to the next, for the 2nd effect and the following ones, we obtain the figures given in Table 32.13.

The heat is supplied to the 1st effect. Heat exchange takes place from one vessel to the next but decreases from the 1st to the last, each vessel losing compared with the preceding one not only the heat given up to the surroundings but also that taken away by the condensate.

(b) Circulation of condensates from vessel to vessel. (See Fig. 32.43). By calculating exactly as in the foregoing, we obtain the figures of Table 32.14.

Here each vessel loses compared with the preceding one only the heat given up by it to the surroundings: the last vessel therefore transfers a quantity of heat only slightly inferior to that exchanged by the first.

If we summarise the evaporation done by the quadruple in the two extreme cases examined, we obtain Table 32.15.

We see then that:

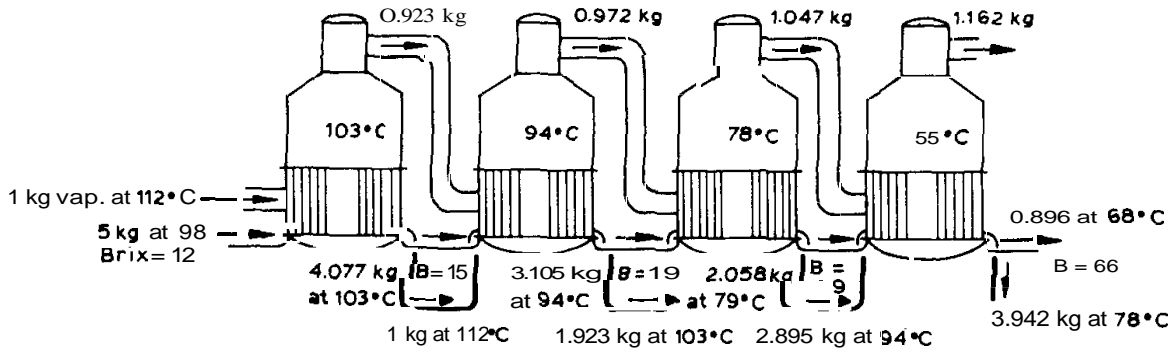


Fig. 32.43. Circulation of condensates from vessel to vessel.

(1) The complete circulation of condensates permits of a gain of about 5% in evaporation, relative to that with separate extraction of condensates

TABLE 32.13

Heat entering		Heat leaving	
<i>2nd effect</i>			
Vapour: 0.923 kg x 640 kcal = 591 kcal		Loss: 973 x 0.01 = 10 kcal	
Juice: 4.077 x 103° x 0.91 = 382		Water: 0.923 x 103° = 95	
		Vapour: 0.958 x 637 kcal = 610	
		Juice: 3.119 x 94° x 0.88 = 258	
	973		973
<i>3rd effect</i>			
Vapour: 0.958 kg x 637 kcal = 610 kcal		Loss: 868 X 0.0075 = 7 kcal	
Juice: 3.119 x 94° x 0.88 = 258		Water: 0.958 x 94° = 90	
		Vapour: 1.002 x 631 = 632	
		Juice: 2.117 x 79 x 0.83 = 139	
	868		868
<i>4th effect</i>			
Vapour: 1.002 kg x 631 kcal = 632 kcal		Loss: 771 x 0.0050 = 4 kcal	
Juice: 2.117 x 79° x 0.83 = 139		Water: 1.002 x 78° = 78	
		Vapour: 1.040 X 621 = 646	
		Syrup: 1.077 x 60 x 0.67 = 43	
	771		771

(2) With or without circulation of condensates, a quadruple effect evaporates very close to 4 units of water per unit weight of steam supplied to the calandria of the 1st effect.

A similar relationship, close to 1 unit of water per effect, would apply for a triple or a quintuple. We may therefore enunciate the following approximate relationship:

Unit weight of heating steam introduced to the calandria of the first vessel of a multiple effect without vapour bleeding will produce an evaporation of unit weight of water from the juice in each body of the multiple effect.

Hence this corollary immediately follows:

A multiple effect of n effects without vapour bleeding evaporates n units of water from the juice per unit weight of heating steam supplied to the set.

TABLE 32.14

<i>Heat entering</i>		<i>Heat leaving</i>	
<i>1st effect</i>			
Steam: 1.000 kg at 112°	= 643 kcal	Loss: 1.099 x 0.0125	= 14 kcal
Juice: 5.000 x 98 x 0.93	= 456	Water: 1.000 x 112°	= 112
		Vapour: 0.923 x 640 kcal	= 591
		Juice: 4.077 x 103 x 0.91	= 382
	1,099		1,099
<i>2nd effect</i>			
Vapour: 0.923 kg x 640 kcal	= 591 kcal	Loss: 1.085 x 0.01	= 11 kcal
Juice: 4.077 x 103 x 0.91	= 382	Water: 1.923 x 103°	= 198
Water: 1.000 x 112°	= 112	Vapour: 0.972 x 637 kcal	= 619
		Juice: 3.105 x 94 x 0.88	= 257
	1,085		1,085
<i>3rd effect</i>			
Vapour: 0.972 kg x 637 kcal	= 619 kcal	Loss: 1.074 x 0.0075	= 8 kcal
Juice: 3.105 x 94 x 0.88	= 257	Water: 2.895 x 94°	= 272
Water: 1.923 x 103°	= 198	Vapour: 1.047 x 631	= 661
		Juice: 2.058 x 79 x 0.82	= 133
	1,074		1,074
<i>4th effect</i>			
Vapour: 1.047 x 631 kcal	= 661 kcal	Loss: 1.066 x 0.0050	= 5 kcal
Juice: 2.058 x 79 x 0.82	= 133	Water: 3.942 x 78°	= 307
Water: 2.895 x 94°	= 272	Vapour: 1.162 x 621	= 722
		Syrup: 0.896 x 60 x 0.6	= 32
	1,066		1,066

TABLE 32.15

EVAPORATION IN EACH EFFECT PER UNIT OF STEAM SUPPLIED TO FIRST VESSEL

	<i>Without circulation of condensates</i>	<i>With circulation</i>	<i>% gain by circulation</i>
Steam supplied	1	1	
Water evaporated by 1st effect	0.923	0.923	0
Water evaporated by 2nd effect	0.958	0.972	1.5
Water evaporated by 3rd effect	1.002	1.047	4.5
Water evaporated by 4th effect	1.040	1.162	11.2
Total evaporation	3.923	4.104	4.6

Influence of the temperature of the juice entering. In the preceding calculation it has been ascertained that each vessel actually supplies slightly more vapour than the preceding one. This excess is due to three main causes:

(1) *Juice.* Liberation of a quantity of sensible heat of the juice corresponding to the difference in temperature between vessels. Hence flash in the following vessel.

(2) *Vapour.* A progressive decrease of the total heat of the vapour with pressure in the vessel, the increase in latent heat of vaporisation being less rapid than the decrease in sensible heat.

(3) *Water.* In the case of circulation of condensates, the drop in pressure similarly causes a flash of the water, which increases the quantity of heating fluid on the vapour side of the calandria.

The effect of differences in specific heat is not all in the one direction and, moreover, it is small. The heat lost, which is of minor importance, is always largely compensated by the three causes above.

The 1st vessel only gives out less vapour than it receives: 0.923 kg in our example, in place of 1 kg. This is due to the low temperature (98°C) of the juice entering: it is necessary first to heat this juice before evaporating it. If the juice were to enter at the same temperature as the vapour, as in the other vessels, we should find that the 1st vessel would evaporate 1.042 kg in place of 0.923 kg, and all the following vessels would react favourably to this.

We see here the importance of the initial temperature of juice entering a multiple effect, from the point of view of the evaporation to be obtained.

Evaporation with vapour bleeding

We shall pass on to the general case of a multiple effect with vapour bleeding from one or more vessels. The vapour bleeding obviously does not modify at all the equilibrium which we have just indicated between the quantity of vapour admitted into each calandria and the quantity of water which it evaporates in that vessel. We shall assume then that, in each vessel, each unit of vapour arriving in the calandria evaporates an equal weight of water.

We shall take then a quadruple effect (Fig. 32.44) from the vessels of which vapour is bled in the following quantities per t.c.:

P_1 from the 1st vessel

P_2 from the 2nd vessel

P_3 from the 3rd vessel

P_4 from the 4th vessel

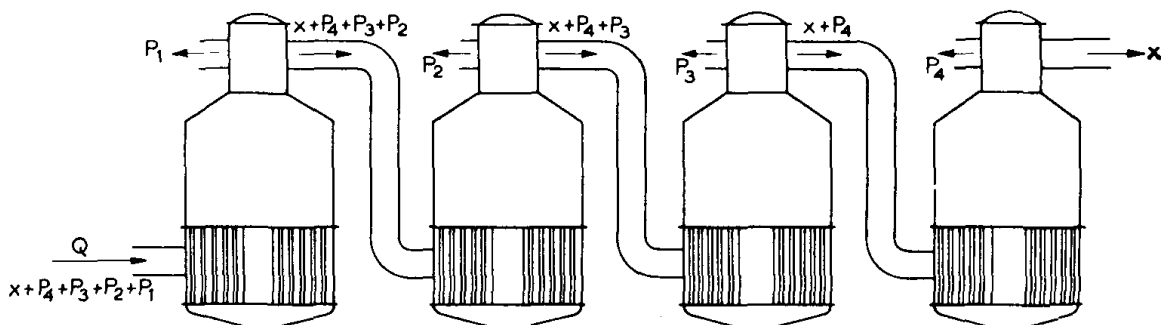


Fig. 32.44. Quadruple effect with vapour bleeding.

Now let:

E = total quantity of water evaporated per t.c. in the quadruple effect

x — the quantity of vapour passing from the last effect to the condenser.

Since the last vessel evaporates a weight of $(x + P_4)$, it should receive from the third an equal quantity. The third, evaporating $(x + P_4) + P_3$, should receive from the second the same quantity $(x + P_4 + P_3)$. The first should in the same way supply to the second $x + P_4 + P_3 + P_2$ and it will be necessary to supply to it, as exhaust steam:

$$Q = x + P_4 + P_3 + P_2 + P_1$$

The total evaporation E achieved by the quadruple effect will therefore be:

$$\begin{array}{rcl} \text{4th effect:} & x + & P_4 \\ \text{3rd effect:} & x + & P_4 + P_3 \\ \text{2nd effect:} & x + & P_4 + P_3 + P_2 \\ \text{1st effect:} & x + & P_4 + P_3 + P_2 + P_1 \\ \hline \end{array}$$

$$\text{Multiple effect: } E = 4x + 4P_4 + 3P_3 + 2P_2 + P_1 \quad (32.20)$$

We see then that the quantity of water evaporated by the quadruple effect is equal to:

$$\begin{array}{l} 4 \text{ times the quantity evaporated by the 4th vessel} \\ + 3 \text{ times the quantity withdrawn from the 3rd vessel} \\ + 2 \text{ times the quantity withdrawn from the 2nd vessel} \\ + 1 \text{ times the quantity withdrawn from the 1st vessel} \end{array}$$

Economy obtained by vapour bleeding

Vapour bleeding is carried out for use in juice heaters or vacuum pans. If there is no vapour bleeding from the multiple effect, the heaters and the pans have to be heated with either direct or exhaust steam. Let:

R = the quantity of steam (direct or exhaust) consumed by the heaters and pans

Q = the quantity of steam (direct or exhaust) consumed by the evaporators.

In the case of a quadruple without vapour bleeding, the steam used will be:

$$D = R + Q$$

If we bleed vapour from the quadruple to supply the heaters or to boil the pans, each unit of such vapour will save close to 1 unit of exhaust steam, since the latent heat is approximately the same, and since we assume, obviously, that the bled vapour has a temperature sufficient to effect the heating or the stage of heating which is expected of it.

If the heaters or pans which previously required a weight R of direct or exhaust steam are now heated by vapour from the juice, we shall have:

$$P_1 + P_2 + P_3 + P_4 = R \quad (32.21)$$

and the new steam consumption, D' , will be reduced to the quantity supplied to the 1st vessel of the evaporators, say Q' . Hence:

$$D' = Q'$$

Now:

$$Q' = x + P_1 + P_2 + P_3 + P_4 \quad (32.22)$$

Eliminating x , which does not interest us, from eqns. (32.20) and (32.22), and expressing the steam consumption $D' = Q'$ as a function of the total evaporation E , which is fixed and readily determined if we fix the brix of the syrup, we have:

$$Q' = \frac{E}{4} + \frac{3}{4}P_1 + \frac{2}{4}P_2 + \frac{1}{4}P_3 \quad (32.23)$$

Without vapour bleeding, the steam consumption was:

$$D = Q + R = \frac{E}{4} + P_1 + P_2 + P_3 + P_4 \quad (32.24)$$

The economy realised e is therefore:

$$e = D - D' = (Q + R) - Q' = \frac{3}{4}P_1 + \frac{2}{4}P_2 + \frac{1}{4}P_3 + \frac{4}{4}P_4 \quad (32.25)$$

If we had made the calculations for the general case of a multiple effect with n effects, we should have found:

$$e = \frac{1}{n}P_1 + \frac{2}{n}P_2 + \frac{3}{n}P_3 + \dots + \frac{p}{n}P_p + \dots + \frac{n-1}{n}P_{n-1} + P_n \quad (32.26)$$

In other words: *when a multiple effect of n effects is available and when, for some heating duty in the factory, there is substituted for exhaust steam vapour bled from effect No. p , a saving in steam is realised equal to the fraction p/n of the quantity of steam consumed by this heating duty.* The saving increases then with the place of the vessel in the multiple effect, and becomes a complete saving when the bleeding is made from the last vessel.

Unfortunately, the temperature of the vapour from the last vessel (of the order of 55°C (130°F)) scarcely offers any possibilities for heating. Hence the advantage of multiple effects under pressure, where the vapour from the last vessel is in the neighbourhood of 100°C , and in which it is possible in consequence to utilise practically the whole of the vapour from the juice.

In an ordinary multiple effect, one must generally be content with bleeding vapour from the 1st and 2nd vessels. It is also possible to effect a slight heating of cold juices by passing them through a heater-condenser interposed in the vapour pipe from the last vessel (and also perhaps from the pans) to the condenser.

Heating surface. We may comment in passing that it is not possible, in a heater, to obtain with bled vapour the same outlet juice temperature as with exhaust steam for example. To obtain this temperature (if the temperature of the vapour used will permit), a heater of greater heating surface is necessary.

Economy obtained by thermo-compression

We shall now compare, from the point of view of steam consumption, a multiple effect furnished with a thermo-compressor with one not so equipped.

Since vapour bleeding does not come into this question, we shall suppose, to simplify the problem, that there is no vapour bleeding (Fig. 32.45).

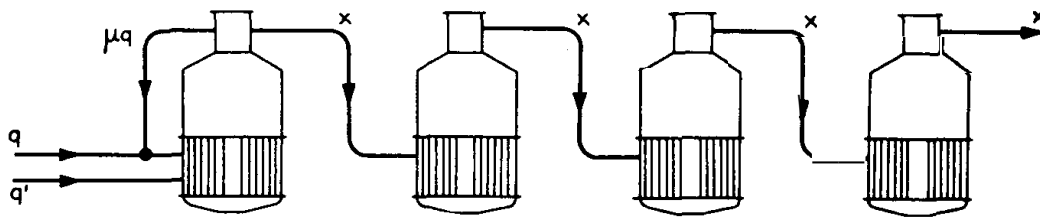


Fig. 32.45. Quadruple effect with thermo-compressor.

Without the thermo-compressor the steam consumption is:

$$D = \frac{E}{4}$$

With the thermo-compressor, this becomes (Fig. 32.45):

$$D' = q + q'$$

q = consumption of actuating steam by the thermo-compressor

q' = quantity of exhaust steam introduced directly into the calandria.

Equating the quantity of steam for heating the 1st vessel to the quantity of water evaporated, we have:

$$q + q' + \mu q = \mu q + x \quad \text{hence:} \quad q + q' = x$$

but the evaporation is:

$$E = 4x + \mu q$$

hence:

$$D' = q + q' = \frac{q}{4} + \frac{3q}{4} = q \quad (32.27)$$

Hence the saving:

$$e = D - D' = \frac{\mu q}{4}$$

or in general:

$$e = \frac{\mu}{n} q \quad (32.28)$$

e = saving due to the thermo-compressor

μ = entrainment ratio of the thermo-compressor

q = quantity of direct steam consumed by the thermo-compressor

n = number of effects.

This saving is added to that from vapour bleeding (eqn. (32.26)) in the case where we have at the same time vapour bleeding and thermo-compression.

Equivalent number of effects with thermo-compression. Let us assume that all the vapour introduced to the multiple effect is the actuating steam of the thermo-compressor: $q' = 0$ (Fig. 32.46).

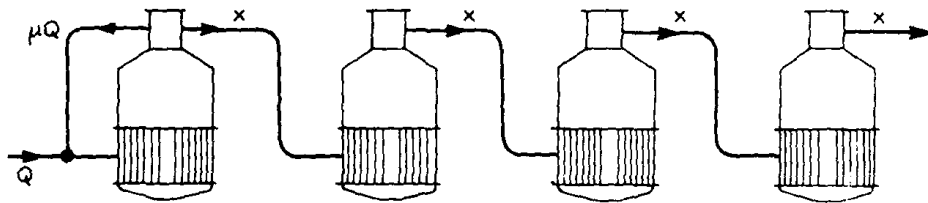


Fig. 32.46. Equivalent number of effects.

The 1st vessel then gives:

$$Q + \mu Q = x + \mu Q$$

or:

$$x = Q$$

The evaporation is given by the expression:

$$E = 4x + \mu Q = 4Q + \mu Q$$

In the general case:

$$E = nQ + \mu Q = (n + \mu) Q \quad (32.29)$$

The actual number of effects of a multiple set being represented by the ratio E/Q , we see that a multiple with n effects, furnished with a thermo-compressor of entrainment ratio μ , is actually equivalent to an ordinary multiple effect with a number n' of effects:

$$n' = \frac{E}{Q} = n + \mu \quad (32.30)$$

since 1 kg of steam admitted to the calandria produces an evaporation of $(n + \mu)$ kg of water from the juice.

In practice, it is not possible to reduce the admission of steam to the 1st vessel to the value of the actuating steam of the thermo-compressor (*cf.* p. 603). It is not possible, therefore, to reduce q' to 0. We should say then that the portion of the steam q' , represented by exhaust steam, admitted to the calandria of the multiple effect, works at n effect, and the portion q , representing the actuating steam of the thermo-compressor, at $(n + \mu)$ effect.

HEATING SURFACE

As for heaters (*cf.* p. 461), the heating surface of an evaporator vessel should logically be calculated from the interior surface of the tubes. It would be appropriate in calculating this surface to use the length of the tubes which is actually heated, *i.e.* the height included between the inside faces of the tube plates, and to add to this the heated surface of the tube plates themselves, *i.e.* the area of the plates, deducting the area of the holes drilled for the passage of the tubes.

This calculation would be somewhat lengthy, so the general practice is to calculate the surface for the inside of the tubes, reckoned on their length outside the tube plates, the excess length thus included compensating very closely for the remaining area of the tube plates which is neglected.

We may recall that English and American engineers calculate the heating surface according to the area of tubes in contact with the heating fluid, that is, they use the exterior surface area of the tubes. It follows that the heating surface of a given heater or evaporator vessel is not given by a simple conversion to square metres of the area in square feet quoted by English or American manufacturers. Although 1,000 sq.ft. is equal to 92,90 m², an American vessel of 10,000 sq.ft., for example, would be equivalent to a French vessel of 852 m² and not of 929 m², or approximately 8% less.

Heat-transfer and evaporation coefficients

There are several figures for comparing the work of the evaporators. We shall distinguish:

(a) *The evaporation coefficient.* This is the weight of vapour furnished by the vessel per hour per unit heating surface ($\text{kg}/\text{m}^2/\text{h}$; $\text{lb.}/\text{sq. ft.}/\text{h}$).

(b) *The specific evaporation coefficient.* This is the weight of vapour supplied by the evaporator per hour, per unit heating surface and per degree drop of temperature between steam and juice ($\text{kg}/\text{m}^2/^\circ\text{C}/\text{h}$; $\text{lb.}/\text{sq. ft.}/^\circ\text{F}/\text{h}$). We sometimes distinguish between the apparent specific evaporation coefficient, in which the juice temperature is taken as equal to that of the vapour produced from it, and the real coefficient, which takes into account the actual temperature of the juice in the tube, obtained by adding to the vapour temperature (1) the boiling-point elevation due to the brix, and (2) the elevation due to hydrostatic pressure.

(c) *The uncorrected heat-transfer coefficient.* This is the number of heat units transmitted per hour per unit heating surface and per degree drop in temperature ($\text{kcal}/\text{m}^2/^\circ\text{C}/\text{h}$; $\text{B.T.U.}/\text{sq. ft.}/^\circ\text{F}/\text{h}$). Here also we distinguish between apparent and real coefficients, according as we use the apparent temperature drop, or the real temperature drop, as for the specific evaporation coefficient.

(d) *The corrected heat-transfer coefficient.* The heat transmission is more rapid according as (1) the steam is at a higher pressure, (2) the juice is at a lower brix in the vessel. If it is desired to compare evaporator vessels with each other, and to know if there is one whose heat transmission is abnormally bad, it is necessary to take into account these last two factors. Unfortunately, different people are not in agreement on the exact mode of allowing for this. Nevertheless, the corrected coefficient of heat transmission is the uncorrected coefficient reduced, by means of formulae taking into account these two factors, to a value which should be common to all identical vessels, working in the same state of cleanness, whatever be the steam pressure and the brix of the juice.

In the same way the evaporation coefficient may be modified to give a corrected specific evaporation coefficient.

Heat-transfer coefficients for the various vessels. In calculations, it is not necessary to start from corrected coefficients. Certain manufacturing firms design their projects by assuming *a priori* a certain heat-transfer coefficient for each vessel of a multiple effect. The values vary greatly according to the various business firms and their respective ideas. Table 32.16 gives the commonest mean values, taking scaling into account, and based on *apparent* temperature drops.

The values given are mean and conservative figures, to be adopted for design purposes, and are often exceeded in practice. Webre⁶⁷ gives coefficients ranging from 6800 $\text{kcal}/\text{h}/\text{m}^2/^\circ\text{C}$ (1,400 $\text{BTU}/\text{h}/\text{sq. ft.}/^\circ\text{F}$) for first vessels heated by steam at 115°C (240°F), to 2200 (450) for last vessels heated by vapour at 65°C (150°F).

We do not recommend the use of these uncorrected heat-transfer coefficients, which take

into account neither the actual brix of juice in the vessel, nor the actual value of the temperature. They are "omnibus" coefficients established for mean values of these two factors. Their only advantage is that they avoid the calculation of brix figures and are based on apparent temperature drops, which simplifies the calculation.

TABLE 32.16

APPARENT HEAT-TRANSFER COEFFICIENTS

(1) *Pre-evaporator* heated by direct steam at 2.5–3.0 kg/cm² (35–43 p.s.i.g.), the pressure in the vessel being approx. 0.5–0.7 kg/cm² (7–9 p.s.i.g.): 3,000 kcal/m²/°C/h) (600 B.T.U./sq.ft./°F/h).

(2) *Multiple effect.*

	<i>(kcal/m²/°C/h)</i>			<i>(B.T.U./sq.ft./°F/h)</i>		
	<i>Triple</i>	<i>Quadruple</i>	<i>Quintuple</i>	<i>Triple</i>	<i>Quadruple</i>	<i>Quintuple</i>
1st vessel	2,000–2,500	2,000–2,400	2,000–2,400	400–500	400–500	400–500
2nd vessel	1,500–1,800	1,400–1,800	1,400–1,800	300–375	275–375	275–375
3rd vessel	800–1,000	1,000–1,400	1,300–1,500	150–200	200–275	250–300
4th vessel		600–800	800–1,000		125–150	150–200
5th vessel			500–600			100–125

Dessin formula. Conscious of these disadvantages, the French engineer Dessin proposed a formula permitting the evaporation coefficient to be calculated for any vessel of a multiple effect:

$$c = 0.001 (100 - B) (T - 54) \quad \left(c = \frac{(100 - B) (T - 130)}{16,000} \right) \quad (32.31)$$

c = specific evaporation coefficient for the evaporator, in kg of vapour/m²/°C/h (lb./sq.ft./°F/h) (real temperature drop)

B = brix of the juice leaving the vessel (see remark below)

T = temperature of the heating steam in the calandria, in °C (°F).

Coutanceau, in Mauritius, has commented that it is more logical to take the average brix of the juice in the vessel rather than the brix of the juice leaving, and that results obtained in this way would be closer to the truth. We have adopted this point of view as correct, and it is from the mean brix that we shall calculate the evaporation coefficients.

This formula also takes into account the effect of scale, and so there is no need to use another coefficient to take such effects into account. On the other hand, it is appropriate to draw attention to the fact that the formula assumes a heat transmission appreciably better than that corresponding to the uncorrected coefficients of Table 32.16. If it is desired to retain the same margin of safety, it is necessary to replace the coefficient of 0.001 by 0.0008 or 0.0007 (or the divisor 16,000 by 20,000 or 23,000 in the formula in British units).

The value of 0.001 (divisor 16,000) corresponds to good normal conditions and to an apparatus remaining relatively clean.

A value of 0.0008 (20,000 for the divisor) is to be used in design work, in order to allow

a margin of safety and to take into account the eventual formation of rather heavy scale.

The value 0.0007 (divisor 23,000) corresponds to a multiple effect functioning in mediocre conditions or subject to rapid scaling.

Pre-evaporator. A formula is used in Puerto Rico⁶⁸ applicable to pre-evaporators:

$$\tau = 3.5 \Delta t \quad (T = 0.4 At) \quad (32.32)$$

T = evaporation coefficient of the pre-evaporator, in kg/m²/h (lb./sq.ft./h)

At = apparent temperature drop, in °C (°F), between the heating steam and vapour space.

This formula is very conservative, and certainly takes into account the average scale formed during the week. One could now generally use:

$$T = 4.5 At \quad (T = 0.5 At) \quad (32.33)$$

Evaporation coefficient and number of effects

Suppose we have a triple effect with 3 vessels of equal area, and that we change it into a quadruple by the addition of a fourth vessel identical with the preceding ones, without altering the initial steam pressure or the vacuum. Knowing the work of the vessels as a triple, what should this become when it forms a quadruple?

The overall temperature drop $T_0 - t_v$ between the exhaust steam and condenser, remaining the same, will distribute itself over 4 vessels instead of 3. The general formula (32.1) for heat transmission also shows that, if the mean heat-transfer coefficient of the vessels remains the same, which is generally the case very closely, each vessel will then give only three-quarters as much evaporation as it gave as a triple, since the temperature drop under which it works will have been reduced in the proportion of 4 : 3. If each vessel previously gave as a triple 4,000 kg of vapour per hour, it will not give more than 3,000 as a quadruple, but the total evaporation of the multiple effect will remain the same, since it is:

$$\begin{array}{ll} \text{As a triple:} & 3 \times 4,000 = 12,000 \text{ kg/h} \\ \text{As a quadruple:} & 4 \times 3,000 = 12,000 \text{ kg/h} \end{array}$$

In other words, the evaporation coefficient of a multiple effect, in kg of water/h/m² of heating surface, for a given total temperature drop, and for the same individual heating surface per vessel, is inversely proportional to the number of effects: the evaporation coefficient of a quadruple of 4 vessels each of 1,000 m² will be three-quarters of that of a triple of 3 vessels each of the same heating surface.

Consequently, the total evaporation of a multiple effect depends, not on its total heating surface, but on the individual heating surface of each vessel.

As an average and for a normal degree of scaling, we may estimate that *the total quantity of water, in kg (lb.), evaporated per hour by a multiple effect with vessels of equal heating surface is equal to the product of its individual heating surface per vessel, expressed in m² (sq.ft.), multiplied by 100 (20).*

If it is desired to relate the performance to the total heating surface of a multiple effect, estimates may be based on the evaporation coefficients given in Table 32.17.

TABLE 32.17

EVAPORATION COEFFICIENT PER UNIT TOTAL HEATING SURFACE		
	(kg/m ² /h)	(lb./sq.ft./h)
Single	100	20
Double effect	50	10
Triple effect	33	7
Quadruple effect	25	5
Quintuple effect	20	4

These coefficients represent good values for design and estimating purposes. In practice they are often exceeded, sometimes by more than 50%.

Vapour bleeding. If the evaporators are arranged for well-designed vapour bleeding from the earlier vessels, these will not be limited so closely by the following effects, and, since their heat-transfer coefficient is superior to the average, will be able to work at a higher rate and so improve the mean evaporation coefficient for the set. This is why a well-designed multiple effect, with heating surfaces decreasing and very unequal for the different vessels, and consequently with heavy vapour bleeding from the earlier vessels, can more easily attain very satisfactory average evaporation coefficients.

Variation of evaporation coefficient with temperature limits. For a very similar reason, the increase in evaporation is by no means the same for an increase in the overall temperature drop $T_0 - t_v$, under which the multiple effect is operating, according as this increase is made at the lower limit or at the upper limit.

Let us consider a normal temperature drop of 65°C (117°F) obtained with:

$$(A) \quad \left. \begin{array}{l} T_0 = 120^\circ\text{C} \\ t_v = 55^\circ\text{C} \end{array} \right\} \quad T_0 - t_v = 65^\circ\text{C} \text{ (117}^\circ\text{F)}$$

If we increase this temperature drop by 1° at the lower limit, say:

$$(B) \quad \left. \begin{array}{l} T_0 = 120^\circ\text{C} \\ t_v = 54^\circ\text{C} \end{array} \right\} \quad T_0 - t_v = 66^\circ\text{C} \text{ (119}^\circ\text{F)}$$

the increase will show its effects almost entirely on the temperatures in the last vessel, a little less on those of the preceding one, and hardly at all on the first.

Conversely, if we increase the temperature drop by 1° at the upper limit, so that we have:

$$(C) \quad \left. \begin{array}{l} T_0 = 121^\circ\text{C} \\ t_v = 55^\circ\text{C} \end{array} \right\} \quad T_0 - t_v = 66^\circ\text{C} \text{ (119}^\circ\text{F)}$$

it is the temperature of the 1st vessel that will increase by 1° or near it, and the last which will be scarcely affected.

Taking numerical values for example, let us assume the case of a quadruple, and that the temperatures reach equilibrium at the following figures for the 3 cases concerned:

	Temperatures ($^{\circ}\text{C}$)		
	A	B	C
Exhaust steam	120	120	121
Vapour from 1st vessel	111	110.9	111.9
Vapour from 2nd vessel	100	99.8	100.8
Vapour from 3rd vessel	85	84.6	85.6
Vapour from 4th vessel	55	54	55

Let us assume finally that the brix figures and the temperature drops in each vessel are as given in Table 32.18.

TABLE 32.18

	Brix of juice			Temperature drop, Δt ($^{\circ}\text{C}$)				
	Entering	Leaving	Mean	Apparent		B.P.E	True	
				A	B&C		A	B&C
1st vessel	12	18	15	9	9.1	0.8	8.2	8.3
2nd vessel	18	26	22	11	11.1	1.3	9.7	9.8
3rd vessel	26	32	29	15	15.2	2	13	13.2
4th vessel	32	64	48	30	30.6	6.3	23.7	24.3

The Dessin formula gives:

	1st vessel	2nd vessel	3rd vessel	4th vessel	Average ($\text{kg}/\text{m}^2/\text{h}$)
Case (A) Evaporation coefficient c	5.61	4.45	3.27	1.61	
Evaporation per m^2/h	46	43.2	42.5	38.2	42.7
Case (B) Evaporation coefficient c	5.61	4.44	3.25	1.59	
Evaporation per m^2/h	46.6	43.5	42.9	38.6	43.1
Case (C) Evaporation coefficient c	5.69	4.52	3.32	1.64	
Evaporation per m^2/h	47.2	44.3	43.8	39.9	44

We see that an increase of $1^{\circ} = 1.5\%$ at the high end of the temperature scale produces an increase in evaporation of:

$$\frac{44 - 42.7}{42.7} = 3\%$$

whereas an equal gain on the vacuum end produces a gain of only:

$$\frac{43.1 - 42.7}{42.7} = 0.9\%$$

or $\frac{1}{3}$ as much.

It is easily seen that this difference is due to the fact that, when the vacuum is increased, the pressure in the calandrias is decreased and consequently the heat-transfer coefficient in each vessel is decreased, and this partly offsets the gain due to the increased temperature drop; whereas when the pressure at the steam end is increased, the coefficients are increased and this effect is added to the gain in the overall temperature drop.

Choice of number of effects

From the point of view of steam economy, it is obviously of interest that the number of effects should be as high as possible. Why then is the number of effects generally limited to 3, 4 or 5?

This is due to four principal causes:

(1) The capacity of a multiple effect being determined by the individual surface of its vessels, its cost will be approximately proportional to the number of effects.

(2) In changing from a set of the order of $n - 1$ to a multiple set of the order of n , the saving in steam consumption effected has the value:

$$Q' - Q = \frac{E}{n - 1} - \frac{F}{n} = \frac{E}{n(n - 1)} \quad (32.34)$$

As a ratio of the previous steam consumption, this saving represents:

$$\frac{Q' - Q}{Q} = \frac{\frac{E}{n(n - 1)}}{\frac{E}{n - 1}} = \frac{1}{n} \quad \text{or:} \quad \frac{100}{n} \%$$

This therefore decreases rapidly with the number of effects.

(3) It is relatively simpler and less expensive to increase the number of effects by the installation of a thermo-compressor, which at the same time takes up less space.

(4) Losses by radiation and convection are approximately proportional to the number of vessels, that is, to the number of effects. Further, the loss in temperature drop due to boiling-point elevation increases with the number of effects. It is already almost equal in the last vessel of a multiple effect to that which would occur with single effect. The analogous loss due to hydrostatic pressure similarly increases with the number of effects. Finally, these three losses contribute to reduce rapidly the theoretical gain to be obtained by the addition of a further effect.

Table 32.19 gives an example of the losses in temperature drop which can be experienced with a modern evaporator set (b = due to brix; h = due to hydrostatic pressure; t — total).

TABLE 32.19

LOSS OF TEMPERATURE DROP IN MULTIPLE EFFECT ($b + h = t$)

	Triple effect	Quadruple effect	Quintuple effect
1st effect	0.2 + 0.8 = 1 °C (0.4 + 1.4 = 1.8°F)	0.2 + 0.6 = 0.8°C (0.4 + 1.1 = 1.5°F)	0.2 + 0.6 = 0.8°C (0.4 + 1.1 = 1.5°F)
2nd effect	0.5 + 1.3 = 1.8°C (0.9 + 2.3 = 3.2°F)	0.4 + 0.9 = 1.3°C (0.7 + 1.6 = 2.3°F)	0.3 + 0.8 = 1.1°C (0.5 + 1.4 = 1.9°F)
3rd effect	1.9 + 4.1 = 6 °C (3.4 + 7.4 = 10.8°F)	0.7 + 1.4 = 2.8°C (1.3 + 2.5 = 3.8°F)	0.6 + 1 = 1.6°C (1.1 + 2.8 = 2.9°F)
4th effect		1.9 + 4.2 = 6.1°C (3.4 + 7.6 = 11.0°F)	1.0 + 1.7 = 2.7°C (1.8 + 3.1 = 4.9°F)
5th effect			2.5 + 4.2 = 6.7°C (4.5 + 7.6 = 12.1°F)

This small table has been compiled on the assumptions: $T_0 = 115 - 120^\circ\text{C}$ ($240 - 250^\circ\text{F}$), $t_v = 55^\circ\text{C}$ (131°F), height of calandria = 1.8-2.1 m (6-7 ft.), with typical vapour bleeding and heat balance.

The total temperature drop is therefore reduced in the proportions shown in Table 32.20.

TABLE 32.20

LOSS IN TOTAL TEMPERATURE DROP

	Total drop		Loss		Net drop		Loss % of drop	
	(°C)	(°F)	(°C)	(°F)	(°C)	(°F)	Total	Previous net
Triple effect	63	114	8.8	15.8	54.2	98.2	13.9	-
Quadruple effect	63	114	10.3	18.6	52.7	95.4	16.3	2.8
Quintuple effect	63	114	12.9	23.2	50.1	90.8	20.4	4.9

In order to increase from a quintuple to a sextuple effect, it would thus be necessary to tolerate in the same way a further loss of more than 7% in the capacity of the equipment, although it is increased by one vessel. To this loss in capacity must be added the losses of heat by radiation and convection, which themselves increase with the number of effects: e.g. 2%, 5% and 9% for triple, quadruple and quintuple respectively. It is seen that the expense of a supplementary effect becomes very difficult to pay off.

In practice, a substantial gain is made by increasing from a triple to a quadruple and a further attractive gain from a quadruple to a quintuple, for large factories. Very approximately, we would install:

A triple in factories of less than 50 t.c.h.

A quadruple in factories of 50-200 t.c.h.

A quintuple in factories of more than 200 t.c.h.

Sextuple effects are quite exceptional and would not be definitely payable, even in a very large factory, under normal conditions.

Comments. (7) *Height of tubes.* We would comment that the loss suffered by the temperature drop is due mainly to the hydrostatic pressure, and increases with the length of the tubes. In other words, there is proportionately less advantage in increasing the number of effects if the tubes are longer.

(2) *Evaporation coefficient.* The consideration of the losses above leads to a modification of the theoretical Table 32.17. Taking this into account, and increasing by 20% to obtain analogous evaporation coefficients, corresponding to current values, we would have the figures given in Table 32.21.

These figures, however, represent only mean and conservative values to be adopted for preliminary design figures, but frequently exceeded in practice.

TABLE 32.21

EVAPORATION COEFFICIENT, IN $\text{kg}/\text{m}^2/\text{h}$ ($\text{lb.}/\text{sq. ft.}/\text{h}$)

	<i>Length of tubes</i>	
	<i>2.4 m (8 ft.)</i>	<i>1.5 m (5 ft.)</i>
Single effect	120 (24)	125 (25)
Double effect	58 (11.6)	60 (12)
Triple effect	37 (7.4)	39 (7.8)
Quadruple effect	27 (5.4)	28 (5.6)
Quintuple effect	20 (4)	22 (4.4)

TABLE 32.22

PRACTICAL EVAPORATION COEFFICIENTS (WEBRE)

	T_0	t	Evap. coeff.	T_0	t_v	Evap. coeff.
	(°C) *		($\text{kg}/\text{m}^2/\text{h}$)	(°F)		($\text{lb.}/\text{sq. ft.}/\text{h}$)
Ordinary triple effect	108	52	51	111	125	10.5
Ordinary quadruple effect	108	52	39	227	125	8
Ordinary quintuple effect	108	52	29	225	125	6
Pressure double effect	123	102	32	253	216	6.5
Pressure triple effect	123	102	22	253	216	4.5
Pauly or pre-evaporator			29 – 39			6–8

Webre⁶⁹ has indicated evaporation coefficients which he considered normal, and Table 32.22 shows figures he has given more recently⁷⁰.

These values are about 30% higher than those of Table 32.21.

Birkett^{70a}, in Louisiana, gives values very close to those of Webre:

Triple effect:	54 kg/m ² /°C/h	(11 lb./sq.ft./°F/h)
Quadruple effect:	39 kg/m ² /°C/h	(8 lb./sq.ft./°F/h)
Quintuple effect:	29 kg/m ² /°C/h	(6 lb./sq.ft./°F/h)

CALCULATIONS AND DESIGN OF A MULTIPLE EFFECT

Distribution of pressure drop

When a design is made of an entirely new multiple-effect installation, and when all the conditions of operation are therefore under control, the heating surfaces and the respective temperature drops for the vessels would be determined after a calculation of the maximum economy; this calculation will be discussed later.

However, the results of such a calculation are not always acceptable, and must sometimes *be* modified in order to allow for certain secondary considerations which necessitate a departure from the conditions calculated for maximum economy.

One of these considerations is the following: it is important to distribute the total pressure drop of a multiple effect between the vessels in such a way that the individual pressure drops under which the different vessels are working are approximately equal, but decrease slightly from the first to the last vessel.

The objects of this equalisation of pressure drops are to give uniform heights of siphon between vessels, to avoid withdrawals of juice, water or incondensable gases too great in certain vessels and too small in others, and above all to avoid risks of entrainment which would be caused by giving the earlier vessels too high a pressure drop. Such risks are reduced by equalising the pressure drops as much as possible. And it is because risks of entrainment increase from the first to the last vessel that a scale of pressure drops is adopted decreasing slightly rather than drops which are rigorously equal.

TABLE 32.23

DISTRIBUTION OF PRESSURE DROP BETWEEN VESSELS				
Triple effect	$\frac{11}{30}$	$+$	$\frac{10}{30}$	$+$ $\frac{9}{30}$
Quadruple effect	$\frac{11}{40}$	$+$	$\frac{10.3}{40}$	$+$ $\frac{9.7}{40}$ $+$ $\frac{9}{40}$
Quintuple effect	$\frac{11}{50}$	$+$	$\frac{10.5}{50}$	$+$ $\frac{10}{50}$ $+$ $\frac{9.5}{50}$ $+$ $\frac{9}{50}$

Values to be adopted. If the above principle could be followed entirely, the total pressure drop could be distributed as shown in Table 32.23. In the case where the overall temperature drop is 120° - 55° = 65°C (117°F), this would give the absolute pressures and the corresponding temperatures shown in Table 32.24.

Vapour available at each effect

This scale of pressures leads us to a very interesting conception, that of the weight of vapour

available in the various vessels of a given multiple effect. This idea is very useful for giving a rapid immediate idea of the possibilities of vapour bleeding from an existing set.

The optimum pressure drops being fixed according to the foregoing principle, the intervals of temperature are deduced from these. The temperature drops under which the different vessels operate do not vary greatly when the upper and lower limits of temperature T_0 and t_v alter, provided of course that it is a case of a standard multiple effect, working between a back-pressure of $0.5 - 1.5 \text{ kg/cm}^2$ ($7 - 20 \text{ p.s.i.}$) and a vacuum of 61 cm (24 in.) of mercury.

Assuming a normal brix for each vessel, and basing our calculations on the exact temperatures corresponding to the above pressures, the application of the Dessin formula (coefficient 0.0008) gives the figures for evaporation capacity shown in Table 32.25.

TABLE 32.24

PRESSURE AND TEMPERATURE DROPS, IN $^{\circ}\text{C}$ AND kg/cm^2 ($^{\circ}\text{F}$, p.s.i.a.)
($T_0 = 120^{\circ}\text{C}$ (248°F); $t_v = 55^{\circ}\text{C}$ (131°F); pressure distribution as Table 32.23)

		Steam	1st effect	2nd effect	3rd effect	4th effect	5th effect
Triple effect	Pressure	2.0 (28.8)	1.3 (19)	0.7 (10.2)	0.16 (2.3)		
	Temperature	120° (248°)	107° (225°)	90° (194°)	55° (131°)		
Quadruple effect	Pressure	2.0 (28.8)	1.5 (21.5)	1.0 (14.5)	0.58 (8.2)	0.16 (2.3)	
	Temperature	120° (248°)	111° (232°)	100° (212°)	85° (185°)	55° (131°)	
Quintuple effect	Pressure	2.0 (28.8)	1.6 (23)	1.2 (17.4)	0.85 (12.1)	0.5 (7)	0.16 (2.3)
	Temperature	120° (248°)	113° (235°)	105° (221°)	95° (203°)	80° (175°)	55° (131°)

TABLE 32.25

EVAPORATION CAPACITY OF THE SEVERAL VESSELS OF A MULTIPLE EFFECT WORKING UNDER THE CONDITIONS OF TABLE 32.24, IN $\text{kg/m}^2/\text{h}$ (lb./sq.ft./h) (decreasing pressure drops)

	1st vessel	2nd vessel	3rd vessel	4th vessel	5th vessel
Triple effect	53 (11)	48 (9.8)	43 (8.8)		
Quadruple effect	37 (7.6)	35 (7.2)	32 (6.6)	28 (5.7)	
Quintuple effect	28 (5.7)	26 (5.3)	25 (5.1)	25 (5.1)	17(3.5)

If a multiple effect were available working between the temperature limits of 112° and 55°C (234° and 131°F) (instead of 120°C and 55°C), with a slightly lower brix, it would be possible to obtain slightly different values. These are shown as follows, calculated from a coefficient of 0.001 so as to render them more comparable with the preceding figures; units are kg/mVh (in brackets, lb./sq.ft./h):

Triple:	50 (10)	47 (9.5)	38 (7.8)		
Quadruple:	35 (7.2)	33 (6.8)	29 (6)	23 (4.7)	
Quintuple:	26 (5.3)	25 (5.1)	23 (4.7)	21 (4.3)	16 (3.3)

Frequently, with quadruple and quintuple effects, the possibilities of the second last vessel

are not utilised, as it is considered that it is not worth the trouble. In this case, the two latter vessels will adjust themselves automatically, and Table 32.25, for example, becomes (units as above):

Quadruple:	37 (7.6)	35 (7.2)	30 (6.2)	30 (6.2)	
Quintuple:	28 (5.7)	26 (5.3)	25 (5.1)	21 (4.3)	21 (4.3)

Thus the vapour available from the antepenultimate vessel is considerably increased.

Example. Two quadruple effects each with four equal vessels, one of 500 m², the other of 5,000 sq.ft. each, working between 120 and 55°C (248 and 131°F), would be able to furnish the quantities for bleeding as shown in Table 32.26.

TABLE 32.26

	DISTRIBUTION OF VAPOUR IN A QUADRUPLE					
	Vessels of 500 m ²			Vessels of 5,000 sq.ft.		
	Evap. rate (kg/m ² /h)	Total evap. (kg/h)	Available (kg/h)	Evap. rate (lb./sq.ft./h)	Total evap. (lb./h)	Available (lb./h)
1st effect	37	18,500	1,000	7.6	38,000	2,000
2nd effect	35	17,500	1,500	7.2	36,000	3,000
3rd effect	32	16,000	2,000	6.6	33,000	4,500
4th effect	28	14,000	14,000	5.7	28,500	28,500

Comment. *Evaporation coefficients and number of effects.* We have already passed from the theoretical Table 32.17 to Table 32.21 by the consideration of losses. If we accept the Dessin formula (which is approximate, but gives an acceptable approximation), the figures of Table 32.25, for example, show that the advantage of the triple effect over the quadruple, and of the quadruple over the quintuple, is still more marked than Table 32.21 indicated. Actually, addition of the capacities of Table 32.25 (for example, adjusted to a coefficient of 0.000812 (denominator of 19,700), and related to the total surface of the multiple effect (assumed of equal vessels)) gives:

$$\begin{aligned}
 \text{Triple effect:} \quad & \frac{0.000812}{0.001} \times \frac{53 + 48 + 43}{3} = \\
 & = 0.812 \times \frac{144}{3} = 39 \text{ kg/mVh (8 lb./sq.ft./h)} \\
 \text{Quadruple' effect:} \quad & \frac{0.000812}{0.001} \times \frac{37 + 35 + 32 + 28}{4} = \\
 & = 0.812 \times \frac{132}{4} = 26.8 \text{ kg/mVh (5.5 lb./sq.ft./h)}
 \end{aligned}$$

$$\begin{aligned} \text{Quintuple effect: } & \frac{0.000812}{0.001} \times \frac{28 + 26 + 25 + 25 + 17}{5} = \\ & = 0.812 \times \frac{121}{5} = 19.7 \text{ kg/m}^2/\text{h} \text{ (4 lb./sq.ft./h)} \end{aligned}$$

This limits the number of effects even sooner than the remarks made after Table 32.20 would lead one to suppose.

Determination of heating surface of the vessels

It is possible to obtain from each vessel of a multiple effect practically any rate of evaporation, provided it is given the necessary heating surface, corresponding to the pressure of the vapour which heats it, to the brix of the juice in it, to the temperature drop under which it is operating, etc.

But it is apparent that the heating surfaces of the different vessels should be proportioned in a logical manner, and the most logical is to have a total heating surface for the multiple effect which is a minimum.

Proportioning to obtain a minimum total surface

We shall seek then to find how the vessels should be proportioned in order to arrive at a minimum total heating surface.

We shall assume the general case of a multiple effect of n vessels, with vapour bleeding.

Let:

$S_1, S_2, S_3, \dots, S_n$ be the respective heating surfaces of the n vessels

$\Delta_1, \Delta_2, \Delta_3, \dots, \Delta_n$ be the net temperature drops in the n vessels

$d_1, d_2, d_3, \dots, d_n$ be the boiling-point elevations

$\Delta'_1, \Delta'_2, \Delta'_3, \dots, \Delta'_n$ be the corresponding apparent temperature drops: $\Delta' = A + d$

$c_1, c_2, c_3, \dots, c_n$ be the real specific evaporation coefficients for the n vessels

$q_1, q_2, q_3, \dots, q_n$ be the quantities of water evaporated by the n vessels

$b_1, b_2, b_3, \dots, b_n$ be the mean brix values of the juice in these vessels.

These quantities could be expressed in either metric or British units, since the only quantity affected by the system of units is the numerical basic temperature in the Dessin formula, and even this is eliminated from the calculation; hence the calculations are entirely independent of the system of units employed.

The quantities of water evaporated by the different vessels are respectively:

$$q_1 = c_1 S_1 \Delta_1 \quad q_2 = c_2 S_2 \Delta_2 \quad \dots \quad q_n = c_n S_n \Delta_n$$

and:

$$q_1 + q_2 + q_3 + \dots + q_n = Q$$

Q , being the total quantity of water to be evaporated, is part of the essential data for the problem; and:

$$\Delta'_1 + \Delta'_2 + \Delta'_3 + \dots + A; = \Delta'$$

$$\Delta_1 + \Delta_2 + \Delta_3 + \dots + \Delta_n = A$$

A' being the total apparent temperature drop and A the net drop between exhaust steam and condenser. Once the length of the tubes is chosen, A may be considered as part of the fundamental data as well as Δ', since (1) the brix values depend on the respective q values, which are assumed given, and (2) the various d values depend entirely on the brix values and the lengths of the tubes.

We shall first study the proportioning of the first vessel in relation to the remainder, assuming tentatively that the other vessels are already properly proportioned between themselves, and that the condition at which we shall arrive will be such that these proportions do not vary when the surface of the first vessel varies. We shall return later to this hypothesis.

To simplify the calculations, we shall assume that the boiling-point elevations d are proportional to the temperature drops A, which is approximately true, particularly if the last effect is not considered, the figure for which (d_n) is not required for our illustration. We may thus assume:

$$\Delta_1 = m \Delta \quad \Delta'_1 = m \Delta'$$

then:

$$\begin{array}{ll} \Delta_2 = k_2 (1 - m) A & \Delta'_2 = k_2 (1 - m) \Delta' \\ A) = k_3 (1 - m) A & \Delta'_3 = k_3 (1 - m) \Delta' \\ \dots\dots\dots & \dots\dots\dots \\ \Delta_n = k_n (1 - m) A & A; = k_n (1 - m) \Delta' \end{array}$$

k₂, k₃, k₄, ..., k_n being the coefficients of proportionality for the temperature drops of the other vessels among themselves, the relative values of which do not vary when m varies.

Finally, we shall postulate that the basic temperature of the Dessin formula (54°C; 130°F) may be substituted for the temperature corresponding to the vacuum (which is always only slightly different from the truth, and which will enable us to avoid using a figure for the difference e between these two temperatures).

The surface of the 1st vessel has a value:

$$S_1 = \frac{q_1}{c_1 \Delta_1} = \frac{q_1}{c_1 m A}$$

and from the Dessin formula:

$$c_1 = 0.001 (100 - b_1) (T_0 - 54) = a_1 \Delta'$$

By substituting:

$$0.001 (100 - b_1) = a_1$$

then:

$$S_1 = \frac{q_1}{a_1 m \Delta \Delta'}$$

Similarly:

$$S_2 = \frac{q_2}{c_2 \Delta_2}$$

and:

$$c_2 = 0.001 (100 - b_2)(\Delta' - \Delta'_1) = a_2 (\Delta' - \Delta'_1) = a_2 \Delta' (1 - m)$$

or:

$$S_2 = \frac{q_2}{a_2 k_2 (1 - m)^2 \Delta \Delta'}$$

In the same way:

$$S_3 = \frac{q_3}{c_3 \Delta_3}$$

and:

$$c_3 = a_3 (\Delta' - \Delta'_1 - \Delta_2) = a_3 [\Delta' - m \Delta' - k_2 (1 - m) \Delta'] = a_3 \Delta' (1 - k_2) (1 - m)$$

or:

$$S_3 = \frac{q_3}{a_3 k_3 (1 - k_2) (1 - m)^2 \Delta \Delta'}$$

In general:

$$S_n = \frac{q_n}{a_n k_n (1 - k_2 - k_3 - \dots - k_{n-1}) (1 - m)^2 \Delta \Delta'}$$

Finally, the total heating surface S of the multiple effect will have the value:

$$S = S_1 + S_2 + S_3 + \dots + S_n$$

If we seek its minimum value, when m varies, we should write:

$$\frac{dS}{dm} = \frac{dS_1}{dm} + \frac{dS_2}{dm} + \frac{dS_3}{dm} + \dots + \frac{dS_n}{dm} = 0$$

or:

$$-\frac{q_1}{a_1 \Delta \Delta' m^2} + \frac{2q_2}{a_2 k_2 \Delta \Delta' (1 - m)^3} + \frac{2q_3}{a_3 k_3 (1 - k_2) \Delta \Delta' (1 - m)^3} + \dots + \frac{2q_n}{a k (1 - f a - f a - \dots - k_{n-1},) \Delta \Delta' (1 - m)^3} = 0$$

or:

$$\frac{S_1}{m} = \frac{2S_2}{1 - m} + \frac{2S_3}{1 - m} + \dots + \frac{2S_n}{1 - m}$$

$$\frac{2m}{1 - m} = \frac{S_1}{S_2 + S_3 + \dots + S_n}$$

Thus, with net drops:

$$\frac{S_1}{S_2 + S_3 + \dots + S_n} = \frac{2A_1}{\Delta_2 + \Delta_3 + \dots + \Delta_n} \tag{32.35}$$

Or, with apparent drops:

$$\frac{S_1}{S_2 + S_3 + \dots + S_n} = \frac{2A_1}{\Delta_2' + \Delta_3' + \dots + \Delta_n'} \tag{32.36}$$

In other words: *To obtain a minimal heating surface for the multiple effect, the ratio of the heating surface of a vessel to the sum of the heating surfaces of the following vessels is twice the ratio of the temperature drop for that vessel to the sum of the temperature drops of the following vessels.*

If we substitute:

$$\frac{q_1}{c_1} = t_1 \quad \frac{q_2}{c_2} = t_2 \quad \dots \quad \frac{q_n}{c_n} = t_n$$

we have:

$$S_1 = \frac{t_1}{\Delta_1} \quad S_2 = \frac{t_2}{\Delta_2} \quad \dots \quad S_n = \frac{t_n}{\Delta_n} \tag{32.37}$$

and these equations show that, in eqn. (32.35), we may substitute t/Δ for S in each case.

Having thus fixed the optimum values of S_1 and Δ_1 , it remains to find the optimum value of the heating surfaces and the temperature drops, for the other vessels. But it is easy to see

that once the 1st vessel is fixed, the 2nd vessel becomes, in relation to the following ones, virtually a 1st vessel, in the sense that its coefficient c becomes fixed in its turn. It is just as if we were dealing with a multiple effect of the order $n - 1$ and a total temperature drop of $A - \Delta_1$. The calculation would follow an identical pattern, and we should find in the same way:

$$\frac{S_2}{S_3 + S_4 + \dots + S_n} = \frac{2\Delta_2}{\Delta_3 + \Delta_4 + \dots + A_n} \quad (32.38)$$

and so in turn, for the double effect formed by the last two vessels:

$$\frac{S_{n-1}}{S_n} = \frac{2A_{n-1}}{\Delta_n} \quad (32.39)$$

Case of a quadruple. In the case of a quadruple, for example, we have:

$$\frac{S_1}{S_2 + S_3 + S_4} = \frac{2\Delta_1}{\Delta_2 + \Delta_3 + \Delta_4} \quad \text{or} \quad \frac{\frac{t_1}{\Delta_1}}{\frac{t_2}{\Delta_2} + \frac{t_3}{\Delta_3} + \frac{t_4}{\Delta_4}} = \frac{2\Delta_1}{\Delta_2 + \Delta_3 + \Delta_4} \quad (32.40)$$

$$\frac{S_2}{S_3 + S_4} = \frac{2\Delta_2}{\Delta_3 + \Delta_4} \quad \text{or} \quad \frac{\frac{t_2}{\Delta_2}}{\frac{t_3}{\Delta_3} + \frac{t_4}{\Delta_4}} = \frac{2\Delta_2}{\Delta_3 + \Delta_4} \quad (32.41)$$

$$\frac{S_3}{S_4} = \frac{2\Delta_3}{\Delta_4} \quad \text{or} \quad \frac{\frac{t_3}{\Delta_3}}{\frac{t_4}{\Delta_4}} = \frac{2\Delta_3}{\Delta_4} \quad (32.42)$$

$$\Delta_1 + \Delta_2 + \Delta_3 + \Delta_4 = A \quad (32.43)$$

Now let:

$$r_1 = \frac{A}{\Delta_1} \quad r_2 = \frac{\Delta_1}{\Delta_2} \quad r_3 = \frac{\Delta_2}{\Delta_3} \quad r_4 = \frac{\Delta_3}{\Delta_4}$$

From (32.42) we have:

$$r_4 = \frac{\Delta_3}{\Delta_4} = \sqrt{t_3} \quad (32.44)$$

that once the 1st vessel is fixed, the 2nd vessel becomes, in relation to the following ones, virtually a 1st vessel, in the sense that its coefficient c becomes fixed in its turn. It is just as if we were dealing with a multiple effect of the order $n - 1$ and a total temperature drop of $A - \Delta_1$. The calculation would follow an identical pattern, and we should find in the same way:

$$\frac{S_2}{S_3 + S_4 + \dots + S_n} = \frac{2\Delta_2}{As + \Delta_4 + \dots + \Delta_n} \quad (32.38)$$

and so in turn, for the double effect formed by the last two vessels:

$$\frac{S_{n-1}}{S_n} = \frac{2A}{\Delta_n} \quad (32.39)$$

Case of a quadruple. In the case of a quadruple, for example, we have:

$$\frac{S_1}{S_2 + S_3 + S_4} = \frac{2\Delta_1}{\Delta_2 + \Delta_3 + \Delta_4} \quad \text{or} \quad \frac{\frac{t_1}{\Delta_1}}{\frac{t_2}{\Delta_2} + \frac{t_3}{As} + \frac{t_4}{\Delta_4}} = \frac{2A}{\Delta_2 + As + \Delta_4} \quad (32.40)$$

$$\frac{S_2}{S_3 + S_4} = \frac{2\Delta_2}{\Delta_3 + \Delta_4} \quad \text{or} \quad \frac{\frac{t_2}{\Delta_2}}{\frac{t_3}{\Delta_3} + \frac{t_4}{\Delta_4}} = \frac{2\Delta_2}{\Delta_3 + \Delta_4} \quad (32.41)$$

$$\frac{S_3}{S_4} = \frac{2\Delta_3}{\Delta_4} \quad \text{or} \quad \frac{\frac{t_3}{\Delta_3}}{\frac{t_4}{\Delta_4}} = \frac{2\Delta_3}{\Delta_4} \quad (32.42)$$

$$\Delta_1 + \Delta_2 + As + \Delta_4 = A \quad (32.43)$$

Now let:

$$r_1 = \frac{A}{\Delta_1} \quad r_2 = \frac{\Delta_1}{\Delta_2} \quad r_3 = \frac{\Delta_2}{As} \quad r_4 = \frac{As}{\Delta_4}$$

From (32.42) we have:

$$r_4 = \frac{\Delta_3}{\Delta_4} = \sqrt{\frac{t_3}{2t_4}} \quad (32.44)$$

Substituting for Δ_4 in (32.41):

$$r_3 = \frac{\Delta_2}{\Delta_3} = \frac{\sqrt{\left(1 + \frac{1}{r_3}\right) t_2}}{\sqrt{2(t_2 + t_3 r_3)}} \quad (32.45)$$

Substituting in (32.40):

$$r_2 = \frac{\Delta_1}{\Delta_2} = \sqrt{\frac{\left(1 + \frac{1}{r_3} + \frac{1}{r_3 r_4}\right) t_1}{2(t_2 + t_3 r_3 + t_4 r_4 r_3)}} \quad (32.46)$$

Finally, (32.43) gives:

$$r_1 = \frac{A}{\Delta_1} = 1 + \frac{1}{r_2} + \frac{1}{r_2 r_3} + \frac{1}{r_2 r_3 r_4} \quad (32.47)$$

These last four equations permit the four A values to be calculated by a reduction formula. The application of this is very rapid: we shall give an example of it later (p. 591).

Case of a triple. The value of the corresponding expressions is obtained very readily, by a change of the indices, in the case of a triple or a quintuple effect. For the triple, for example, we have:

$$r_1 = \frac{A}{\Delta_1} = 1 + \frac{1}{r_2} + \frac{1}{r_2 r_3} \quad (32.48)$$

$$r_2 = \frac{\lambda \Delta_1}{\Delta_2} = \sqrt{\frac{\left(1 + \frac{1}{r_3}\right) t_1}{2(t_2 + t_3 r_3)}} \quad (32.49)$$

$$r_3 = \frac{\Delta_2}{\Delta_3} = \sqrt{\frac{t_2}{2t_3}} \quad (32.50)$$

Comment. We may now establish that the hypothesis on which our reasoning was based (cf. p. 583) was correct: the condition of optimum proportion between the vessels following the 1st, at which we have arrived, is such that their temperature drops remain proportional between themselves when the temperature drop of the 1st varies. **Actually**, if our reasoning had been made for the case of a quintuple, for example, and if, in conditions such as (32.40) to (32.43), assumed to apply to the last 4 vessels of this quintuple, the temperature drop A became λA , all the conditions would remain unchanged when Δ_1 was replaced by $\lambda \Delta_1$, Δ_2 by $\lambda \Delta_2$, etc., since the coefficients λ could then be placed as a factor in the numerator and

denominator in the second term of expressions (32.40) to (32.42) and in both sides of eqn. (32.43).

MULTIPLE-EFFECT CALCULATIONS

There are several methods of calculating for the design of a multiple effect. We shall study the simplest case, and then give the more important variations of this.

To give definite values and to simplify the exposition we shall treat a concrete case.

Calculations are given in the metric system, with the main results given in brackets in the British system. Full calculations for the same example are given in British units in the second edition, pp. 578 ff.

(A) Normal scale of pressures

Data. We wish to calculate a complete quadruple effect for a sugar factory, operating under the following conditions:

Crushing rate:	100 t.c.h.
Weight of mixed juice % cane:	100
Brix of clarified juice:	13
Brix required for syrup:	65
Temperature of juice before heating:	30°C (86°F)
Back-pressure:	1 kg/cm ² (14 p.s.i.)
Vacuum at condenser:	64 cm (25 in.)

Procedure to be followed. (a) *Calculation of total evaporation.* The total quantity of water to be evaporated is (eqn. (32.9)):

$$E = 100 \times 1,000 \left(1 - \frac{13}{65}\right) = 80,000 \text{ kg/h (179,200 ib./h)}$$

(b) *Choice of temperatures.* We adopt for this multiple effect the pressure scale of Table 32.23. At the same time we choose a figure for the length of tubes of the various vessels, which permits us to determine approximately the boiling-point elevations d in the vessels (pp. 499—502), hence the temperatures given in Table 32.27.

TABLE 32.27

	SEQUENCE OF TEMPERATURES, IN °C (°F)			
	Vapour	d	Juice	True temp. drop
Exhaust steam	120 (248)			
1st vessel	111 (232)	0.8 (1.4)	111.8 (233.4)	8.2 (14.6)
2nd vessel	100 (212)	1.3 (2.3)	101.3 (214.3)	9.7 (17.7)
3rd vessel	85 (185)	2.1 (3.8)	87.1 (188.8)	12.9 (23.2)
4th vessel	55 (131)	6.1 (11.0)	61.1 (142)	23.9 (43)
				54.7 (98.5)

(c) *Choice of vapour bleeding.* By means of these temperatures, we now choose the procedure for juice heating. In order to select economic conditions, we shall observe the margins of temperature recommended in Table 30.1 (p. 456).

We shall carry out the heating thus:

- By 3rd vessel: from 30 to 65°C (85 to 150°F)
- By 2nd vessel: from 65 to 85°C (150 to 185.°F)
- By 1st vessel: from 85 to 103°C (185 to 217°F)

We shall take further from the first vessel the vapour necessary for $\frac{2}{3}$ of the vacuum pans, the consumption of which is estimated to total 120 kg/t.c. (270 lb./t.c.).

The latent heats of the vapour are (Table 41.1):

- 1st vessel, 111°C (232°F): 532 kcal/kg (957 B.T.U./lb.)
- 2nd vessel, 100°C (212°F): 539 kcal/kg (970 B.T.U./lb.)
- 3rd vessel, 85°C (185°F): 548 kcal/kg (987 B.T.U./lb.)

Thus we arrive at the following figures for the bled vapour:

From 3rd vessel:

$$P_3 \approx \frac{1,000 \times 100 \times 0.9 (65 - 30)}{548 \times 0.95} = 6,050 \text{ kg/h (13,760 lb./h) vapour}$$

From 2nd vessel:

$$P_2 = \frac{1,000 \times 100 \times 0.9 (85 - 65)}{537 \times 0.95} = 3,515 \text{ kg/h (7,657 lb./h) vapour}$$

From 1st vessel:

(a) Heaters:	$P_1 = \frac{1,000 \times 100 \times 0.9 (103 - 85)}{532 \times 0.95} = 3,204$	}	11,204 kg/h (25,095 lb./h) vapour
(b) Pans:	$P_1 = \frac{100 \times 120 \times 2}{3} = 8,000$		

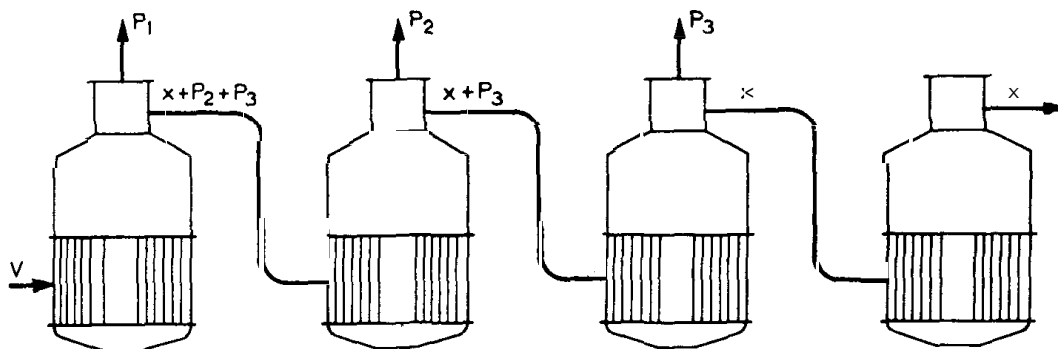


Fig. 32.47. Calculations for vapour bleeding.

(d) *Calculation of the individual evaporation of the vessels.* Let x be the quantity of vapour supplied by the 4th vessel and passing to the condenser. Since each vessel must supply the vapour necessary for the following one, plus the vapour for juice heating (Fig. 32.47), the several vessels will have to evaporate:

$$\begin{array}{rcl}
 \text{4th vessel:} & x & \\
 \text{3rd vessel:} & x + 6,050 & \\
 \text{2nd vessel:} & x + 6,050 + 3,515 & \\
 \text{1st vessel:} & x + 6,050 + 3,515 + 11,204 & \\
 \hline
 & 4x + 3 \times 6,050 + 2 \times 3,515 + 11,204 &
 \end{array}$$

Equating these totals to the total evaporation required, we have:

$$4x + 18,150 + 7,030 + 11,204 = 80,000$$

hence:

$$x = 10,904 \text{ kg/h (24,378 lb./h)}$$

The evaporations in the several vessels will therefore be:

$$\begin{array}{rcl}
 \text{4th vessel: } q_4 & = 10,904 & = 10,904 \\
 \text{3rd vessel: } q_3 & = 10,904 + 6,050 & = 16,954 \\
 \text{2nd vessel: } q_2 & = 16,954 + 3,515 & = 20,469 \\
 \text{1st vessel: } q_1 & = 20,469 + 11,204 & = 31,673
 \end{array}$$

$$\text{Total evaporation } E: \quad \underline{\quad\quad\quad} 80,000 \text{ kg/h}$$

(e) *Calculation of the brix figures.* We may now calculate the brix of the juice in each vessel as shown in Table 32.28.

TABLE 32.28

CALCULATION OF BRUX VALUES

		Average brix
Entering 1st vessel: $B_0 =$	13	} 16 = b_1
Leaving 1st vessel: $B_1 = 13 \times \frac{100,000}{100,000 - 31,673} =$	19	
Leaving 2nd vessel: $B_2 = 13 \times \frac{100,000}{68,327 - 20,469} =$	27.1	} 23.1 = b_2
Leaving 3rd vessel: $B_3 = 13 \times \frac{100,000}{47,858 - 16,954} =$	42.1	
Leaving 4th vessel: $B_4 = 13 \times \frac{100,000}{30,904 - 10,904} =$	65	} 34.6 = b_3

(f) *Calculation of the specific evaporation coefficients.* The Dessin formula (32.31) applied with the coefficient of 0.001 (or denominator 16,000) gives the coefficients of Table 32.29:

TABLE 32.29

CALCULATION OF SPECIFIC EVAPORATION RATES, IN kg/m²/°C/h (lb./sq.ft./°F/h)

1st vessel: $c_1 = 0.001 (100 - 16) (120 - 54) = 5.54 (0.620)$
2nd vessel: $c_2 = 0.001 (100 - 23.1) (111 - 54) = 4.38 (0.490)$
3rd vessel: $c_3 = 0.001 (100 - 34.6) (100 - 54) = 3.01 (0.335)$
4th vessel: $c_4 = 0.001 (100 - 53.5) (85 - 54) = 1.44 (0.160)$

(g) *Calculation of heating surfaces.* For each vessel we have:

$$S_p = \frac{q_p}{c_p \Delta_p}$$

Hence:

$S_1 = \frac{31,673}{5.54 \times 8.2} = 697 \text{ m}^2$	Say: 700 m ² (7,850 sq.ft.)
$S_2 = \frac{20,469}{4.38 \times 9.7} = 482 \text{ m}^2$	Say: 500 m ² (5,300 sq.ft.)
$S_3 = \frac{16,954}{3.01 \times 12.9} = 437 \text{ m}^2$	Say: 450 m ² (5,000 sq.ft.)
$S_4 = \frac{10,904}{1.44 \times 23.9} = 317 \text{ m}^2$	Say: 350 m ² (3,600 sq.ft.)
Total	1,933 m² Say: 2,000 m² (21,750 sq.ft.)

When the multiple effect is intended for a factory for which the properties of the juice are known, and where the existing evaporators do not suffer bad scaling, the value 0.001 may be used, as above, in the Dessin formula. Otherwise it is wise to replace it by 0.0009 or 0.0008 (or adopt a denominator of 18,000 or 20,000 in place of 16,000, for British units).

Whenever possible, the value should be one obtained from the performance of the existing multiple effect. A higher coefficient (lower denominator) for the Dessin formula should not be taken unless the forecasts of the performance or technical improvements in the new equipment justify it.

(B) Minimum total heating surface

The scale of pressures given in Table 32.23 is not an essential condition for the operation

of a multiple effect; instead of starting from this condition, we may set out to realise the conditions already found for a minimum total heating surface, in which case the scale of pressures obtained will be determined by these initial conditions. The calculation is somewhat lengthy. Theoretically it is made by successive approximations, starting precisely from the normal scale of pressures. Actually, it almost always suffices to recalculate once the temperatures so utilised.

The procedure to be followed remains the same. Calculations (a), (b), (c), (d), (e), (f) are carried out in turn. Keeping the same data as for the calculation (A), we shall proceed to continue the calculation of our quadruple effect with the object of obtaining the minimum total heating surface.

(h) *Calculation of the evaporation capacities t.* We shall call $t = q/c$ the evaporation "capacity" of a vessel. Actually, what is of value in a multiple effect is the heating surface and the temperature drop. Now, the quantity t is a measure of the expenditure in heating surface and temperature drop which must be made for an evaporator vessel, according to its place in the set:

$$t = \frac{q}{c} = S \cdot \Delta$$

We have:

$$t_1 = \frac{q_1}{c_1} = \frac{31,673}{5.54} = 5,717 \text{ (114,340)}$$

$$t_2 = \frac{q_2}{c_2} = \frac{20,469}{4.38} = 4,673 \text{ (93,460)}$$

$$t_3 = \frac{q_3}{c_3} = \frac{16,954}{3.01} = 5,632 \text{ (113,845)}$$

$$t_4 = \frac{q_4}{c_4} = \frac{10,904}{1.44} = 7,572 \text{ (152,362)}$$

(i) *Calculation of the recurrence quotients r.* This gives:

$$r_4 = \frac{\Delta_3}{\Delta_4} = \sqrt{\frac{t_3}{2t_4}} = \sqrt{\frac{5,632}{2 \times 7,572}} = 0.6098$$

$$r_3 = \frac{\Delta_2}{\Delta_3} = \frac{\sqrt{\left(1 + \frac{1}{r_4}\right) t_2}}{2(t_3 + t_4 r_4)} = \frac{\sqrt{(1 + 1.6398) \times 4,673}}{2(5,632 + 4,617)} = 0.7758$$

$$r_2 = \frac{\Delta_1}{\Delta_2} = \frac{\sqrt{\left(1 + \frac{1}{r_3} + \frac{1}{r_3 r_4}\right) t_1}}{2(t_2 + t_3 r_3 + t_4 r_4 r_3)} = \frac{\sqrt{(1 + 1.2891 + 2.1138) \times 5,717}}{2(4,673 + 4,369 + 3,582)} = 0.9985$$

$$r_1 = \frac{\Delta}{\Delta_1} = 1 + \frac{1}{r_2} + \frac{1}{r_2 r_3} + \frac{1}{r_2 r_3 r_4} = 1 + 1.0015 + 1.2910 + 2.1171 = 5.4096$$

(f) Calculation of the new theoretical temperature drops A. We have:

$$\Delta_1 = \frac{A}{r_1} = \frac{54.7}{5.4096} = 10.1^\circ\text{C} (18.1^\circ\text{F})$$

$$\Delta_2 = \frac{\Delta_1}{r_2} = \frac{10.112}{0.9985} = 10.1^\circ\text{C} (18.2^\circ\text{F})$$

$$\Delta_3 = \frac{\Delta_2}{r_3} = \frac{10.127}{0.7758} = 13.1^\circ\text{C} (23.6^\circ\text{F})$$

$$\Delta_4 = \frac{\Delta_3}{r_4} = \frac{13.054}{0.6098} = 21.4^\circ\text{C} (38.6^\circ\text{F})$$

$$A = 54.7^\circ\text{C} (98.5^\circ\text{F})$$

(k) Correction of the temperature drops found. The hypothesis of p. 583, which aimed to allow us to substitute the net temperature drops by the apparent drops, in the calculation of minimal heating surface, is only approximate. The boiling-point elevations *d* are not exactly proportional to the net temperature drops. They increase in fact more rapidly than the temperature drops from the 1st to the last vessel. It follows that the real specific evaporation coefficients which we found for the 2nd, 3rd and especially for the 4th vessels, in utilising these drops, and the calculated *d* values of Table 32.27, would be too high, since the temperature *T* of vapour entering into the Dessin formula (*T* — 54) would be somewhat higher than our calculation had indicated.

On the other hand, if we were to correct fully the net temperature drops found above for the difference between the proportional *d* and the true *d*, and if we incorporated these corrections in the temperature drop so as to give the exact values for *T*, this would give net temperature drops which would no longer correspond to the proportions required by the calculation: they would be too high in the 1st vessel and too low for the last.

A detailed calculation shows that the correction to be applied consists of adding the following amounts, as a very close approximation in most cases, to the net temperature drops obtained in (f):

For a Quadruple For a Quintuple

For the 1st vessel:	+0.8%	+0.9% of the total net drop
For the 2nd vessel	+0.6%	+0.7%
For the 3rd vessel:	+0.3%	+0.4%
For the 4th vessel:	-1.7%	-0.3%
For the 5th vessel:		-1.7%

which gives us the following correct temperature drops:

$$\begin{aligned}\Delta_1 &= 10.1 + (54.7 \times 0.008) = 10.5^\circ\text{C} (18.9^\circ\text{F}) \\ \Delta_2 &= 10.1 + (54.7 \times 0.006) = 10.4^\circ\text{C} (18.8^\circ\text{F}) \\ \Delta_3 &= 13.1 + (54.7 \times 0.003) = 13.3^\circ\text{C} (23.9^\circ\text{F}) \\ \Delta_4 &= 21.4 - (54.7 \times 0.017) = 20.5^\circ\text{C} (36.9^\circ\text{F}) \\ &\qquad\qquad\qquad \underline{\qquad\qquad\qquad} \\ &\qquad\qquad\qquad 54.7^\circ\text{C} (98.5^\circ\text{F})\end{aligned}$$

TABLE 32.30

SEQUENCE OF CORRECTED TEMPERATURES ($^\circ\text{C}$)

	Vapour	d	Juice	True temp. drop
Exhaust steam	120			
1st vessel	108.7	0.8	109.5	10.5 (18.9°F)
2nd vessel	97	1.3	98.3	10.4 (18.8°F)
3rd vessel	81.6	2.1	83.7	13.3 (23.9°F)
4th vessel	55	6.1	61.1	20.5 (36.9°F)
Total net drop				<u>54.7 (98.5°F)</u>

(*l*) *Corrected scale of temperatures.* It follows that the new temperature figures are those given in Table 32.30.

(*m*) *Calculation of heating surfaces.* Taking the new values for real specific evaporation coefficient corresponding to the foregoing temperatures, the heating surfaces of the different vessels become:

$$\begin{aligned}S_1 &= \frac{31,673}{5.54 \times 10.5} = 544 \text{ m}^2 & \text{Say: } & 550 \text{ m}^2 (6,050 \text{ sq.ft.}) \\ S_2 &= \frac{20,469}{4.21 \times 10.4} = 467 \text{ m}^2 & \text{Say: } & 475 \text{ m}^2 (5,150 \text{ sq.ft.}) \\ S_3 &= \frac{16,954}{2.81 \times 13.3} = 454 \text{ m}^2 & \text{Say: } & 450 \text{ m}^2 (5,150 \text{ sq.ft.}) \\ S_4 &= \frac{10,904}{1.28 \times 20.5} = 414 \text{ m}^2 & \text{Say: } & 425 \text{ m}^2 (4,650 \text{ sq.ft.}) \\ & & \underline{\qquad\qquad\qquad} & \underline{\qquad\qquad\qquad} \\ & & 1,879 \text{ m}^2 & 1,900 \text{ m}^2 (21,000 \text{ sq.ft.})\end{aligned}$$

Strictly speaking, we should now return to step (*h*) in the calculation with the new values for the evaporation coefficient, and deduce successively the values for t , r , uncorrected and

corrected A, and only then calculate the heating surfaces. In practice, when the corrected temperatures are as close to the original temperatures as those of Table 32.30 are to those of Table 32.27, this step is superfluous, and we may adopt the heating surfaces found.

Heating of the juice in the first vessel. In the preceding calculations, we have not taken into account the temperature of the juice as it enters the 1st vessel, which is the same as saying that we have assumed that the juice enters at a temperature equal to that of the juice in the vessel.

This, however, is generally not the case. The juice most often enters the 1st vessel at a temperature t_0 5- 15°C (10-25°F) lower than that of the boiling juice in the vessel. It is the 1st vessel which must supply the heat necessary to bring it to this boiling point. This requires some additional heating surface.

In practice, it is sufficient to take this into account by adding to the heating surface S_1 calculated for the 1st vessel a supplementary surface s_1 :

$$s_1 = 0.1 Q (t_1 - t_0) \quad (S_i = 0.6 Q (t_1 - t_0)) \quad (32.51)$$

s_1 = additional heating surface to be provided for the 1st vessel, in m² (sq.ft.).

Change in total heating surface. If we compare the total heating surface which we have obtained by method (B) with that obtained by the simple calculation (A), we are immediately struck with the small difference between the two results: not even 3%. This is due to two causes:

(a) The scale of pressures obtained by starting from conditions for minimum total surface is generally very close to the standard scale of pressures. In calculating a multiple effect conforming to the standard scale of pressures, we generally find, therefore, conditions very close to the optimum.

(b) If we take as a variable any one of the individual temperature drops for the vessels, and consider the total heating surface of the set as a function of this variable, the variation in heating surface in the neighbourhood of its minimum value is very small and very slow. We undergo only a small error then in departing, even quite appreciably, from the conditions of minimum total surface.

It follows that the heating surfaces obtained by either of the two preceding methods should be considered only as an indication. There is no great disadvantage in modifying them to an extent which if necessary can be substantial, if other considerations make this desirable.

In particular, if thermo-compression is used, it is necessary to limit the uncorrected temperature drop of the 1st vessel to a maximum of 5 — 10°C (10 — 20°F). The 1st vessel will sometimes be increased in heating surface as a consequence, and the following vessels may be decreased, but by a quantity which should obviously on the total be slightly less than the quantity by which the 1st has been increased (since we have departed from the conditions of minimum total heating surface).

In the case of a proposition for a multiple effect with thermo-compressor, the scheme will then be as follows: the apparent temperature drop for the 1st vessel will be fixed, for example at 115 — 110°C (240 — 230°F), then the other 3 vessels (in the case of a quadruple) will be treated as a triple effect operating between 110° and 55°C (230° and 131°F).

Multiple effect with equal vessels. Multiple effects were frequently found of which the different vessels are identical. The British firms in particular often supplied these, probably with the object of standardisation.

We may readily ascertain whether such an evaporator will operate under the conditions of minimum heating surface. In effect, if such were the case, the application of formulae such as (32.40) to (32.43) shows immediately that if:

$$S_1 = S_2 = S_3 = S_4$$

and if we take Δ_4 as unity, we have

$$\begin{aligned} \Delta_4 &= 1 \\ \frac{2\Delta_3}{\Delta_4} &= \frac{S_3}{S_4} = 1 & \Delta_3 &= \frac{\Delta_4}{2} \approx \frac{1}{2} \\ \frac{2\Delta_2}{\Delta_3 + \Delta_4} &= \frac{S_2}{S_3 + S_4} = \frac{1}{2} & \Delta_2 &= \frac{1 + \frac{1}{2}}{2 \times 2} \approx \frac{3}{8} \\ \frac{2A_1}{\Delta_2 + \Delta_3 + \Delta_4} &= \frac{S_1}{S_2 + S_3 + S_4} = \frac{1}{3} & \Delta_1 &= \frac{1 + \frac{1}{2} + \frac{3}{8}}{2 \times 3} = \frac{5}{16} \end{aligned}$$

The temperature drops Δ_1 , Δ_2 , Δ_3 , Δ_4 should then be respectively in the proportions:

$$\frac{5}{16} : \frac{3}{8} : \frac{1}{2} : 1, \quad \text{or:} \quad 5 : 6 : 8 : 16$$

With a triple effect, we should evidently have found 6 : 8 : 16. For a quintuple, we should have had 35 : 40 : 48 : 64 : 128.

Influence of inequality in the vessels. While multiple effects of equal vessels are still encountered, cases are often found also where one of the vessels is considerably smaller or larger than the others.

To avoid complicating our exposition of the problem, we shall disregard the case of vapour bleeding and base our calculations on a quadruple effect of 4 equal vessels. Starting with this quadruple, it is of interest to study what would be the influence on the total evaporation of a decrease or an increase of a certain amount in one of the vessels.

Take for example a standard quadruple effect with 4 equal vessels (a) each of 500 m², the total heating surface S being 2,000 m². We wish to resolve the following two problems:

1st problem. By how much would the evaporation obtained from this standard quadruple be decreased if one of the vessels, the 3rd for example, is replaced by a vessel of 400 m² i.e. if the surface S_3 is decreased by 20%, and in consequence the total surface S by 5%?

In other words, we should compare the evaporation obtained in the quadruple:

$$(b) \quad S_1 = 500 \text{ m}^2 \quad S_2 = 500 \text{ m}^2 \quad S_3' = 400 \text{ m}^2 \quad S_4 = 500 \text{ m}^2 \quad S' = 1,900 \text{ m}^2$$

with that obtained in the quadruple with 4 equal vessels (a) above.

In the same way, by how much would the evaporation be increased if the 3rd vessel were replaced by a much larger vessel, of 1,000 m² for example? That is, if the surface S₃ is increased by 100% and consequently the total surface S by 25%:

$$(c) \ S_1 = 500 \text{ m}^2 \quad S_2 = 500 \text{ m}^2 \quad S_3'' = 1,000 \text{ m}^2 \quad S_4 = 500 \text{ m}^2 \quad S'' = 2,500 \text{ m}^2$$

2nd problem. Would the increase or decrease of evaporation be the same whatever the place in the set of the vessel which differs from the others? In other words, if we had the choice between the quadruples (b) and (c) above, would it be better to place the vessel of 400 m² (or that of 1,000 m²) as the first body or as the fourth body?

The case of “doubling”, that is, of one vessel much larger and of a quadruple of the type (c), arises rather frequently, in particular when an evaporator set is enlarged while seeking for reasons of economy to keep the purchases of new equipment to a minimum. In deciding to use old material, there is often the possibility of doubling one of the vessels by putting two of the old vessels in parallel.

Now many people hold erroneous ideas on these questions. It is sometimes thought that the capacity of a multiple effect is determined almost entirely by the smallest vessel; and conversely, that “doubling” scarcely modifies the capacity of the evaporator set.

We shall solve the two problems at the same time. We shall assume that:

(1) The upper and lower limits of temperature remain the same in all cases, for example, 112° and 55°C (234° and 131°F). It follows that the specific evaporation coefficient of the 1st vessel remains constant, since it depends only on the upper limit of temperature (112°C, cf. eqn. (32.31)).

(2) The scale of brix values does not change. This is the same as saying that the quadruple effect receives the juice at the same brix B₀ and concentrates it to the same brix B₄, the intermediate brix values obviously remaining unchanged (cf. p. 562). This assumes that the rate of working of the factory is adjusted so that in each case it corresponds to the capacity of the multiple effect.

TABLE 32.31

EVAPORATION OBTAINED IN A QUADRUPLE EFFECT HAVING 3 VESSELS OF EQUAL HEATING SURFACE AND ONE OF A DIFFERENT HEATING SURFACE; RELATIVE TO THAT OF A QUADRUPLE WITH 4 VESSELS EACH IDENTICAL WITH ONE OF THE 3 EQUAL VESSELS

<i>H.S. of odd vessel % H.S. of each of others</i>	<i>H.S. of quad. % H.S. of quad. of 4 equal vessels</i>	<i>Evaporation obtained when odd vessel is:</i>			
		<i>1st</i>	<i>2nd</i>	<i>3rd</i>	<i>4th</i>
60%	90 %	87.8	87.2	85.9	83.7
80%	95 %	95	94.7	94.1	93
100%	100 %	100	100	100	100
150%	112.5%	107.8	108.3	109.5	111.6
200%	125 %	112.3	113.2	115.3	119

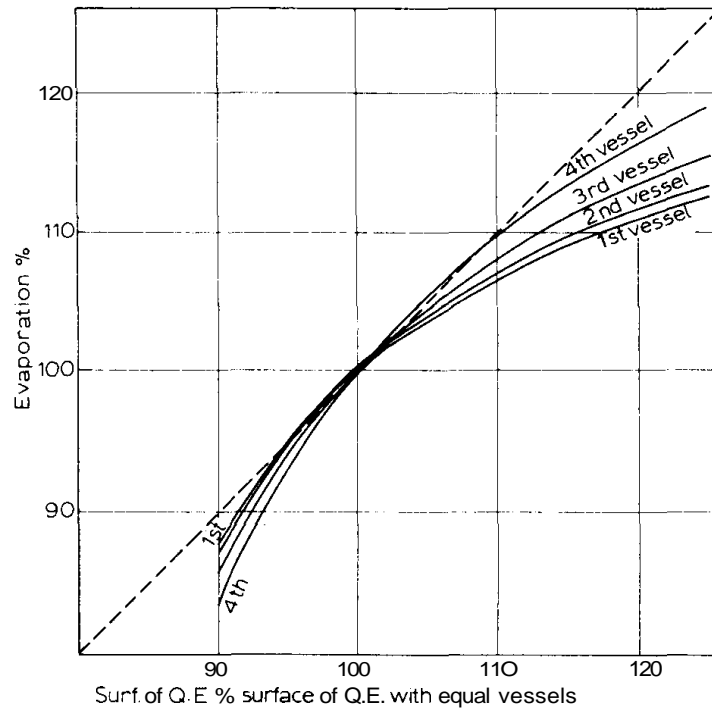


Fig. 32.48. Evaporation obtained in a quadruple effect having three vessels of equal area and one vessel of different area.

The calculation, carried out by Dessin's formula, leads to the result which we have illustrated by the graph in Fig. 32.48 and have summarised in Table 32.31. These graphs have been established by plotting as ordinate the evaporation obtained in the quadruple effect studied, which has 1 vessel of different surface from the others, this evaporation being expressed as a percentage of that of the standard quadruple effect with 4 equal vessels which would be formed if the different vessel were modified to the same heating surface as the other 3. We have plotted as abscissa the total heating surface of the quadruple effect studied, similarly expressed as a percentage of the heating surface of the standard quadruple.

Example. Suppose we have the quadruple effect (c) above. Its total heating surface S'' is equal to 125% of the surface S of the standard quadruple effect with 4 equal vessels (a). The evaporation which it should give will be found from Fig. 32.48 at the 125 point on the abscissa on the curve marked "3rd vessel", since it is the 3rd vessel which is non-uniform. We find then on the ordinate 115.3; that is, although the surface of the quadruple under consideration is 125% of the surface of the standard quadruple with equal vessels, the evaporation which it gives is only 115.3% of the evaporation of that standard quadruple.

Comments. (i) We see that, if we have 1 vessel smaller than the others, it is of advantage that it should be No. 1 in the set (it must not be forgotten that we are discussing a quadruple effect without vapour bleeding).

Conversely, if we have 1 vessel larger than the others, it is of advantage that this should be the last one in the set.

(2) "Doubling" is not as much of a disadvantage as is sometimes imagined (especially if the vessel added costs nothing), since in doubling the last vessel, for example, we obtain a gain in evaporation of 19%, which represents more than $\frac{1}{2}$ of the excess surface provided (25%).

(3) We see from the graph that there is hardly any advantage in making the vessels of different size, for a quadruple effect without vapour bleeding. In fact, if the scale of the graph were greater, it would be seen that there is advantage in having a 1st vessel slightly smaller and a 4th vessel slightly larger than the others. For a quadruple without vapour bleeding the most advantageous range of sizes would be 90, 100, 100, 110. This advantage, however, is practically negligible: it is shown by the extent (imperceptible on the graph) by which the two curves cross over, to the left and above the dotted straight line, in the neighbourhood of the points of abscissa 97.5 (1st vessel) or 102.5 (4th vessel).

We see also that any differences on the low side for the 4th vessel, and on the high side for the 3 previous vessels, become rapidly detrimental to the rate of evaporation.

(4) What we have just ascertained for a quadruple without vapour bleeding is applicable to all quadruples, on condition that the standard quadruple with equal vessels is replaced by a quadruple of the minimum heating surface which will cope with the duty required. The differences in heating surface above or below in relation to this optimum would have an analogous effect on the evaporation rate of the quadruple concerned.

In the same way, the conclusions just reached for the quadruple effect would also apply in the case of a triple or quintuple effect, since the fan shape of four curves would be replaced by another fan shape of 3 or of 5 curves disposed in a fully analogous manner.



Fig. 32.49. Quadruple effect (Fives Cail - Babcock)

(5) Equivalent surface. These graphs allow us to introduce the concept of "equivalent surface" of a badly balanced multiple effect. We have just seen that a quadruple effect such as (c) does not have a capacity equal to 125% of that of (a). In fact, it is only:

$$\frac{115.3}{125} = 92.2\%$$

of that of a well-balanced quadruple effect of 2,500 m². We may say that this badly balanced quadruple of 2,500 m² has a surface equivalent only to:

$$2,500 \times 0.922 = 2,305 \text{ m}^2$$

of that of a well-balanced quadruple.

(6) If the quadruple effect has 2 non-uniform vessels, we may still multiply by each other the 2 coefficients (such as 92.2% above) which each of these vessels would give if it were the only one not uniform with the others.

Checking performance of a multiple effect

The performance of a multiple effect may be checked by noting the brix of the juice and of the syrup and the vapour temperatures, and calculating from these the real specific evaporation coefficients.

Example. A quadruple effect is to be checked. At the time the check is made, the following observations are made:

Crushing rate of the factory	98 t.c.h.
Weight of clarified juice per t.c.	960 kg/t.c. (2,150 lb./t.c.)
Temperature of clarified juice entering the 1st vessel	96°C (205°F)
Brix of clarified juice	14
Brix of syrup	56
Vapour bled from 1st vessel	10,000 kg/h (22,400 lb./h)
Vapour bled from 2nd vessel	5,000 kg/h (11,200 lb./h)

	Heating surface (m ²) (sq.ft.)		Temperatures noted (°C) (°F)		
			Steam	115	240
1st vessel	1,000	10,750	1st vessel	105	221
2nd vessel	700	7,525	2nd vessel	94	201
3rd vessel	600	6,450	3rd vessel	81	178
4th vessel	550	5,925	4th vessel	54	129

Performance check. (a) *Calculation of evaporation.* The work of the quadruple is given by:

$$\text{Weight of juice} = 98 \times 960 = 94,080 \text{ kg/h (210,700 lb./h)}$$

$$E = 94,080 \left(1 - \frac{14}{56} \right) = 70,560 \text{ kg/h (158,000 lb./h)}$$

(b) Allowance for juice heating in 1st vessel. The 1st vessel has to heat the juice by about 10°C (106° – 96°). This heating corresponds to the immobilisation of a portion s_1 of the heating surface (eqn. (32.51)):

$$s_1 = 0.1 \times 94.08 (106 - 96) = 94 \text{ m}^2 (1,016 \text{ sq.ft.})$$

We shall therefore consider the 1st vessel exactly as if it were a vessel of heating surface $1,000 - 94 = 906 \text{ m}^2 (9,734 \text{ sq.ft.})$ receiving juice at its boiling point.

(c) Calculation of the individual evaporations. We have (cf. p. 589):

$$4x + 2 (5,000) + 10,000 = 70,560$$

hence:

$$x = \frac{70,560 - 10,000 - 10,000}{4} = 12,640 \text{ kg/h (28,300 lb./h)}$$

thus:

Evaporation from the 4th vessel = 12,640		= 12,600 kg/h
Evaporation from the 3rd vessel = 12,640		= 12,600 kg/h
Evaporation from the 2nd vessel = 12,640 + 5,000		= 17,640 kg/h
Evaporation from the 1st vessel = 17,640 + 10,000		= 27,640 kg/h
		70,560 kg/h (158,000 lb./h)

(d) Calculation of brix figures. The brix of the juice leaving and the average brix values of the juice in the various vessels are:

		<i>Average brix</i>
Entering 1st vessel: $B_0 = 14$	14	
Leaving 1st vessel: $B_1 = 14 \times \frac{94,080}{94,080 - 27,640} = 19.8$		$b_1 = 16.9$
Leaving 2nd vessel: $B_2 = 14 \times \frac{94,080}{66,440 - 17,640} = 27$		$b_2 = 23.4$
Leaving 3rd vessel: $B_3 = 14 \times \frac{94,080}{48,800 - 12,640} = 36.4$		$b_3 = 31.7$
Leaving 4th vessel: $B_4 = 14 \times \frac{94,080}{36,160 - 12,640} = 56$		$b_4 = 46.2$

(e) *Calculation of net temperature drops.* We calculate the boiling-point elevation in the vessels as indicated on pp. 497 – 501, and thus deduce the net temperature drops as shown below:

	Heating vapour		d		Vapour from juice		Net temp. drop	
	(°C)	(°F)	(°C)	(°F)	(°C)	(°F)	(°C)	(°F)
1st vessel	115	240	1.0	1.8	105	221	9	17.2
2nd vessel	105	221	1.5	2.7	94	201	9.5	17.3
3rd vessel	94	201	2	4.5	81	178	10.5	18.5
4th vessel	81	178	5.7	10.3	54	129	21.3	38.7

(f) *Checking the work of the several effects.* In each vessel we have:

$$q = csA = f(100 - b)(T - 54) SA \quad \left(q = csA = f \frac{(100 - b)(T - 130) S\Delta}{16,000} \right)$$

where f is a factor to be applied in the Dessin formula (32.31).

From this we deduce:

$$f = \frac{q}{(100 - b)(T - 54) SA} \quad \left(f = \frac{16,000 q}{(100 - b)(T - 130) SA} \right)$$

We then have:

$$\text{1st vessel: } f_1 = \frac{22,540}{(100 - 16.9)(115 - 54) 906 \times 9} = 0.00067 \text{ (0.65)}$$

$$\text{2nd vessel: } f_2 = \frac{17,640}{(100 - 23.4)(105 - 54) 700 \times 9.5} = 0.00068 \text{ (0.70)}$$

$$\text{3rd vessel: } f_3 = \frac{12,640}{(100 - 31.7)(94 - 54) 600 \times 10.5} = 0.00073 \text{ (0.78)}$$

$$\text{4th vessel: } f_4 = \frac{12,640}{(100 - 46.2)(81 - 54) 550 \times 21.3} = 0.00074 \text{ (0.75)}$$

and we can thus deduce that our quadruple is operating at about 70% of the rate for good normal conditions, as expressed by the coefficient 0.001 (divisor 16,000); but all vessels show similar figures, no one vessel presents an anomaly. It is probable that scale alone is to blame, and that its severity decreases slightly from the first to the last two vessels.

Frequently it is found that one of the vessels is operating less effectively than the others, and the cause will then be sought: scale incompletely removed, defective cleaning, leaks, blockage of a vapour pipe, juice pipe, water or incondensable-gas pipe, etc.

Installation of a thermo-compressor

There are certain conditions to be realised when a thermo-compressor is installed. The more important are the following:

- (1) The weight y of vapour aspirated is limited to a certain maximum
- (2) The weight x of vapour passing from the last vessel to the condenser, on the contrary, cannot be decreased below a certain minimum.

First condition. Maximum weight of vapour aspirated. Thermo-compression involves a saving in steam. This saving is of value insofar as it decreases the make-up steam to be supplied by the boilers to manufacture. However, as soon as one has been successful in reducing this make-up to zero, there is no further advantage in increasing the thermo-compression; the consumption of steam would fall below the quantity of exhaust steam produced, and it would be necessary to allow steam to escape through the roof by the safety valve. Thus the heat saved in the exhaust steam would be lost in the live steam together with loss of actual steam and intermediate heat losses.

There is therefore a maximum economy to be realised by thermo-compression; a means of regulating the thermo-compression should be installed such that the consumption of exhaust steam should always be equal to or slightly greater than the production.

Now the production of exhaust steam corresponds to the quantity furnished by the turbo set or the engines, decreased by losses due to condensation, leaks, etc. Say: M .

The consumption of exhaust steam, Q , is represented by the 3 quantities: steam supplied to the 1st vessel of the evaporators, to the pans, and to the juice heaters. Let:

$$Q = q' + C + R \tag{32.52}$$

- Q = total weight of exhaust steam
- q' = weight of exhaust steam supplied to the 1st vessel of the multiple effect
- C = weight of exhaust steam used at the pans
- R - weight of exhaust steam used at the heaters.

It is necessary that:

$$q' + C + R \geq M \tag{32.53}$$

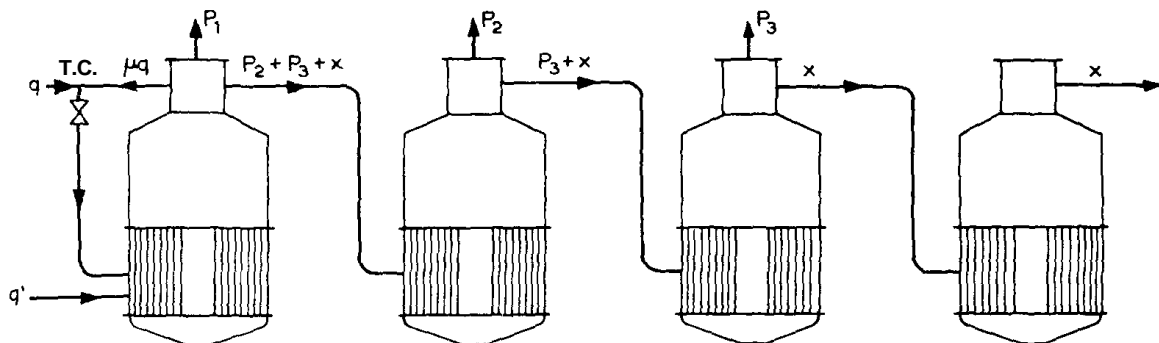


Fig. 32.50. Maximum thermo-compression.

or:

$$q' \geq M - C - R \quad (32.54)$$

This quantity $M - C - R$ is known or is readily calculated.

On the other hand, eqn. (32.20) (p. 566) applied to the case of a multiple effect with thermo-compression (Fig. 32.50) shows that the total evaporation E has a value:

$$E = nx + P_1 + 2P_2 + 3P_3 + \mu q \quad (32.55)$$

and equating the weight of heating steam to the weight of vapour produced, in the 1st vessel, we have:

$$q + \mu q + q' = \mu q + P_1 + P_2 + P_3 + x$$

or:

$$q + q' = x + P_1 + P_2 + P_3 \quad (32.56)$$

Eliminating x by taking its value from eqn. (32.55) and substituting in (32.56), we have:

$$q' = \frac{E + (n - 1) P_1 + (n - 2) P_2 + (n - 3) P_3}{n} - \frac{n + \mu}{n} \cdot q \quad (32.57)$$

Substituting this value in eqn. (32.54), and solving for q , we find:

$$q \leq \frac{1}{n + \mu} [E + (n - 1) P_1 + (n - 2) P_2 + (n - 3) P_3 - n (M - C - R)] \quad (32.58)$$

which may also be written:

$$y \leq \frac{\mu}{n + \mu} [E + (n - 1) P_1 + (n - 2) P_2 + (n - 3) P_3 - n (M - C - R)] \quad (32.59)$$

$y = \mu q$ = weight of vapour entrained by the thermo-compressor.

Effect on heat balance. It is interesting to study the inequality (32.58). It gives the quantity of steam which can be used for thermo-compression in a typical factory, but it also indicates under what conditions there is a risk of having an excess of exhaust steam. There is an excess of exhaust from the moment when the second member of the expression vanishes or becomes

negative. At that moment, the thermo-compression ceases to be of any value:

$$E + (n - 1) P_1 + (n - 2) P_2 + (n - 3) P_3 = n (M - C - R) \quad (32.60)$$

It must be kept in mind that, if there is vapour bleeding to the vacuum pans, it is necessary to add to the quantities P_1, P_2, P_3 used for juice heating, the vapour bled to the pans, P'_1 and P'_2 . In a new factory, or one where the evaporators have been modernised, it is possible to *arrange* for all juice heating by vapour bleeding: thus $R \approx 0$. It would also be possible to boil all the pans on vapour from the first or even the second vessel, and thus $C = 0$. The exhaust steam available will be reduced in this case to the term M . Let us see what happens then to these three terms.

R. As far as juice heating is concerned, it may be commented that for a given factory, heat requirements are practically fixed. Neglecting the few degrees of cooling of the juice before heating, these requirements are limited to a total, per tonne of juice and thus approximately per t.c., of:

$$\frac{1000 \times 0.9 (105 - 30)}{540 \times 0.75} = 132 \text{ kg/h (295 lb./h) of steam} \quad (32.61)$$

P_1 is generally of the order of 20 – 35, P_2 30 - 40, and P_3 60 kg/h. If $P_3 = 0$, P_2 approaches close to 100 kg/h. The 3 terms $(n - 1) P_1 + (n - 2) P_2 + (n - 3) P_3$ make a total of the order of:

- (a) 180 kg/h (400 Ib./h) for quadruple effect with bleeding from 2 effects
- (b) 214 kg/h (480 Ib./h) for quadruple effect with bleeding from 3 effects
- (c) 337 kg/h (755 Ib./h) for quintuple effect with bleeding from 3 effects

C. As for the pans, their vapour consumption is close to:

- (a) 120 kg/h (270 Ib./h) for continuous pans on A, B and C massecuite
- (b) 136 kg/h (305 Ib./h) for continuous pans on B and C, and batch pans for A massecuite
- (c) 148 kg/h (332 Ib./h) for batch pans on all massecuites

With bleeding for example from the first vessel of a quadruple, the 136 kg/h would give:

$$(n - 1) P'_1 = 3 \times 136 = 403 \text{ kg/h (915 Ib./h)}$$

M. Machines driven by steam turbine will consume:

- (a) *Shredder:* If it is of typical power: 2.25 kW/t.c.h. \times 15 kg/kWh \approx 34 kg/h (75 Ib./h). If it is a high-power machine: 5 kW/t.c.h. \times 15 kg/kWh \approx 75 kg/h (165 Ib./h) of steam.
- (b) *Mills:* For 5 mills each taking 1.75 kW/t.c.h.: 8.75 \times 15 kg/kWh \approx 130 kg/h (290 Ib./h). For 6 mills each at 2 kW/t.c.h.: 12 \times 15 \approx 180 kg/h (400 Ib./h) of steam.

The turbo-alternators at the powerhouse, supplying all the electric motors of the boiling house and the knives, consume approximately 18 kW/t.c.h. (24 h.p./t.c.h.) at 11 kg steam

per kWh, or 200 kg/h (450 lb./h) of steam.

Hence we should have, if all requirements are supplied by vapour bleeding, for example:

$$(a) \text{ Steam consumption: } \begin{aligned} 800 + 214 + 408 &= 1,422 \text{ kg/h (3,140 lb./h) (quadruple)} \\ 800 + 337 + 408 &= 1,545 \text{ kg/h (3,400 lb./h) (quintuple)} \end{aligned}$$

$$(b) \text{ Steam production: } \begin{aligned} 34 + 130 + 200 &= 364 \text{ kg/h, leaving 350 kg/h (784 lb./h) of ex-} \\ &\text{haust steam (M.P.)} \\ 75 + 180 + 200 &= 455 \text{ kg/h, leaving 440 kg/h (985 lb./h) of ex-} \\ &\text{haust steam (H.P.)} \end{aligned}$$

Now: $4 \times 350 = 1,400$ kg/h (3,136 lb/h) and $5 \times 440 = 2,200$ kg/h. It will be seen that in the latter case we arrive at a very high excess of exhaust steam. In other words, factories so well advanced in steam economy have no interest in pushing the vapour bleeding for pans to the full extent possible, as this would only serve to cap the factory with a plume of escaping steam - a useless and expensive procedure. Thus thermo-compression is of no further interest to them.

Second condition. Minimum work of the last vessel. The condition (32.59) fixes a maximum duty for the thermo-compressor and for the 1st vessel. However, this involves a condition of minimum duty for the latter vessels.

In practice, E , P_1 , P_2 and P_3 are generally known and in any case eqn. (32.55) shows that, for the maximum imposed on q by (32.58), there corresponds a minimum for x .

To ascertain this value, we take the value of q from eqn. (32.55) and substitute it in (32.58). We then have:

$$x \geq \frac{1}{n + \mu} [E - P_1 - 2P_2 - 3P_3 - \mu (P_1 + P_2 + P_3) + \mu (M - C - R)] \quad (32.62)$$

Variation in steam consumption with thermo-compression. Equation (32.57) may be written:

$$(n + \mu) q + nq' = E + (n - 1)P_1 + (n - 2)P_2 + (n - 3)P_3$$

or:

$$(n + \mu) q + nq' = \text{constant} \quad (32.63)$$

When the regulator of the thermo-compressor comes into action and opens or closes the admission of actuating steam, the respective weights of high-pressure actuating steam (q) and of exhaust steam entering the 1st vessel (q') should vary relatively according to eqn. (32.63).

Example. If: $n = 4$ (quadruple effect) and $\mu = 2$, we shall have:

$$3q + 2q' = \text{constant}$$

In other words: when the thermo-compressor absorbs an additional 2 kg of high-pressure actuating steam, the 1st vessel of the evaporators consumes 3 kg less of exhaust steam.

It is mainly the variations in steam consumption of the pans (*i.e.* of the term C in the expression $M - C - R$) which cause the most important and the most abrupt changes in the regulator of the thermo-compressor, and in its output.

Regulator. Regulation of the output of the thermo-compressor according to the available exhaust steam is obtained by separating the thermo-compressor into several separate nozzles which are opened in turn by a regulator actuated by the exhaust-steam pressure.

For example, 3 nozzles would be chosen with respective outputs 1, 2 and 4, and the regulator would then open in succession:

Nozzle	Relative output
1	1
2	2
1 and 2 together	3
4	4
1 and 4 together	5
2 and 4 together	6
1, 2 and 4 together	7

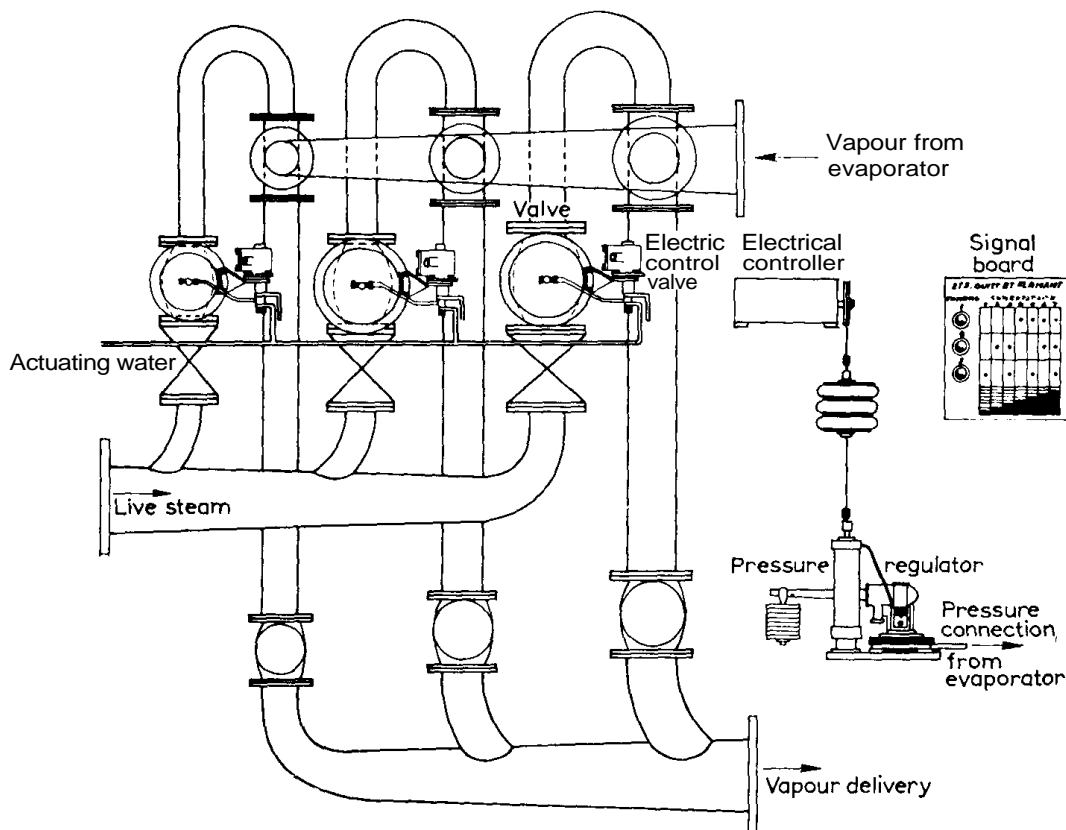


Fig. 32.51. Regulator with three nozzles.

In this way the quantity of actuating steam (or of aspirated vapour) may be divided into 7 steps, permitting the regulator to follow flexibly the conditions determined by the availability of exhaust steam (Fig. 32.51).

Still greater flexibility may be obtained by installing 4 nozzles of outputs 1, 2, 4 and 8, respectively, giving all combinations from 1 to 15.

This method maintains the idea of having each nozzle work at its full rate, since the efficiency of a nozzle falls very quickly when it works at a rate differing from that for which it has been designed.

Combination 7 (1st case) or 15 (2nd case) is opened when the exhaust-steam pressure tends to fall very low. In proportion as it rises, the regulator closes in succession combinations 7, 6, 5, ..., and finally closes nozzle No. 1. If the exhaust-steam pressure continues to rise, the safety valve on the exhaust main will then discharge through the roof.

The designers have succeeded in giving these regulators such a sensitivity that all the steps of output from the opening of combination 7 to the closing of nozzle 1 may be obtained within a pressure difference of $\pm 0.025 \text{ kg/cm}^2$ ($\pm 0.35 \text{ p.s.i.}$) from the value chosen.

EVAPORATOR OPERATION

Capacity of evaporators

Table 32.32 gives figures for the practical capacity of multiple effects, taken from summaries embodying all factories of a given country. They are given in m^2 (sq.ft.) total heating surface per t.c.h.

TABLE 32.32

INSTALLED CAPACITY OF MULTIPLE EFFECTS IN VARIOUS COUNTRIES

Country	<i>n</i>	Min.	Max.	Average	
				($\text{m}^2/\text{t.c.h.}$)	(sq.ft./t.c.h.)
South Africa	Triple	17.3	28	25.5	275
	Quadruple	25.7	54	31.4	338
	Quintuple	30.9	41.9	36	388
Puerto Rico	Triple	13.7	17.1	15.7	169
	Quadruple	13.9	33	23.8	256
	Quintuple	15.8	37	26.5	285
Cuba	Triple			20	215
	Quadruple			25	269
	Quintuple			30	323
Louisiana	Triple			18	194
	Quadruple			23	248
	Quintuple			33	355

We give the lowest value stated, the highest⁷¹, and the general average or the standard value accepted in the country^{72, 73}.

The capacity of a multiple effect depends very largely on the back-pressure. Estimated values for a normal vacuum of 64 cm (25 in.) of mercury, with various values on back-pressure, are given in Table 32.33.

TABLE 32.33

MEAN HEATING SURFACE TO BE PROVIDED IN MULTIPLE EFFECTS, IN m²/t.c.h. (sq.ft./t.c.h.)

Back pressure, in kg/cm ² (p.s.i.)	0.5 (7)	1.0 (14)	1.5 (21)
Steam temperature To, in °C (°F)	111 (232)	120 (248)	127 (260)
Triple effect	20 (215)	17 (180)	16 (170)
Quadruple effect	28 (300)	24 (260)	22 (235)
Quintuple effect	36 (390)	31 (335)	28 (300)

Distribution of heating surface between vessels

Basing our considerations on a typical quadruple effect, with extreme pressure of 1 kg/cm² (120°C), we indicate below the normal vapour bleeding which could be envisaged, also the corresponding evaporation rates for each vessel:

TABLE 32.34

Effect no.	Vapour from juice		6	True Δt	c	Evap. (kg/t.c.)	H.S. (m ² A.c.h.)	Vapour bleeding			
	Press.	Temp. (°C)						Pans (kg/h)	Heaters (kg/h) T - t	H.S. (m ²)	
1	500 g	111	1.5°	7.5°	5.5	370	10	140	28	105-90	1.8
2	0	100	2°	9°	4.4	200	6		36	90-70	1.5
3	-33 cm	85	3.5°	11.5°	3	160	5		70	70-30	2
4	-61 cm	55	7°	23°	1.4	90	4				
							820	25	140	134	

We may reckon approximately on the following distribution of total heating surface between the vessels (per t.c.h.):

Bleeding from:	Quadruple				Quintuple	
	1st, 2nd effects (m ²)	effects (sq.ft.)	1st, 2nd, 3rd effects (m ²)	effects (sq.ft.)	1st, 2nd, 3rd effects (m ²)	effects (sq.ft.)
1st effect	10	110	10	110	10	110
2nd effect	6	66	6	66	7	77
3rd effect	4	44	5	55	6	66
4th effect	5	55	4	44	4	44
5th effect					5	55
	25	275	25	275	32	352

Flow of juice or syrup for each tube. When the calculation of a multiple effect has been completed with the dimensions of vessels and their tubes, and the number of tubes, it is advisable to verify the flow of juice or syrup for each tube. Generally, for regular operation, it is necessary to maintain 40 kg/h (90 lb./h) of juice or syrup per tube, calculated on the quantity entering the vessel. This rate is easily attained for the first vessel, but not for the later ones. Where it is not reached, it is necessary to provide for recycling juice between the outlet and the inlet of the vessel, in order to assure complete wetting of the tubes and avoiding periods of operation with partly dry tubes at times of low juice flow. The rate of 40 kg/h/tube is considered as optimal, and the recycling pump is calculated on this quantity.

Control of the multiple effects

During the period when the multiple effects required a special operator (generally known as the effect boiler) responsible for the control of the station, the three principal points to which he had to pay attention were:

- (a) The juice level in the vessels (*cf.* p. 533)
- (b) The incondensable gases (*cf.* p. 528)
- (c) Brix of the syrup (*cf.* p. 535)

Today, levels and brix are controlled by automatic regulators and there remain only the incondensable-gas valves to be opened to a sufficient and not excessive extent.

Where the evaporators are of ample capacity, the brix of syrup may be regulated either by means of the steam admission valve to the first effect, or by regulating the vacuum at the last effect. The latter process is preferable as it allows the vapour temperatures of the various effects to be increased in turn, each one influencing the preceding, and thus enables a better efficiency to be obtained from the vapour bleeding and thus from juice heaters, pans and from the multiple effects themselves.

Where a special heater-condenser is installed for the multiple effects, it is similarly of advantage to place the vacuum control of the multiple effects after the heater-condenser, so that the latter may benefit by the arrangement.

Maximum vacuum. We have already seen (p. 541) that it is advantageous not to exceed a certain vacuum, in order to avoid entrainment. Furthermore, when the vacuum is high, the syrup temperature falls in consequence. It was considered in Java that the minimum temperature acceptable for the syrup at entry to the pans was 60°C (140°F). Below this, false grain is liable to form at each charge of syrup, and in order to avoid this disadvantage, it would be necessary to re-heat the syrup; this would be troublesome.

Now 60°C (140°F) in the syrup corresponds to approximately 55°C (131°F) for its vapour, say 64 cm (25 in.) vacuum, and we should also take into account the cooling of the syrup in the storage tanks. It would be of advantage to lag these carefully.

When the equipment permits the evaporators and the pans to be operated at different vacua, the same consideration should lead to operating the evaporators at a lower vacuum than the pans. This allows the syrup to leave the multiple effects at a higher temperature than that ruling in the vacuum pans, and consequently, allowing for a certain unavoidable cooling and for the boiling-point elevation of the massecuite, greater than that of the syrup, allows the syrup to enter the pans at a temperature substantially equal to that of the boiling massecuites.

Leaks. It happens sometimes that the vacuum does not hold, although all visible leaks have been stopped. It is possible then that the cast iron of the vessels or the vacuum piping may be porous.

In this case the best procedure is to run the vacuum pump to establish a certain vacuum, and then to paint the whole installation with a thick paint. The paint will penetrate into the pores of the metal, and will seal them as it dries.

Locating leaks. When the vacuum does not hold, a search will be made for leakages. For this purpose, close all the cocks, valves, etc., of juice, condensate, incondensable gases, vacuum, etc. and establish a vacuum.

If this does not reach 68.5 cm (27 in.), the leak is located in the vacuum piping.

If the vacuum attains 68.5 cm, the vacuum valve to the last vessel will then be opened. If the vacuum here is almost as high, then the incondensable-gas valve from the last vessel will be opened, which will put the third vessel under vacuum. This is continued with all vessels in turn, until a more or less rapid fall in vacuum discloses the vessel at fault.

It is considered that a vessel is sufficiently free from leaks when, after raising vacuum for the test to 63.5 cm (25 in.), it does not fall more than 13 mm ($\frac{1}{2}$ in.) in a quarter of an hour.

Purity rise during evaporation. Nearly always a rise in purity of juice is detected in the course of evaporation. In Java this purity rise has been found approximately equal to the following values:

Defecation factories:	0.8
Sulphitation factories:	0.9
Carbonatation factories:	0.4

According to Marches⁷⁴, the causes of this rise are the following:

(1) *True purity rise.* A certain elimination of non-sugars occurs during the evaporation owing to:

(a) Gases, CO₂, NH₃ and glyoxal, which escape with the vapour. This corresponds to a purity rise of approximately 0.03.

(b) Materials in solution which deposit on the tubes. This corresponds to a rise of about 0.04, or 0.1 at most.

(2) *Apparent purity rise.* On the other hand:

(a) A transformation of reducing sugars occurs, which shows up as a decrease in their rotatory power. Hence an apparent purity rise slightly greater than the two preceding ones.

This increase is greater when the temperature is higher. It is greater with a pre-evaporator or evaporation under pressure than in an ordinary evaporator.

(b) In the laboratory, the syrup is diluted for analysis, while the juice is not. This causes an error in the brix and the polarisation, due to the concentration, and leads to an apparent purity rise which is difficult to evaluate.

Possible causes of unsatisfactory operation of a multiple effect

When a multiple effect is not giving the results which could be expected of it, it is necessary to seek the cause or causes of this poor operation.

This cause could be common to all vessels, or peculiar to one vessel. In the latter case it is sometimes difficult to locate it. There would be no question of attempting to do this by a measure or an estimation of the quantity of vapour given off (or of the corresponding condensate), since bad work at one vessel influences the others and since, in a quadruple without vapour bleeding, for example, the quantities of water evaporated in each of the 4 vessels would in all cases be approximately equal.

On the contrary, the object will be to find out what temperature drop each vessel requires in order to furnish, with the available heating surface, the evaporation which it is giving. For this the method of checking already described (p. 600) will be employed. But it is still possible to reach a negative result, according to the real cause of the trouble. For example, if it is a case of a leak in the bottom of a calandria, the calculation will not give any conclusion: it would be necessary, in this case, to be able to determine the brix of the juice leaving each vessel, and to compare it with that which normal operation should give. Unfortunately there is generally no means of obtaining a sample of the juice leaving the various vessels, except the last one.

The following are several possible causes of faulty operation:

(a) Excessive deposits of scale in one or several vessels. Insufficient cleaning, leaving a hard layer sometimes difficult to detect (*cf.* the case mentioned in the second paragraph under *Distribution of scale* on p. 547).

(b) Layer of grease on the exterior of the tubes of the 1st vessel. In addition to the methods which we have discussed for removing this during the slack season (*cf.* p. 546), Selman and Plomley⁷⁵ give a method applicable during the operation of the equipment: inject into the steam reaching the first calandria about 150 l (30 gal.) of kerosene, in the ratio of 2.5–5 l/min (0.5 - 1 gal./min), by means of a hand pump. Care must be taken during this operation, to discard the condensate from this calandria.

(c) A leak at the top or particularly at the base of a tube or of a calandria. To locate this, try the several vessels under water pressure, taking advantage of the weekend shutdown.

(d) Blockage or poor location, or insufficient opening, of the incondensable-gas withdrawal pipes.

(e) Entrance of air into a vapour pipe, a vessel or a calandria.

(f) Blockage of a connecting juice pipe by a foreign body.

(g) Juice level too far from the optimum (*cf.* p. 533). This fault is particularly serious in the last two vessels.

SPECIAL MULTIPLE EFFECTS

In addition to the standard type, which provides by far the great majority of multiple effects, there exists a multitude of types of evaporator body, but generally they are of restricted interest only. The only special type which is fairly interesting and widespread is the Kestner.

Kestner evaporator

This model was designed at the beginning of the 20th century by the French engineer Paul Kestner. It is based on the principle of the "climbing film": when a liquid is heated in a very long vertical tube, the vapour bubbles which form at the base of the tube increase in size as they rise, and their diameter rapidly attains that of the tube. From then on, the upper part of the tube is traversed only by, first, a chain of bubbles, then a film of juice, against the tube, entrained by the friction of the upward current of vapour. The transmission of heat is thus favoured by:

(1) The decrease in the mean distance between the particles of liquid and the heating surface

(2) The very high velocity of the liquid.

Length of tubes. Kestner evaporators generally have tubes 7 m (23 ft.) in length. However, Claassen⁷⁶ has shown that the coefficient of heat transfer of a Kestner passed through a maximum with a tube length of 4.9 m (16 ft.) (The diameter of the tubes concerned is not stated.)

Diameter of tubes. The tubes have diameters varying from 27 to 38 mm (1 - 1½ in.). In general, a certain relationship is maintained between the diameter and length of the tubes, the tube length being greater as the diameter increases (this is the opposite of what is done in ordinary evaporators, *cf.* p. 511), so as to obtain the climbing-film effect which here is indispensable.

Juice level. The Kestner principle, by realisation of the climbing film, permits of a decrease of the hydrostatic pressure which would be caused by the greater length of the tubes. In effect, the bubbles, when they occupy the full diameter of the tube, break the continuity of the liquid column, in such a way that the ordinary hydrostatic law is not applicable. The pressure which this column exerts at the bottom corresponds to a column partly liquid, partly gas, and much lighter than a continuous liquid column in which small bubbles do not affect the pressure. Further, the climbing film produces, from the friction of the vapour with the film of juice, a negative dynamic effect, just as a liquid which is flowing in the opposite direction towards the bottom would produce on the liquid at the bottom a positive dynamic effect which would be added to the static pressure.

It follows that the optimum level of juice in a Kestner corresponds to a liquid column of height equal to about 20% of the length of the tubes, instead of 30 or 35% in ordinary evaporators.

Vapour separator. A Kestner evaporator may be arranged in two different ways, from the point of view circulation of vapour and juice. There is the ordinary arrangement (Fig. 32.52), most frequent in the beet industry, and the arrangement with an independent tangential separator (Fig. 32.53), which is recommended in the cane sugar factory.

The separator, in the latter case, is located between two successive vessels, and slightly towards the rear. It is sometimes rather difficult to find space for it.

Conditions for good performance. The theoretical advantage of the climbing film is not generally realised in practice with the Kestner. When a Kestner installation is designed, the

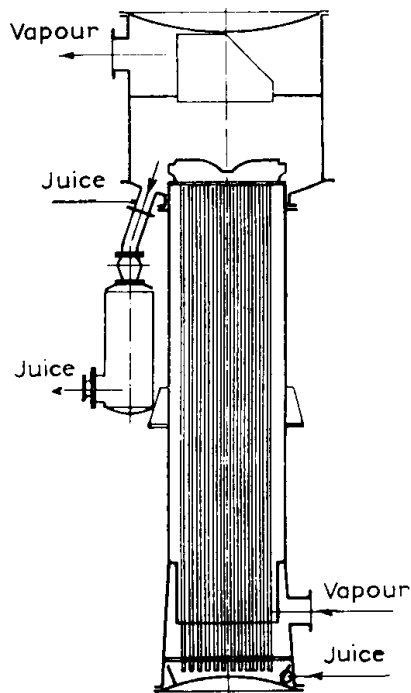


Fig. 32.52. Kestner evaporator.

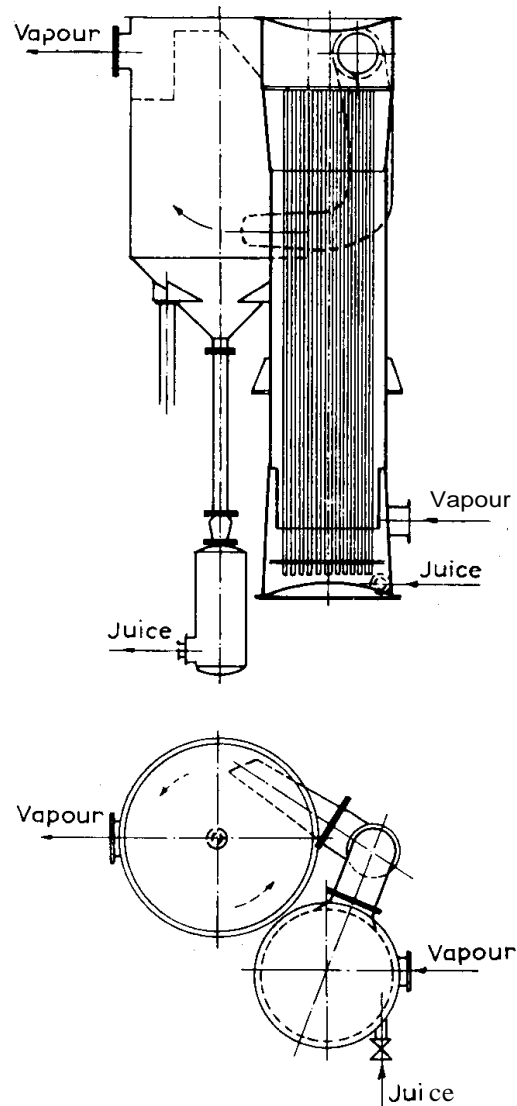


Fig. 32.53. Kestner with independent tangential separator.

same heat-transfer coefficients are used as for ordinary vessels. The difference, if there is one, is very slight.

On the other hand, the Kestner is very exacting from certain points of view. The climbing film actually exists only on condition that the juice is boiling. If it arrives in the vessel at a temperature appreciably lower than its boiling temperature, it must be heated up to that temperature. It is necessary then that a certain portion of the heating surface should serve as a heater. Now in the Kestner the juice traverses the tubes only once, as it rises; and the heat transfer is much worse in the fraction of the tube which works under submerged conditions and performs the juice heating, than in the portion which works as an evaporator, where the liquid is climbing. The fraction functioning as a heater thus corresponds to the submerged portion of the tubes, since a climbing film cannot exist there. It is obviously desirable that this fraction be as short as possible.

It follows that a Kestner *will not function satisfactorily unless it receives juice very close to its boiling point*. This is the point in which the Kestner is inferior to ordinary evaporators which, themselves, are very well suited for heating the entering juice as well as evaporating it.

Since it is especially as a first vessel that Kestner evaporators are installed, this point is important. A Kestner will only be installed as a first effect on condition that it is fed with juice at a temperature within 3°C (9°F) of the boiling point of the juice in that vessel. Preferably the juice would be passed through a heater using exhaust steam before the Kestner, so as to raise the juice to a temperature equal to or higher than that of the juice in the vessel.

Application of Kestner. The juice forms a climbing film only when it is light; a thick and viscous syrup climbs badly. In the same way, steam at high temperature causes a more violent ebullition than vapour under vacuum. It is for these two reasons that the tendency is to employ the Kestner rather as a 1st vessel than as a complete multiple effect. However, triple- and even quadruple-effect Kestners are sometimes found.

Advantages and disadvantages. (a) *Advantages.* The Kestner takes up less floor space. The juice passes through it very rapidly, in about 30 s, and this reduces risks of inversion and coloration when working at high pressure and temperature. If it is functioning well, scale formation is much slower, and we know of one factory possessing a Kestner first effect, very generously designed, which is never cleaned either with mechanical cleaners or with acid during the whole crushing season. Cleaning is carried out only in the slack season. This is a great advantage, which unfortunately is impossible to guarantee beforehand.

(b) *Disadvantages.* On the other hand, the Kestner requires high head room, and it is rarely that it can be installed in a building served by a travelling crane without obstructing the passage of the crane. If it is provided with an independent separator, it takes up more space in the horizontal plane. It requires hot juice or a supplementary heater.

On account of the very brief transit time of the juice, a Kestner multiple effect cannot be employed to store the juice, as is often done with ordinary multiple effects, in order to smooth out fluctuations between clarification and the pan stage; on the contrary, it requires a storage tank capable of holding a substantial quantity of clarified juice.

Summary. In short, the Kestner will be employed for preference, either as a 1st effect, or in a triple effect under pressure, on a very clean juice which has been well heated.

EVAPORATION UNDER PRESSURE

We have seen (p. 567) that the steam economy of an evaporator set depended on the possibility of bleeding vapour from the vessels further on in the set. But at the same time, we concluded that the latter, in ordinary multiple effects, offered only slight possibilities of vapour bleeding, since the vapour which they furnish was at low temperature, and had little value and little application.

Consideration has been given to raising the lower limit of the temperature range of a multiple effect in such a way that the vapour from the last vessel would be at a temperature sufficient to play its full part in the principal heating applications in the factory.

At the same time it has been necessary to raise the upper limit of the range of temperatures,

otherwise the overall temperature drop available would not be sufficient to permit the installation of a multiple effect.

On the other hand, the total temperature drop becomes seriously reduced, and it is generally necessary to limit the installation to a triple effect.

Finally, the most frequent solution of a multiple effect under pressure is a triple effect functioning between 135 – 140° and 100 – 105°C (275 – 285° and 212 – 220°F).

Table 32.35 gives several values for temperatures and heat-transfer coefficients obtained with a triple effect functioning under these conditions:

A = net temperature drop in the vessel, in °C (°F) (total drop = 28.5°C)

r = latent heat of vaporisation, in kcal/kg (B.T.U./lb.)

K = heat-transfer coefficient in the vessel, in kcal/m²/°C/h (B.T.U./sq.ft./°F/h)

c = true specific evaporation coefficient for the vessel, in kg/m²/°C/h (lb./sq.ft./°F/h) net temperature drop.

On account of the high temperature reached by the juice in the 1st vessel, it is of advantage to make this triple effect in the form of a Kestner, so as to reduce the time of exposure of the juice to these high temperatures.

TABLE 32.35

TRIPLE EFFECT UNDER PRESSURE: TYPICAL EXAMPLE - BEET FACTORY

	<i>Temp. of heating steam</i>		<i>Temp. of vapour from</i>		<i>Temp. of juice</i>		Δ	<i>r</i>	<i>K</i>	<i>c</i>
	(°C)	(°F)	(°C)	(°F)	(°C)	(°F)	(°C) (°F)			
1st effect	138	280	128	262	128	262.5	10 17.5	520 (936)	2,587 (530)	5.04 (0.574)
2nd effect	128	262	116	241	117	243	11 19	528 (950)	1,611 (330)	3.05 (0.347)
3rd effect	116	241	104	219	108	226	8 15	536 (965)	879 (180)	1.69 (0.192)

With the temperatures of juice indicated, most of the heating can be effected with vapour from the last vessel. The latter is therefore calculated with the object of supplying vapour for as much heating as possible; vapour bleeding is not carried out from the 2nd and 1st vessels beyond the minimum necessary for the heating at the highest temperatures. Thus the maximum economy is obtained.

In order to cope with variations in rate and temperature in the factory, the practice is never to use all the vapour theoretically available from the last vessel, and a certain quantity is left to pass to the condenser. A solution sometimes adopted consists of following the 3rd vessel by a "concentrator", which is simply a small 4th vessel under vacuum, designed to absorb the excess vapour left as a safety measure, and preceding the condenser.

Campbell Macdonald⁷⁷ cites a triple effect under pressure, in which the vapour from the 3rd vessel is at 106°C (223°F) and which is followed by a concentrator functioning between 106° and 84°C (223° and 183°F). The heating surfaces suggested in this case are:

	Heating surface		Heat-transfer coeff., <i>K</i>		Evap. coeff., <i>r</i>	
	(m ²)	(sq.ft.)	(kcal/m ² / °C/h)	(B.T.U./ sq.ft. /°F/h)	(kg/m ² / °C/h)	(lb./sq.ft./ °F/h)
1st effect	1,000	10,760	2,616	535	5.01	0.57
2nd effect (2 vessels of 750 m ²)	1,500	16,140				
3rd effect	2,803	30,128	1,680	344	3.16	0.36
(4 vessels of 700 m ²)			852	175	1.58	0.18

The author proposes that the lengths of tube should decrease and their diameters increase from the 1st to the last vessel, in keeping with the increase in brix.

Application. The multiple effect under pressure functioning, for example, between 135 and 105°C (275 and 221 °F) would obviously require a greater heating surface than that of a triple, or of a quadruple, or even of a quintuple, functioning between 135 and 55°C (275 and 131°F). However, the difference would be in the opposite direction if the pressure triple effect were compared with a normal quadruple working between 115 and 55°C (239 and 131°F), and the steam economy would be far superior in the case of the multiple effect under pressure.

This is therefore an excellent solution. But it is possible only in the case where a new factory is being built, or indeed where a concomitant alteration to the plant permits the back-pressure to be raised to the required value. In an existing factory, such a scheme would be limited generally by the impossibility of operating the steam engines or turbines with a back-pressure of 1.5–2 kg/cm² (20–30 p.s.i.), much higher than that for which they were designed.

However, if the back-pressure can be raised to a moderate extent, an intermediate solution could be adopted, for example, a quadruple partly under pressure, partly under vacuum, which would then permit a gain of one stage in the use of vapour for heating.

In practice, evaporation fully under pressure is never used for cane sugar. The system is generally arranged so as to draw the maximum from the last vessel under pressure; this is followed by an evaporator vessel the vapour of which is at 100°C and which, in addition to the vapour bleeding for which it is provided, is arranged to leave an excess of vapour which goes to a concentrator working under vacuum. Thus we have an evaporation which copes more readily with variations in juice supply to the boiling house.

With regard to steam consumption, a German report⁷⁸ estimates that evaporation under pressure permits of reducing the steam consumption to 321 kg/t.c. (709 lb./t.c.), with an economy of bagasse amounting to 46%. It is considered that the first vessel must be of falling-film type, where the juice passes through very quickly; otherwise there would be marked decomposition of sucrose, due to the temperature of 125°C (255°F) required. In the beet sugar industry there is no such problem, since beet juice is thermo-stable even at 125–130°C (255–265°F).

object of some interest in the cane sugar factory. They are a class of evaporator vessels in which the juice travels from top to bottom (Fig. 32.54, 32.55), as distinct from the general evaporators where the juice moves upwards, sometimes known as climbing-film evaporators. When the juice is fed at the top of the tubes, it descends in the tubes entraining with it the vapour produced, and collects in a lower chamber where the vapour separates and from which it is evacuated to the following vessel or the condenser.

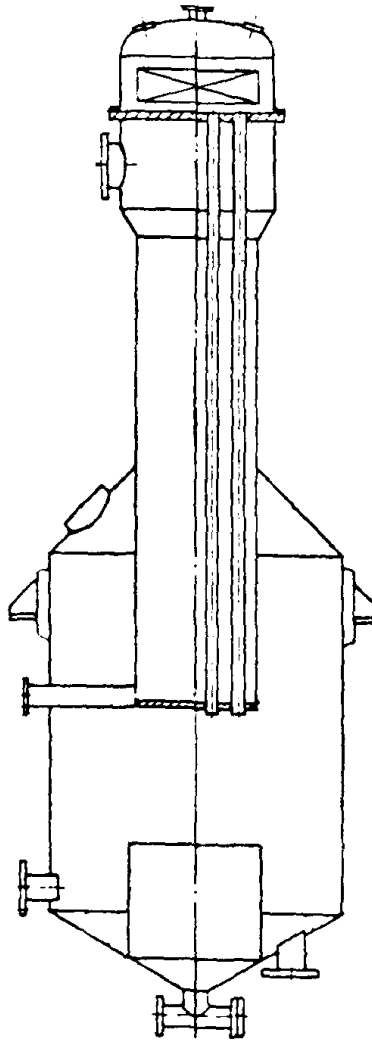


Fig. 32.54. Falling film evaporator (B.M.A.).

Such evaporators show several advantages. They have a good heat transfer, since there is no boiling-point elevation due to hydrostatic pressure, as the juice is not subjected to such pressure. There is no expenditure of energy to produce upward movement of the juice; this is indeed very small, but is produced at the expense of heat exchange. The evaporators are designed so that the juice is in contact with the heating surface in a thin layer over the length of the heating surface, hence in intimate contact, and the vapour, being unable to leave from the upper portion, is entrained with the juice and fills the interior of the tube; hence there are no vapour bubbles to obstruct the upward flow of juice, as in the climbing-film evaporator.

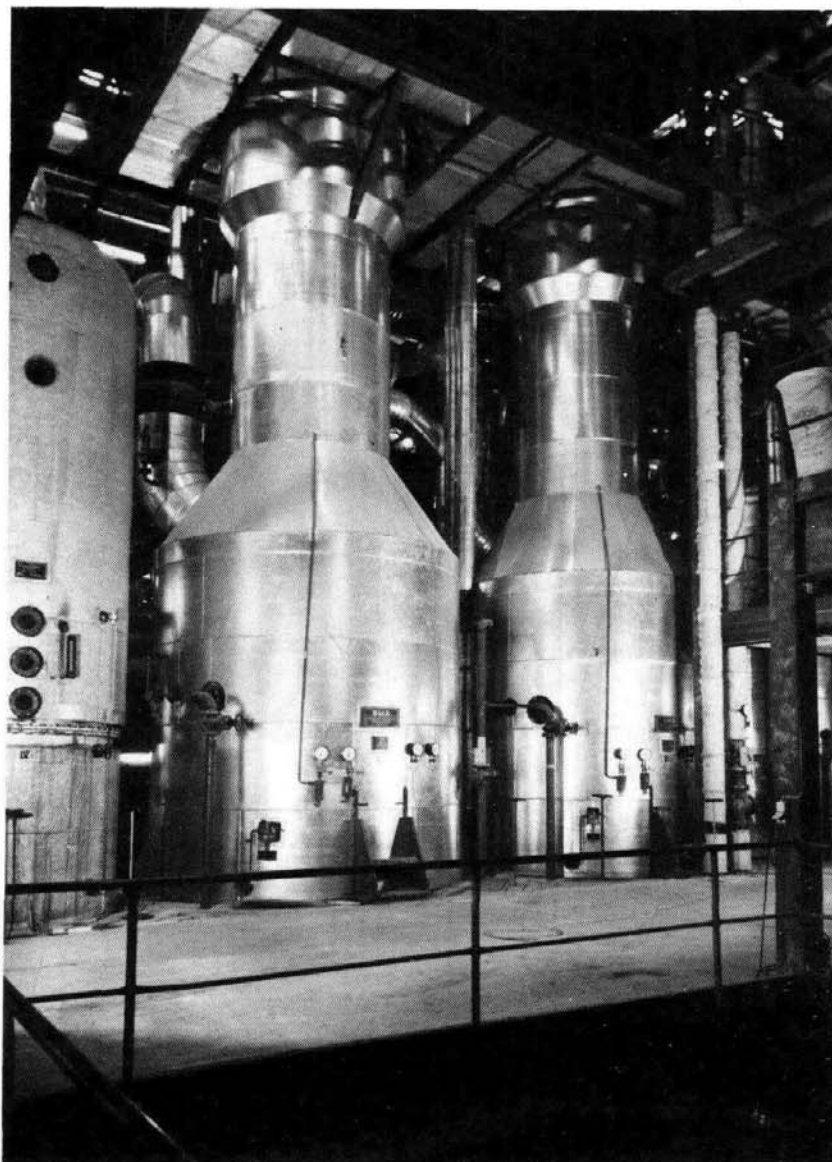


Fig. 32.55. Falling film evaporator (B.M.A.).

We describe here the Fives Cail—Babcock falling-film evaporator vessel, patented in 1979 by André Longuet (Fig. 32.56). It is built with very long tubes (7–8 m (23–26 ft.) at least, and normally 10 m (32 ft.)). It follows that for a given heating surface, the evaporator vessel is of small diameter (*e.g.* 3 m for 2,440 m²). While the lower vessel receiving the juice is obviously larger (*e.g.* 3.8 m), this reduces the area required in the horizontal plane, naturally at the expense of height required.

The main problem, with a descending-film evaporator, is the distribution of juice on the upper tube plate. It is necessary that all tubes from this plate, or from one section of the tube plate, should receive the required volume of juice so that the juice is distributed only in a thin layer around the tube, and that no section of tube remains dry at any moment.

With this object, the total area of the tube plate is divided into enough sections to assure

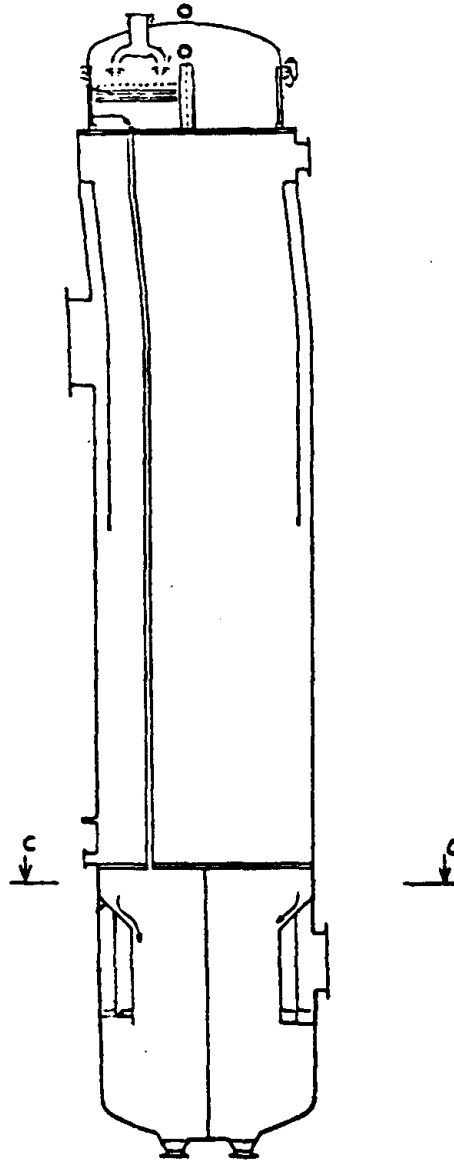


Fig. 32.56. Falling film evaporator license Longuet (Fives Cail – Babcock).

the optimum wetting of the tubes: 1, 2 or 3 sections, according to the size of the vessel. The figure m , expressing the distribution or wetting of the tubes by the juice, is calculated as $Q/n\pi d$, where Q is the weight of juice in kg/h (lb./h), n the number of tubes, and d their diameter. The optimum figure sought varies from 500 to 600 kg/h/m (335 – 400 lb./ft./h) of circumference of tubes, to a maximum of 1,200 – 1,500 (800 – 1,000).

The tubes are in stainless steel, 45.1 x 48.3 mm, thus 1.6 mm ($\frac{1}{16}$ in.) in thickness. The juice arrives at each section by a pipe which divides into several distributors (Fig. 32.57). Each of these delivers onto a set of conical deflectors of decreasing diameter, with a hole in the centre, also of decreasing diameter. The tubes are arranged in staggered spacing, and the juice is delivered in succession onto three rows of bars (split tubes, or angle-pieces with the angle upwards) set parallel in 3 rows one above the other at angles of 120° , each being situated above a row of tubes (Fig. 32.58, 32.59). The final row of these bars is just above

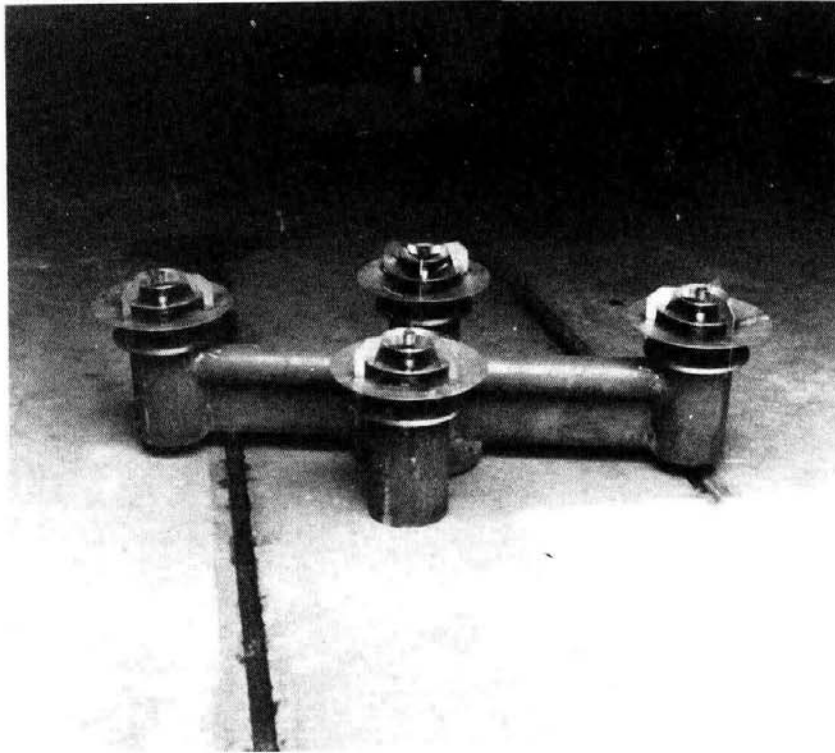


Fig. 32.57. Distributor for falling film evaporator (Fives Cal - Babcock)

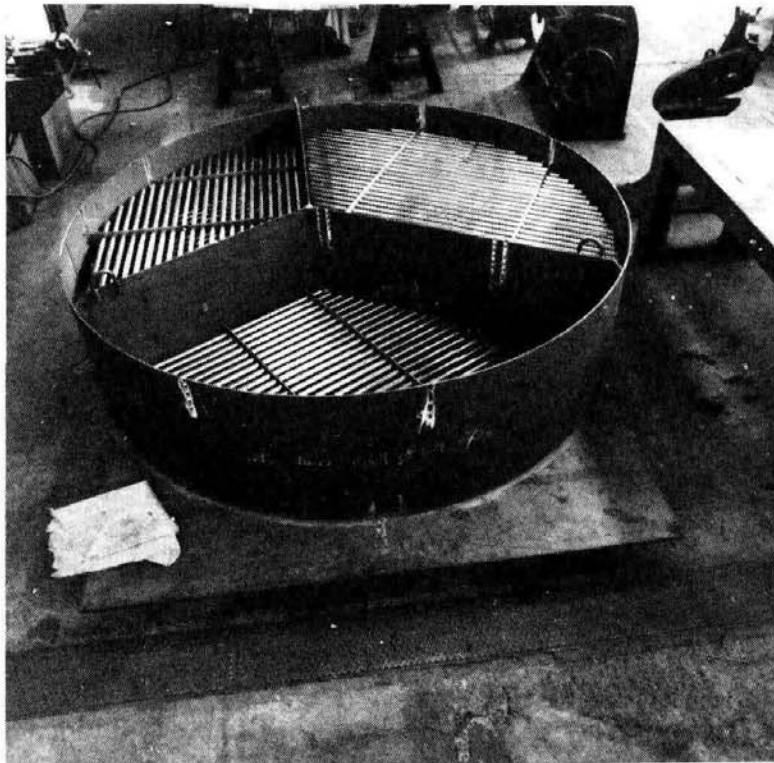


Fig. 32.58. Three-sector distributor for falling film evaporator. First layer of bars.

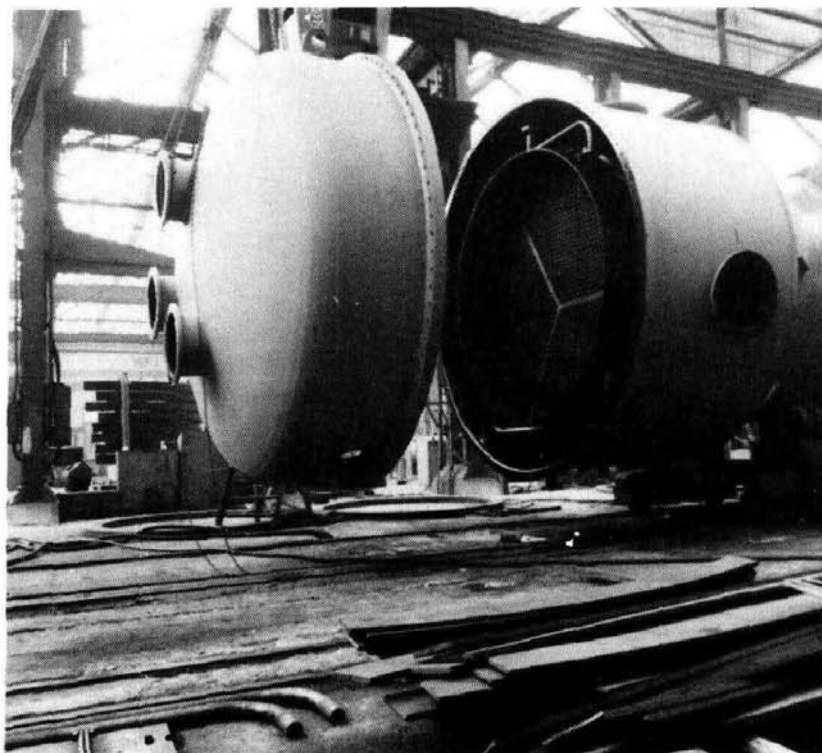


Fig. 32.59. Falling film vessel showing the three-sector distributor (Fives Cail – Babcock)

the tube plate, and the juice falls onto the tube plate in the inter-tube spaces. The juice is prevented from falling directly into the tube by conical or convex caps located above each tube as an umbrella. Hence the juice can enter the tube only by its periphery, thus forming a film on the tube surface. The cap is supported by a plastic piece in the shape of a 3-point star filling the interior tube diameter and notched to support it on the plate; its three vertical wings, a few centimetres in length, have a bevelled edge at the lower end, shaped so that the juice flows towards the wall of the tube. These "valve-caps" are held in place by the third row of bars above them.

At the outlet from the tube, the juice falls to the bottom of the vessel, and the vapour leaves by an opening below the calandria, after traversing an entrainment separator similar to that of Fig. 32.29, which it traverses in series.

The tubes are in stainless steel SF. 17 (Inox 430), which has the same coefficient of expansion as the ordinary steel of the vessel surrounding the calandria; an indispensable condition, on account of the length of the tubes. For this same reason, these falling-film evaporators are located out in the open, as removal of the 10-m tubes is effected from the top.

The heating steam enters in the upper one-third of the calandria; at this position, the bundle of tubes is surrounded by a sleeve which protects it from the impact of the steam, and permits a better distribution of steam (Figs. 32.56 and 32.60). Incondensable gases are evacuated at the top and bottom of the calandria.

This model makes no provision for recycling juice, which would increase the time of the juice in the vessel and is not required where the juice adequately wets the tube surface. For recycling, the calandria must be divided into two or three sections, particularly in the later

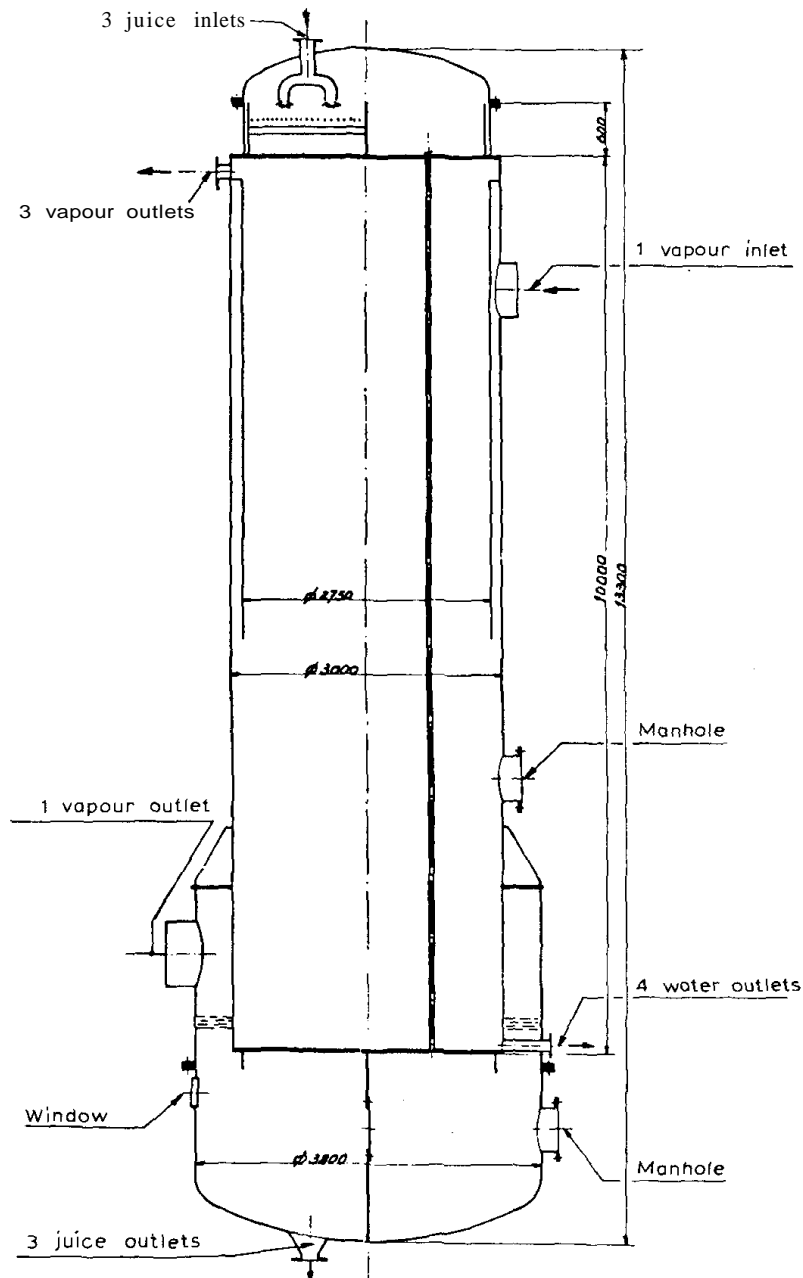


Fig. 32.60. Falling film evaporator with enlarged base (Fives Cail–Babcock).

vessels. Some equipment manufacturers provide for recycling by enlarging the lower portion of the evaporator shell below the calandria (Figs. 32.60 and 32.61), thus providing more space for disengagement of the vapour. In this case, a mesh eliminator can be provided in the path of the vapour. It is then necessary to clean the eliminator at least once per shift, by a suitable device actuated by an automatic timer (Fig. 32.61). This portion of the vessel is illuminated by a lamp at a lateral sight glass, to permit of inspection.

The coefficient of heat transfer is superior to that of climbing-film evaporators; hence the falling-film evaporator can work under an appreciably lower temperature drop. It is thus indicated for use in conjunction with a turbo-compressor (Fig. 32.61).

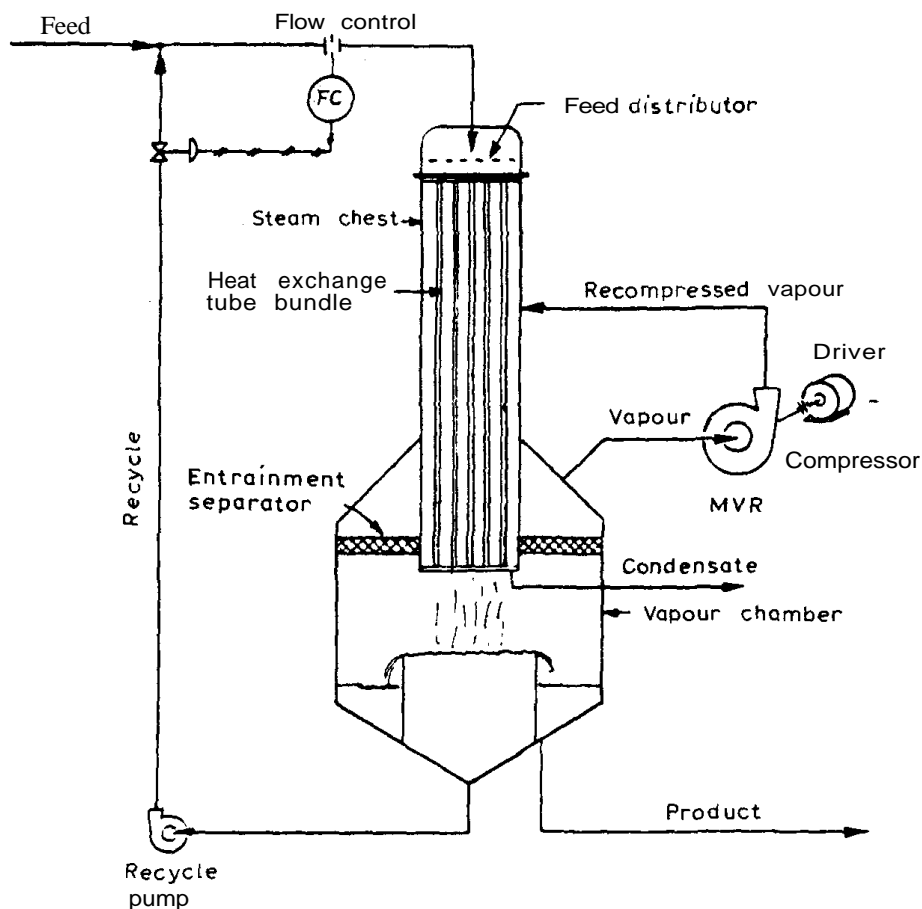


Fig. 32.61. Falling film vessel with mattress separator and turbo-compressor (By courtesy of the Int. Sugar Journal).

Use of falling-film evaporation has not expanded rapidly; the first installations, in Italy, gave some trouble due to uneven distribution of juice. However, they are now operating efficiently, with due observance of the precautions described, and are adopted particularly where a high degree of steam economy is required. They are installed in single evaporators or multiple sets (quintuple effects, for example), and give every satisfaction. In the beet sugar industry they have been installed heated by vapour from vacuum pans. A complete multiple-effect evaporator of this type is unstable in the case of abrupt variations in flow of juice, with frequent drying of the tubes, on account of the low reserve of volume. They must be arranged to operate at a steady rate of juice flow.

On account of the very brief period of juice contact, these evaporators may be operated with steam at 135°C (275°F), or 2.1 kg/cm^2 (30 p.s.i.) pressure. This is their principal advantage.

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33. *Sugar Boiling*

When juice is concentrated, its viscosity rises rapidly with brix and, when the latter reaches 78–80°, crystals begin to appear and the nature of the material changes: it passes progressively from the liquid state to a part-solid, part-liquid condition. It loses its fluidity more and more, and completely different methods are required for handling it. It is then called “massecuite”.

Its consistency no longer permits of boiling it in narrow tubes or of circulating it easily from one vessel to another.

For this reason, a change is made to:

(1) Evaporation in single effect

(2) A type of equipment similar in principle but better adapted in detail to the viscous product which it is to concentrate.

Brix of syrup

We have already seen (p. 503) that the optimum concentration of syrup at which to make the change from multiple effect to vacuum pans is approximately 60 - 70° brix. If the concentration is taken beyond this figure, steam economy is improved because a little more is evaporated in multiple and a little less in single effect; but it becomes more difficult for the sugar boiler to obtain a regular grain.

In the manufacture of white sugar, the syrup is treated between the evaporators and the pans, either by decantation or by filtration. In this case, one must be content with a syrup of 50–55° brix; above this value, the rate of settling of the muds decreases rapidly, as does the rate of filtration.

Brix of massecuite

The brix of the massecuite depends on the substances dissolved in the mother liquor, plus the sugar present in the form of crystals.

Analysis of the massecuite is made by dissolving a certain weight of the massecuite in a known volume of water. The sugar percentage, the purity and the brix are determined on the dilute liquid so obtained. The sugar in crystal form is therefore included, as well as the sugar in solution in the mother liquor.

Limit of concentration obtainable

The concentration of the massecuite is pushed as far as possible. With modern pans and in a well-controlled factory a refractometric brix of 98° (perhaps approximately 94% of actual dry substance) may be obtained and even slightly exceeded. In factories not so well equipped, 96° brix can hardly be exceeded in the heaviest massecuites.

Starting with 1,000 kg of juice at 13° brix and concentrating by multiple effect to 65° brix,

the quantity of water originally present in the juice and evaporated by multiple effect becomes (eqn. (32.9)):

$$E = 1,000 \left(1 - \frac{65}{65} \right) = 800 \text{ kg}$$

The quantity of syrup is therefore:

$$S = 1,000 - 800 = 200 \text{ kg}$$

and the fraction of the water originally present in the juice and evaporated in the vacuum pan thus becomes, if the boiling is carried to 96° brix:

$$E = 200 \left(1 - \frac{65}{96} \right) = 65 \text{ kg}$$

Contraction

Actually a solution of sugar occupies a smaller volume than the combined volumes of the water and sugar. This contraction has a maximum value at a concentration given by Tromp¹ as 56% and by Parr² as 57.3%, and is of the order of 1% at that concentration. It is, therefore, very slight and we shall neglect it in our calculations.

Specific heat of the massecuite

Equation (30.1) gives the specific heat of syrups and molasses with an approximation sufficient for practical purposes. But the massecuite carries with it also crystals of sugar which modify the total specific heat. Roughly we can reckon for the massecuite:

$$c = 1 - 0.007 B \quad (33.1)$$

c = specific heat of massecuite

B = brix of massecuite.

This formula gives a suitable value only when the massecuite is heavy. Otherwise, it is necessary to take a value intermediate between those given by the two formulae (30.1) and (33.1), or to adopt:

$$c = 1 - 0.001 \left(6 + \frac{\hat{r}}{60} \right) B \quad (33.2)$$

\hat{r} = crystal % massecuite by weight.

Saturation

The solubility of pure sucrose in water varies with temperature and increases rapidly with increase of temperature. At 40°C, for example, 1 kg of water is capable of dissolving 2.334 kg of sucrose; at 80°C, 3.703 kg.

When a solution contains the total quantity of sucrose which it can dissolve, it is said to be saturated.

Juice. However, in practice we always have to deal with impure materials, *i.e.* with solutions containing in solution, not only sucrose but other materials accompanying it; glucose, organic substances, mineral salts, etc.

These substances modify the solubility of sucrose. In beet juice, impurities increase the solubility; a given quantity of water dissolves more sucrose in the presence of impurities than it does when such substances are absent. The opposite holds with cane products; here the solubility of sucrose decreases with decreasing purity.

The effect varies with the particular impurities in each case. With cane, it is mainly the reducing sugars which cause the decrease in the solubility of sucrose.

TABLE 33.1

SATURATED CANE SYRUP, DISSOLVED SOLIDS % SOLUTION BY WEIGHT FOR VARIOUS TEMPERATURES AND PURITIES

Purity	Temperature									
	62 °C (144 °F)	64 °C (147 °F)	66 °C (151 °F)	68 °C (154 °F)	70 °C (158 °F)	72 °C (162 °F)	74 °C (165 °F)	76 °C (169 °F)	78 °C (172 °F)	80 °C (176 °F)
92	75.6	76	76.4	76.8	77.2	77.6	78	78.4	78.9	79.3
90	75.9	76.3	76.6	77.1	77.5	77.9	78.3	78.7	79.1	79.5
88	76.2	76.5	77	77.3	77.7	78.1	78.5	78.9	79.3	79.7
86	76.5	76.8	77.2	77.6	78	78.4	78.8	79.2	79.6	80
84	76.7	77.1	77.5	77.9	78.3	78.7	79.1	79.5	79.9	80.3
82	77	77.4	77.8	78.2	78.6	79	79.4	79.7	80.1	80.5
80	77.3	77.7	78.1	78.5	78.9	79.2	79.6	80	80.4	80.8
78	77.6	78	78.4	78.8	79.2	79.5	79.9	80.3	80.7	81.1
76	78	78.4	78.7	79.1	79.5	79.8	80.2	80.6	81	81.4
74	78.3	78.7	79.1	79.4	79.8	80.2	80.5	80.9	81.3	81.7
72	78.7	79	79.4	79.8	80.1	80.5	80.9	81.2	81.6	82
70	79	79.4	79.7	80.1	80.5	80.8	81.2	81.6	81.9	82.3
68	79.3	79.7	80.1	80.4	80.8	81.1	81.5	81.9	82.2	82.6
66	79.7	80.1	80.4	80.8	81.1	81.5	81.8	82.2	82.6	82.9
64	80.1	80.4	80.8	81.1	81.4	81.8	82.2	82.5	82.9	83.2
62	80.4	80.8	81.1	81.5	81.7	82.1	82.5	82.8	83.2	83.5
60	80.8	81.1	81.4	81.8	82.1	82.4	82.8	83.1	83.5	83.8
58	81.1	81.4	81.7	82.1	82.4	82.7	83.1	83.4	83.8	84.1
56	81.4	81.7	82	82.4	82.7	83	83.4	83.7	84.1	84.4
54	81.7	82.1	82.4	82.8	83	83.3	83.7	84	84.4	84.7
52	82.1	82.4	82.7	83	83.3	83.7	84	84.3	84.7	85
50	82.4	82.7	83	83.4	83.7	84	84.3	84.6	85	85.3
48	82.8	83.1	83.4	83.7	84.1	84.3	84.6	85	85.3	85.6
46	83.2	83.5	83.8	84.1	84.4	84.7	85	85.3	85.6	85.9
44	83.5	83.8	84.1	84.4	84.7	85	85.3	85.6	85.9	86.2
42	83.9	84.2	84.5	84.8	85.1	85.4	85.6	85.9	86.2	86.5

Table arranged according to Thieme.

In practice, the brix is substituted for the true dissolved solids above, though the difference is not negligible.

The *solubility coefficient** (s) is the ratio of the quantity of sucrose soluble in a given weight of water in an impure solution, at a certain temperature, to the quantity soluble in pure water at the same temperature:

$$s = \frac{\text{Solubility of sucrose \% water in impure solution}}{\text{Solubility of sucrose \% water in pure solution}} \quad (33.3)$$

The solubility coefficient of cane products increases with temperature, but this effect is relatively small. A graph showing the variation of solubility coefficient with purity, in cane products, is given in Fig. 33.1. This plot is taken from the work of Thieme in Java, who directs attention to the very approximate nature of the graph given; its object is only to give an idea of the relationship, the solubility coefficient being variable and, moreover, difficult to determine with precision.

Table 33.1 gives the weight of material dissolved in a saturated solution of juice, syrup or molasses obtained from cane, as a function of the temperature and purity of the solution. This table is also taken from the work of Thieme in Java.

It will be noted that the solubility lies in the neighbourhood of 80%, for typical temperatures and purities.

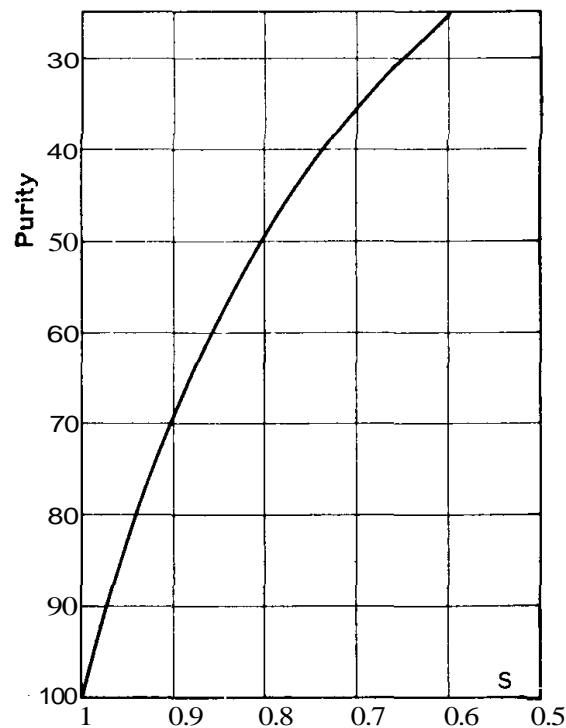


Fig. 33.1. Graph of solubility coefficient.

* This ratio was previously termed the saturation coefficient; but "solubility coefficient" is preferred as being more accurate as well as less liable to confusion with "supersaturation coefficient".

Supersaturation

Saturation is a state of stable equilibrium which for sugar solutions is reached neither rapidly nor easily. If a solution is concentrated by evaporation, or if it is cooled beyond the saturation point, the crystals do not appear immediately (or necessarily) in the material. The sugar still remains in solution and the solution is termed supersaturated.

The *supersaturation coefficient* (S) is the ratio of the weight of sucrose per cent water contained in a supersaturated solution to the weight of sucrose per cent water which would be present in a saturated solution having the same temperature and the same purity:

$$S = \frac{\text{Sucrose \% water in supersaturated solution}}{\text{Sucrose \% water in saturated solution}} \quad (33.4)$$

By the use of Table 33.1, the weight of material dissolved in a supersaturated solution is readily calculated. Let:

B' = weight of material dissolved per cent of supersaturated solution

B = weight of material dissolved in saturated solution of the same purity and at the same temperature

S = supersaturation coefficient.

We have then (since the purities are the same):

$$\frac{\frac{B'}{100 - B'}}{\frac{B}{100 - B}} = S$$

whence:

$$B' = \frac{100 BS}{100 + B(S - 1)} \quad (33.5)$$

Crystallisation

In commercial operation it is essential, in order that crystals should form in a sugar liquor, that there should be a considerable supersaturation. In proportion as the crystals form and grow, the supersaturation of the mother liquor drops. To maintain the supersaturation, therefore, it is necessary to maintain evaporation of water and supply of further sugar-bearing material.

Weber distinguishes three zones in the supersaturated phase:

(1) The *metastable* zone, being that nearest saturation; existing crystals increase in size but new crystals cannot form

(2) The *intermediate* zone; new crystals can form in this zone, but only in the presence of existing crystals

(3) The *labile* zone; in this zone existing crystals grow, and new crystals can form even in the absence of existing crystals.

Davies and Yearwood³ in Trinidad have shown that:

(a) The intermediate zone is very narrow, if indeed it exists at all, and has little practical importance. Ziegler^{3a} also queries the existence of the intermediate zone, and considers the limit between metastable and labile zones as 1.65.

(b) The boundary between the two principal zones varies markedly with the purity. It passes close to the following points of supersaturation (see also Fig. 33.2):

Purity	Supersaturation at boundary between metastable and labile zones
60	1.55
70	1.30
80	1.25

In the course of a strike it is desirable to maintain the mother liquor as near as possible to the upper limit of the metastable zone. At a purity of 60, approximately 23.5% more sugar is deposited in one hour at a supersaturation of 1.55 than at a figure of 1.45⁴.

(c) The boundary between the two zones is much less dependent on temperature. The supersaturation coefficient marking this boundary falls slightly as the temperature increases, but the effect appears negligible.

These considerations, however, are of theoretical importance only. In practice, it has been established that the supersaturation at which sugar commences to crystallise varies between 1.10 and 1.50; 1.20 is regarded as a normal figure but cases have been reported where crystallisation had commenced at 1.07 (Thieme⁵). At other times in pure solutions, crystals did not form before a supersaturation of 1.50 was reached.

The reasons for this variation are not well known, but Thieme considers that it is due to certain characteristics of each juice: viscosity (which retards crystallisation), presence of insoluble matter in suspension (which facilitates crystallisation); another probable cause is leakage of air into the pans, by which particles of sugar dust are introduced which stimulate the formation of crystals, etc.

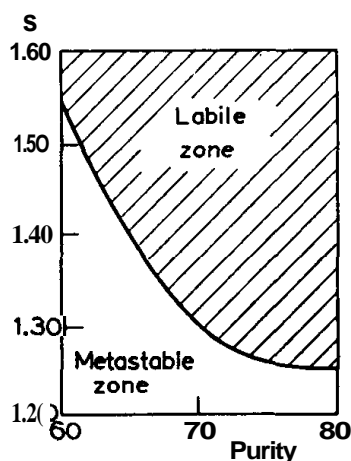


Fig. 33.2. Metastable and labile zones of crystallisation.

Generally, the supersaturation is maintained between 1.25 and 1.40⁶.

Rate of crystallisation. The rate of crystallisation of a massecuite depends on:

- (a) The viscosity
- (b) The temperature (which influences the viscosity)
- (c) The supersaturation
- (d) The purity of the mother liquor.

Temperature. When the temperature falls, in order to maintain the same rate of crystallisation, it would be necessary to increase the supersaturation; e.g. Saint⁷ gives in Table 33.2 temperatures and supersaturation coefficients corresponding to the same rates of crystallisation for C massecuite:

TABLE 33.2

EXAMPLE OF VARIATION OF SUPERSATURATION WITH TEMPERATURE TO MAINTAIN THE SAME RATE OF CRYSTALLISATION

Temperature		Supersaturation
(°C)	(°F)	
70	158	1.25
60	140	1.30
50	122	1.35
40	104	1.40

Supersaturation. It has been established that the rate of absorption of sucrose by the crystals is proportional to the square of the supersaturation⁸; but in practice the supersaturation must not exceed the critical value (stated as 1.44) above which the crystallisation would become out of control, giving abundant false grain.

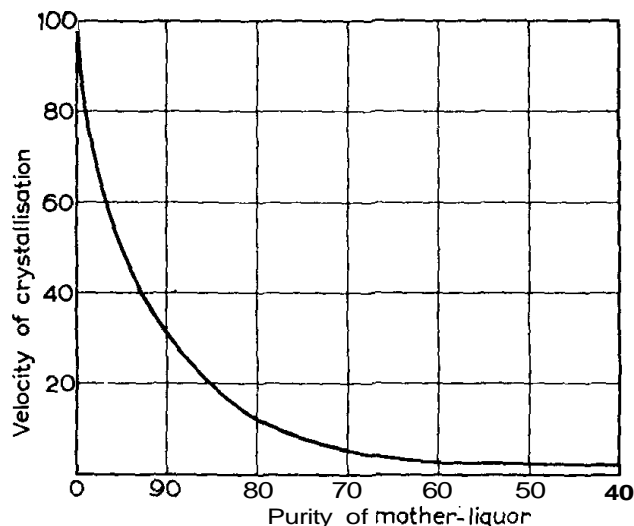


Fig. 33.3. Variation of speed of crystallisation with purity of mother liquor.

Purity. The rate of crystallisation falls very rapidly as the purity of the mother liquor decreases. This is why a low-grade strike requires more time than a first massecuite.

Webre⁹ gives a graph relating rate of crystallisation and purity (Fig. 33.3).

Elevation of boiling point

This question has already been discussed with reference to evaporation (p. 496): Table 32.2 is of course also valid for sugar boiling.

Hydrostatic pressure

Table 32.3 is again applicable here, and for this purpose is of interest mainly in the region of the lines *MC* corresponding to the density (approximately 1.47) of hot massecuite at 80-90° brix.

In sugar boiling, hydrostatic pressure assumes great importance, and its effects will be discussed later.

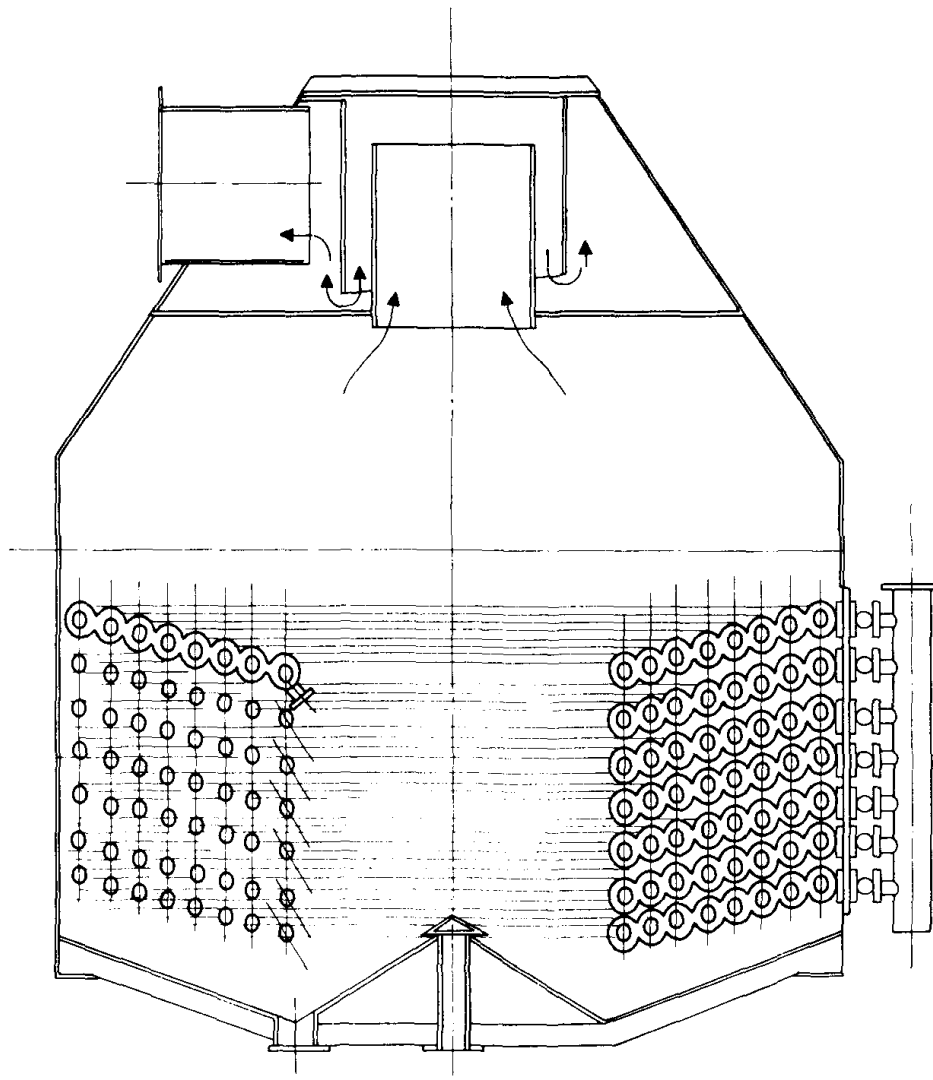


Fig. 33.4. Coil vacuum pan.

VACUUM PANS

The conception of the vacuum pan is entirely analogous to that of an evaporator body. However, two modes of heating are employed:

- (a) Coils (Fig. 33.4)
- (b) Calandria (Fig. 33.5)

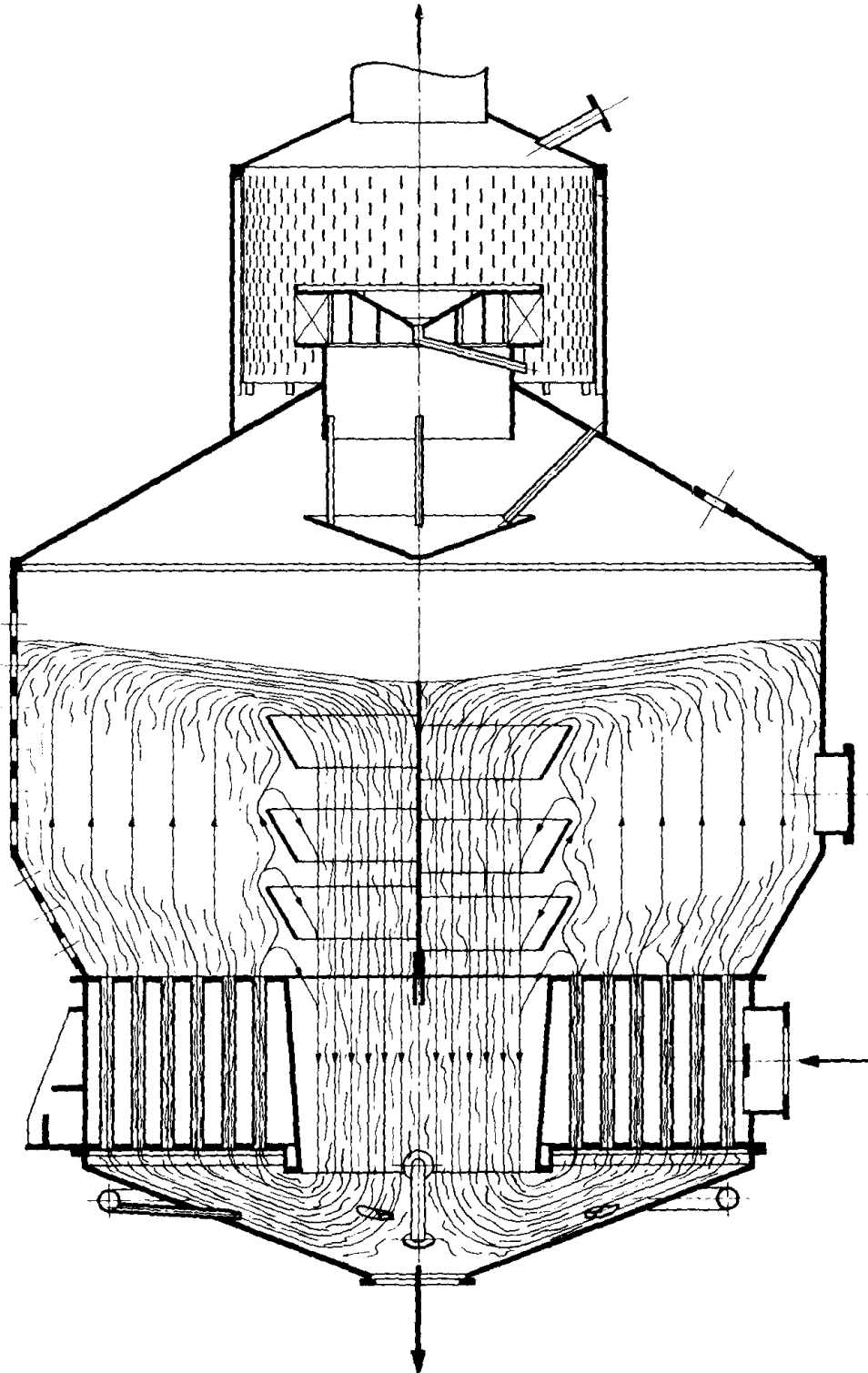


Fig. 33.5. Calandria pan for low-grade massecuite, H.I.W. type.

the latter being broadly similar to that used in multiple-effect evaporators. In some cases, both calandria and coils are used in the one pan.

The removal of condensates and incondensable gases is effected as in evaporators.

(A) Coil pans

A pan may have one, two or more coils. These are held in place by clips and brackets.

Coils of circular section. In the older standard type of pan, the coil consisted of a copper tube commencing at the outer wall of the pan, bent in the form of a spiral around a truncated cone, and terminating near the centre of the pan in a drain for the condensate.

In Queensland, it has been emphasised¹⁰ that the angle of dish, or the angle formed with the horizontal by the generatrices of this truncated cone, should be approximately 45°. French manufacturers are content with a much smaller angle; in this case it is necessary to ensure that the slope of the coil, from beginning to end, is sufficient to ensure effective drainage of the condensate. For this purpose the angle of dish is made no less than 15°. (Half-angle of cone 75°.)

The vertical pitch or spacing between two successive coils should be of the order of 25 – 30 cm (10- 12 in.) between centres. It will be arranged so that the turns of the coils are approximately vertically one above the other, or only slightly displaced from the vertical, so that they do not provide any obstruction to the circulation of the massecuite; this should be in the upward direction in the region of the coils.

The coils are always of copper. Their diameter is about 10 – 15 cm (4-6 in.), but sometimes varies, decreasing from beginning to end of the coil.

The ratio of length to internal diameter varies generally from 75 to 250. This ratio depends principally on the steam used for heating. Short coils are most advantageous, particularly for low-pressure vapours.

Tromp¹¹ recommends that ratios of 200—250 be reserved for high-pressure steam (with low-pressure steam, the later part of the coil would not transmit any heat and so would be useless).

For exhaust steam it is preferable to keep below 100.

For bled vapour, a ratio of 75 is suitable.

It is of advantage to construct the coils of heavy copper, as they wear fairly rapidly. The thickness should be between 2.5 and 3.2 mm ($\frac{3}{32}$ in. and 5 in.).

Coils with flattened section. An improved design introduced by Halpin¹² in Queensland in 1950 permitted the coil pan to hold its own in that country with the calandria pan for a long time.

The heating surface is formed from copper tubes of 12.5 cm (5 in.) diameter flattened in the vertical direction so as to present in external dimensions a width of 7.5 cm (3 in.) and a height of 15.5 cm ($6\frac{1}{8}$ in.), or alternatively a width of 10 cm (4 in.) and a height of 14 cm ($5\frac{9}{16}$ in.) (Fig. 33.4).

Instead of being formed of a single coil wound in a spiral, each coil consists of a series of flat tubes, semicircular and concentric, passing from a steam manifold which follows a generatrix of the truncated cone to a condensate manifold placed on the opposite generatrix.

With this arrangement, the angle of dish of the coil is not of any great importance. It may, for example, be made 15° , in order to facilitate flow of condensate in its manifold, and it is arranged simply that the latter is slightly lower than the steam manifold.

The centre well formed by the superposition of the inner circle of these several coils should have a diameter equal to 38 or 40% of the interior diameter of the pan.

The vertical pitch or spacing between two successive coils should be at least 30 cm (12 in.) from centre to centre of corresponding tubes, and should preferably be 33 cm (13 in.).

The clearance or free space between successive turns of the one coil should be 7.5 - 10 cm (3-4 in.), preferably 8.5 - 9 cm ($3\frac{3}{8}$ - $3\frac{1}{2}$ in.). The top coil is located in such a position that the upper tube is not less than 40 cm (16 in.) below the maximum level proposed for the massecuite.

It is desirable that the cylindrical body of the pan which carries the coils should have a height equal to 50 - 60% of its interior diameter.

When these precautions are followed, a pan may be obtained with a heating surface of $4.5 - 5.25 \text{ m}^2/\text{m}^3$ (1.4 - 1.6 sq.ft./cu.ft.) of working volume, or say approximately $5 \text{ m}^2/\text{m}^3$ (1.5 sq.ft./cu.ft.), definitely higher than that of the older type of coil pan.

Coil pans are no longer made, even in Australia.

(B) Calandria pans

In calandria pans, the tubes are shorter and of larger diameter than in evaporators.

Whether the pan is of coil or calandria type, the massecuite rises past the heating surface, between the turns of the coils or through the tubes of the calandria. There must obviously be an equal descending mass corresponding to this ascending massecuite. With the liquid and mobile material in the multiple effects, this poses no problem. With thick and viscous massecuite, the manner in which this circulation is planned and successfully effected form

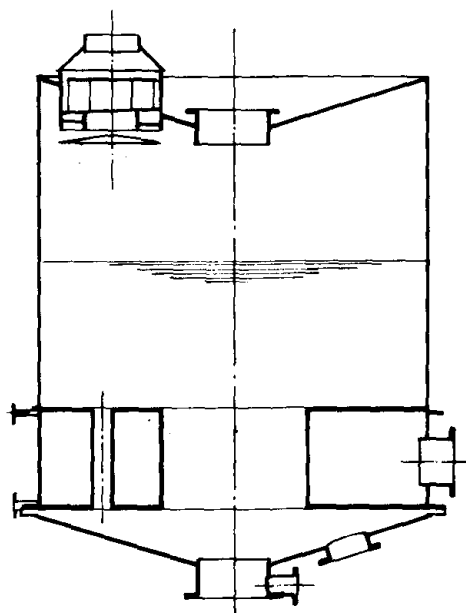


Fig. 33.6. Fives Cail-Babcock pan, type C.T.

the most important characteristic of a vacuum pan. The manner of solving this problem has led to the existence of several types of pan:

- (a) The flat fixed calandria, with central downtake (Fig. 33.5)
- (b) The inclined-plate calandria (Fig. 33.6)
- (c) The suspended ("floating" or "basket") calandria (Fig. 33.8)
- (d) The horizontal pan (Fig. 33.14).

(a) **Flat fixed calandria.** This is the recognised type. The circulation follows the scheme indicated (Fig. 33.9). We shall study later the very important conditions required for good realisation of this circulation.

The Fives Cail - Babcock series of pans (Fig. 33.6) covers a range of 120 - 200 hl (425 - 700 cu.ft.) in steps of 20 hl, and 200 - 600 hl (700 - 2120 cu.ft.) in steps of 50 hl. All are equipped with flat calandrias, with tubes 860, 1,000 or 1,140 mm (34, 39 or 45 in.) in length. Tubes are of stainless steel, S430, 94.4 X 97.4 mm diameter. The ratio of heating surface to volume in all models is 6, 7 or 8 m²/m³ (1.8, 2.1 or 2.4 sq.ft./cu.ft.) respectively for the tube lengths of 860, 1,000 and 1,140 mm.

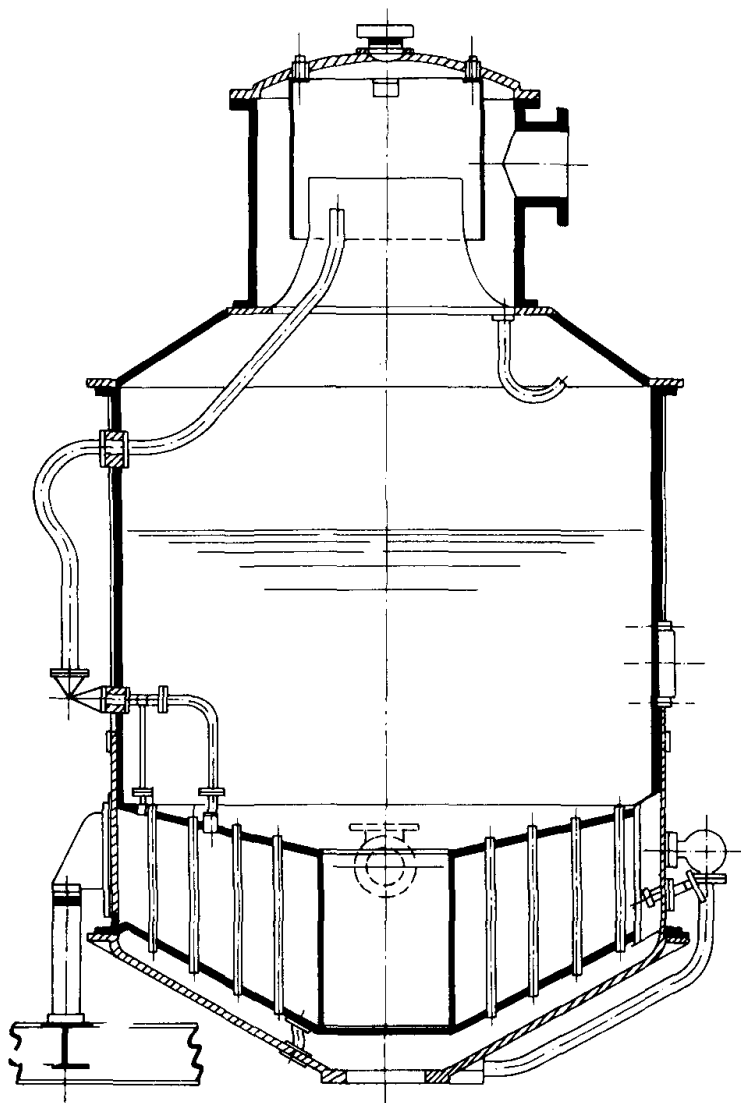


Fig. 33.7. Fixed-calandria pan with inclined tube-plates.

(b) **Inclined-plate calandria.** This may be fixed or suspended, and the plates may both be inclined at the same angle, or the lower plate may be placed at a steeper angle than the upper. French engineers generally adopt 25° in the former case and $10\text{--}25^\circ$ in the latter (Fig. 33.7). The object of this arrangement for the upper plate is to facilitate the washing and removal of the massecuite remaining on the plates after each charge; for the lower plate it aims to compensate for the loss in heating surface and calandria volume due to the dish of the upper surface, without increasing the volume required for graining. At the same time it enables the calandria bottom to follow the form of the bottom of the pan.

This arrangement is frequently adopted in beet sugar factories. We consider, however, that the negative cone of the upper plate is not justified, as the calandria with flat plates offers little or no disadvantage from the point of view of washing (actually a secondary consideration). Moreover, this cone causes a loss of heating surface and necessitates an undesirable increase in the graining volume.

The positive cone of the lower plates, on the other hand, permits of a useful gain in the heating surface and a useful decrease in the graining volume.

(c) **Floating-calandria pan** (Fig. 33.8). This design has long been in existence, but the troublesome habit was followed of providing at the centre of the floating calandria, a centre

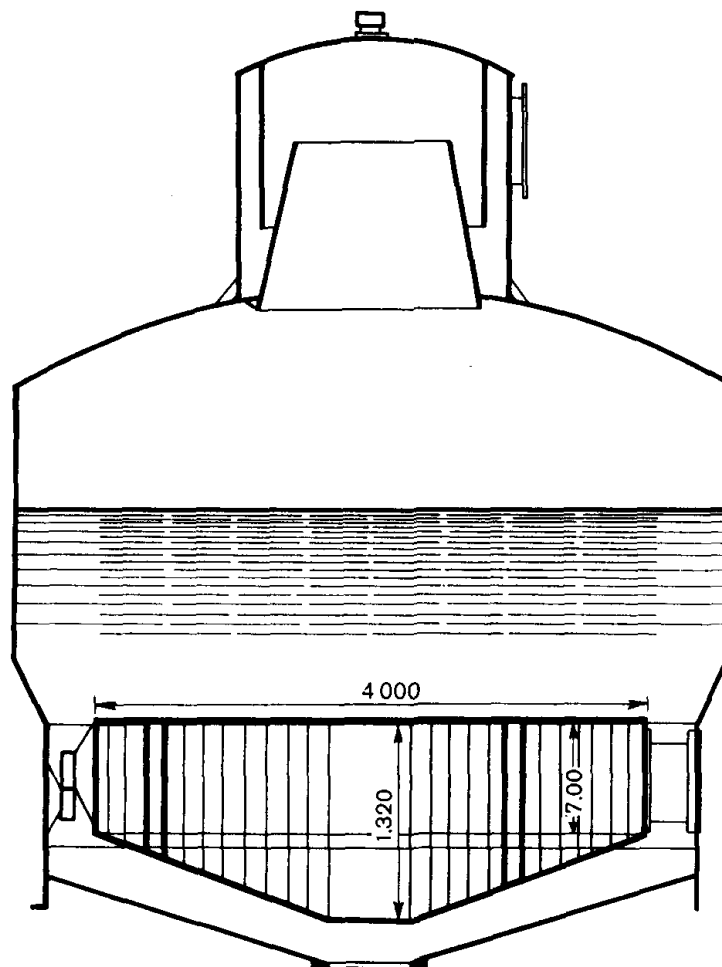


Fig. 33.8. Floating-calandria pan (Fives - Hugot).

well analogous to that of evaporator vessels; this, with the annular space, provided two passages for the descent of the massecuite. This division of the massecuite current into two portions had most unfortunate effects: while intermediate doubtful zones exist where the direction of flow is uncertain, two restricted passages are provided, neither of which offers a free circulation.

The author considered that it would on the contrary be advantageous to accentuate the natural advantages of the floating calandria and to avoid these disadvantages. He accordingly designed, with the aid of Fives Lille - Cail, the Fives - Hugot pan (Fig. 33.8), in which the heating surface at the centre of the pan was increased by adopting a calandria with a conical bottom plate and a flat upper plate. The descent of the massecuite is thus provided with an annular space of ample cross-section designed and calculated mathematically. The massecuite thus receives at the centre the maximum heat input and consequent lifting force, and descends on all sides along the cool outer wall of the pan (Figs. 33.9 and 33.10).

This arrangement lends itself particularly well to the enlargement of the pan body above the calandria (Fig. 33.11); the massecuite descends naturally in the annular space, whereas with a fixed calandria it tends to form eddies and vortices near the wall of the pan. The descent along the outer wall is a favourable factor.

To avoid restricting the circulation, entry of steam to the pan is made by a pipe entering through the save-all and descending vertically to the centre of the calandria. This arrangement moreover greatly facilitates the distribution of steam in the calandria, and the location of incondensable-gas outlets.

For the same reason, attachment of the calandria to the pan wall is effected by three very narrow brackets.

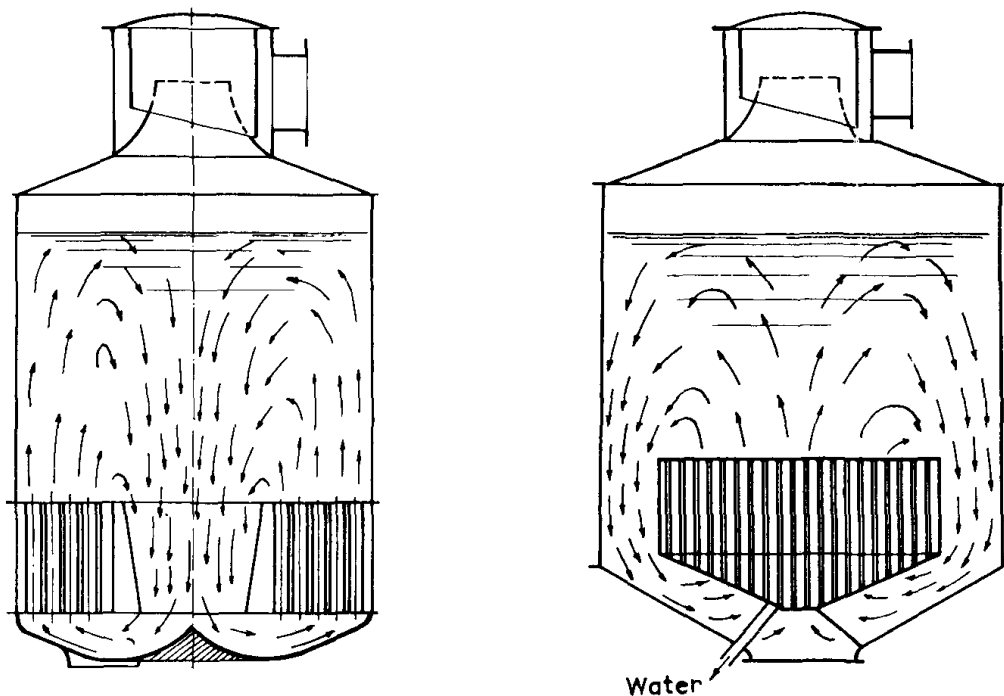


Fig. 33.9. Circulation of massecuite in a fixed-calandria pan.

Fig. 33.10. Circulation of massecuite in a floating-calandria pan.

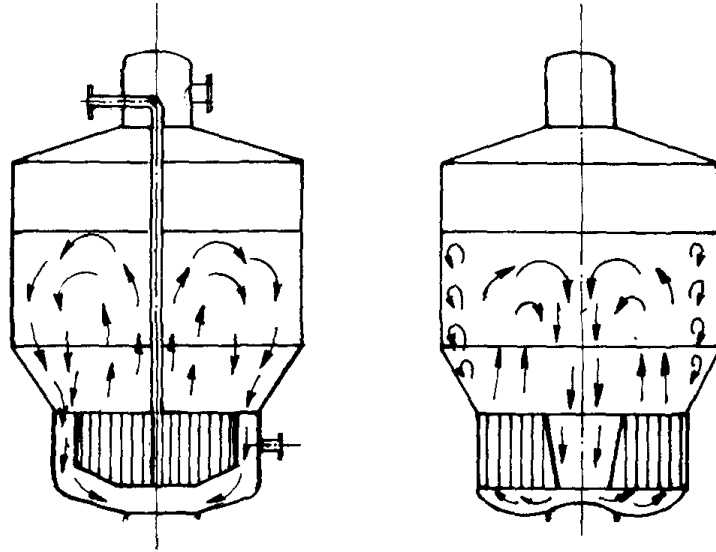


Fig. 33.11. Circulation of massecuite in pan with enlarged body; comparison of floating and fixed calandrias.

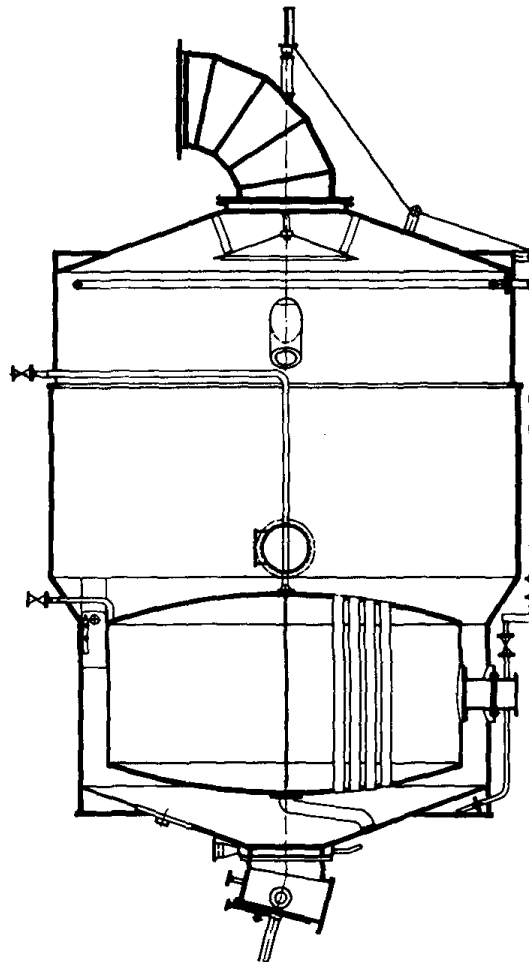


Fig. 33.12. Pan with lenticular calandria (B.M.A.).

Fives Cail-Babcock no longer make the ordinary floating calandria pan. They now supply one in which the calandria is, not cylindro-conical, but lenticular in shape (Fig. 33.12). The upper and lower tube plates of this calandria have the object of facilitating the circulation of the massecuite which, at the upper portion, flows more freely towards the outer downtake and is better directed towards the wall of the pan; below the calandria, the massecuite space follows more closely the bottom of the pan, reduces the footing volume, and gives more positive flow back to the calandria tubes. The longer tubes at the centre encourage a more active circulation in that zone, and facilitate the outward flow of the massecuite.

(d) **Horizontal pan.** Fives Lille – Cail¹² has placed on the market in 1963 a pan of different conception (Figs. 33.13 and 33.14). This pan has no longer an axis of symmetry but a plane of symmetry. Its shape allows of a low graining volume, of the order of 26%, a massecuite height above the calandria of about 1.5 m (5 ft.) and a very favourable circulation due not only to its shape but also to the arrangement of the calandria (Fig. 33.14). The ratio of descending section to ascending, at the top of the calandria, is 0.8. Discharge of massecuite is effected by three openings situated in the plane of symmetry, which is thus particularly rapid: approx. 10 min. The heating surface gives a ratio of S/V of 6-8 m^2/m^3 (1.8-2.4 sq.ft./cu.ft.).

This design of pan should be located with its axis at right-angles to the alignment of the other pans, as it would take up too much space in the other direction, on account of the space

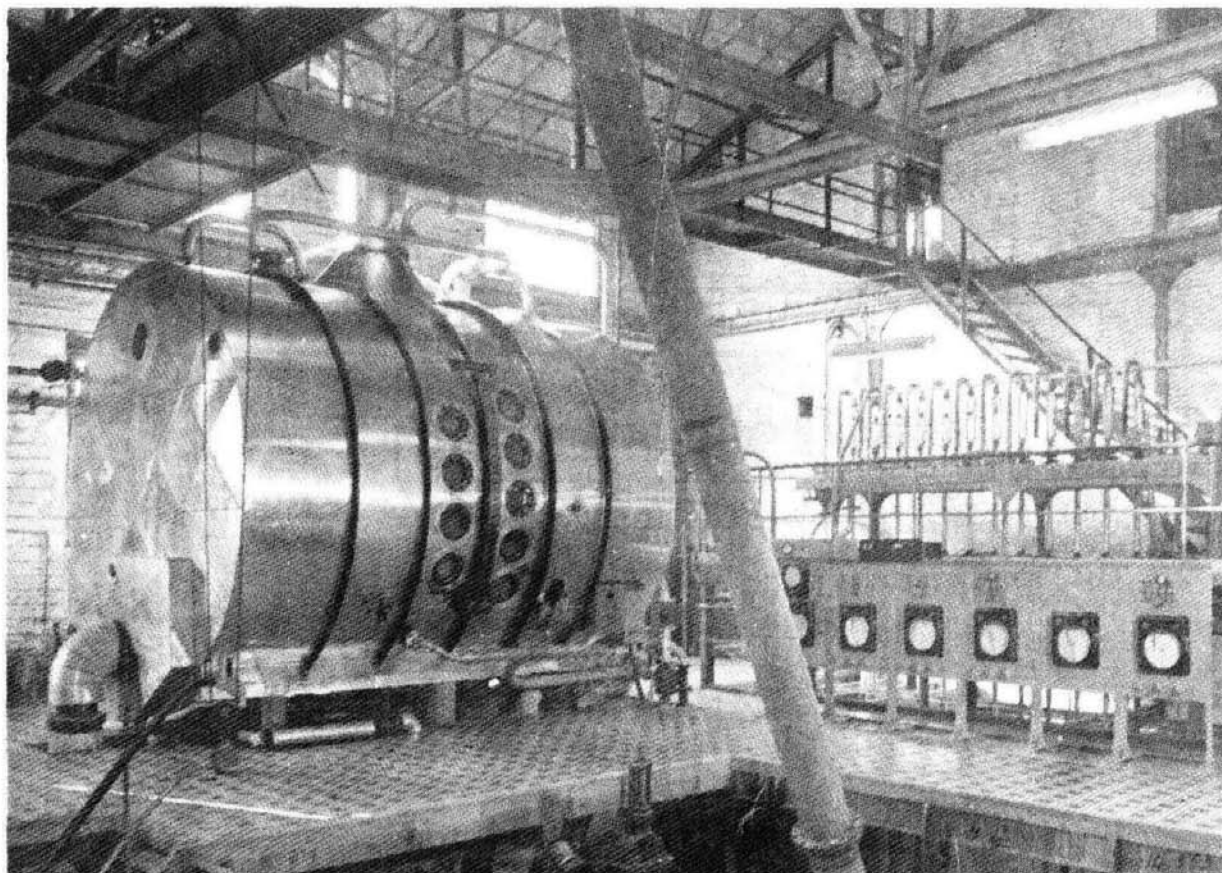


Fig. 33.13. Horizontal pan (Fives Cail - Babcock).

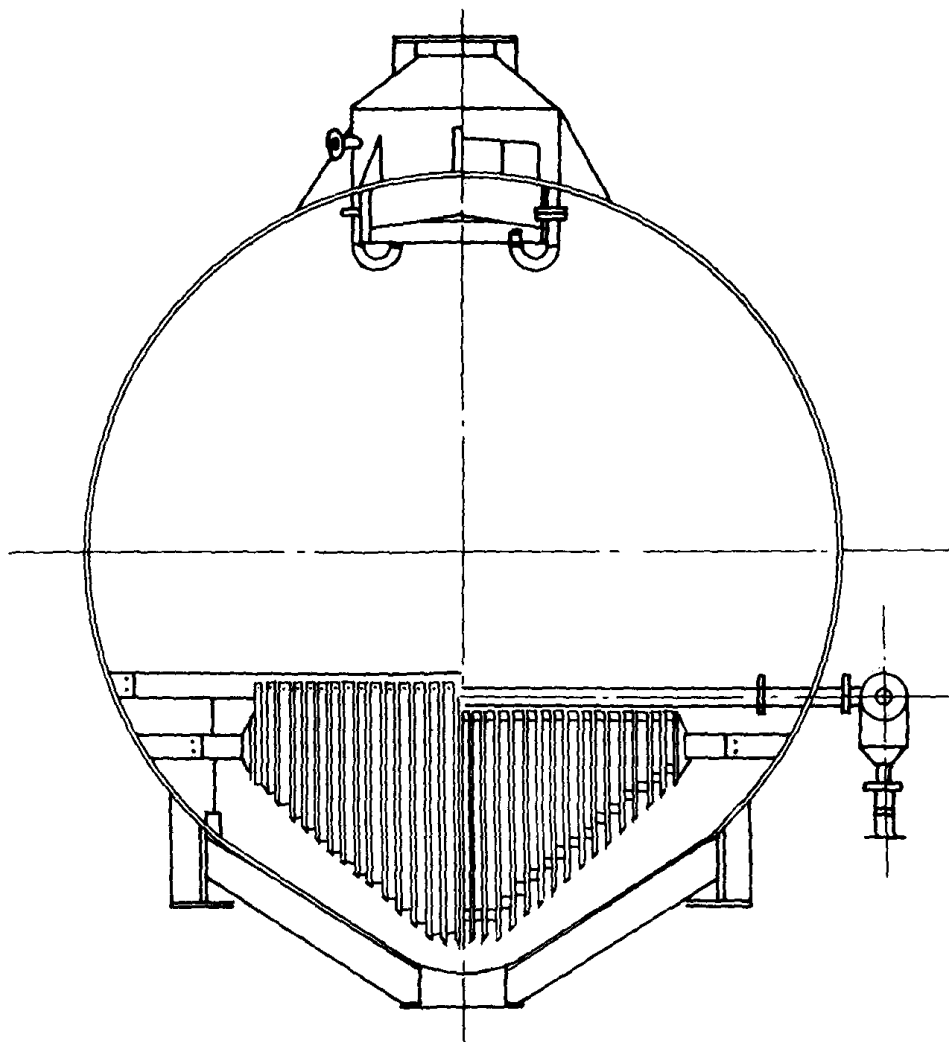


Fig. 33.14. Horizontal pan. Transverse section.

required for removing the calandria. This necessitates provision of chutes from the end discharge openings, to conduct the massecuite to the longitudinal receiver.

This pan offers the advantage of ready conversion to a continuous pan of Fives Cail-Babcock design. However, it is no longer made.

Tubes

Returning to the consideration of ordinary calandrias with tubes: the tubes are of steel, brass, or stainless steel.

The length of tubes and consequently the height of the calandria vary generally from 75 to 125 cm (30-50 in.). In calandrias with inclined lower plates, they may reach 140 cm (55 in.) at the centre.

Certain authors estimate that by reason of the viscosity of the massecuite, and the consequent laminar flow along the wall of the tubes, the portion of the tube length beyond 75 cm (30 in.) according to some, or 100 cm (40 in.) according to others, will be of little use; they argue that metal and volume are wasted, as the length indicated is sufficient to raise to the

required temperature the film of massecuite touching the tube. There is some truth in this argument, since the heating of the massecuite follows the exponential law shown in eqn. (30.4), in which the variable would be surface area S of tube corresponding to its length; however, the conditions are still rather remote from the asymptotic relationship, and it would be more accurate to say that, as the material rises in the tube, each cm of length plays a less and less important part. According to one experiment by Webre¹⁴, the difference in temperature between heating vapour and massecuite, close to the tube at the upper end of 120-cm (48-in.) tubes, is still 30–40°C (55–70°F) in the most unfavourable period. There is thus nothing illogical about installing tubes of 120, 125 or even 140 cm (48, 50 or 55 in.) in length. It is, however, advisable to keep within 125 cm or less and not to exceed this length except at the apex of conical calandrias.

The interior diameter of tubes varies from 89 to 127 mm ($3\frac{1}{2}$ –5 in.) in general. Webre¹⁵ considers that the diameter should not be below 89 mm and that a good dimension is 101.6–114.3 mm (4 – $4\frac{1}{2}$ in.). Claassen¹⁶ considers that tubes should be of 89–101.6 mm ($3\frac{1}{2}$ –4 in.) in diameter, but he is presumably thinking only of beet sugar pans. French manufacturers have adopted a standard dimension of 101.6 mm (4 in.) o.d. for brass or steel tubes. There is a trend towards the use of tubes of 101.6 mm (4 in.) in external diameter, 1.5 mm thickness, if in stainless steel, 2–3.2 mm in ordinary steel, or 2 mm in brass.

We see later (p. 657) that the interior diameter of the tubes influences the circulation of the massecuite, other thing being equal. To have a minimal loss of head in the circulation of the massecuite, it is necessary to proportion the diameter of the downtake to that of the tubes. For example, for a downtake of diameter of 40% of that of the pan, the tube diameter should be 1/10th of the downtake diameter; for a downtake of 33% of the pan diameter, the tube diameter should be 1/17th of that of the downtake ($p = 0.06M'$, cf. p. 658). However, such a rule would lead to completely insufficient heating surfaces in large pans, or to prohibitively large vessels, and the gain in circulation would far from compensate for these disadvantages. Hence, with the object of simplification and homogeneity, the practice is adopted of using a reasonable tube diameter for all sizes of pan, for example, 97.4 or 101.6 mm ($3\frac{1}{2}$ or 4 in.), and proportioning the centre well accordingly. We may comment that with these tube diameters, a downtake of 33% will give a circulation close to the optimal ($M' = 0.33$; $p = 0.020$ – 0.035).

The pitch p of the tubes is the distance between centres of two adjacent tubes (cf. p. 512). The term "ligament" is used to designate the thickness of metal remaining on the tube plate between two adjacent tubes, in their common axial plane. It is often made 20, 25 or 25.4 mm (1 in.). With tubes of 101.6 mm, and ligament of 25.4 mm, the pitch will be: $p = 101.6 + 25.4 = 127$ mm (5 in.).

With stainless-steel tubes of 101.6 mm, the true ligament will be:

$$\text{pitch - hole} = 127 - 101.6 - 1.5 = 23.9 \text{ mm (0.94 in.)}$$

In the case cited, the coefficient k_i already defined (p. 513) relative to evaporator bodies will be:

$$k_i = \frac{\pi}{2\sqrt{3}} \cdot \frac{d_i^2}{p^2} = 0.907 \times \frac{9.722}{16.129} = 0.55$$

Taking into account, by a coefficient k_2 , the space lost on the surface of the calandria by stay-rods, incondensable-gas pipes, steam pipes and on the edges, we have:

$$k_2 = 0.70-0.80 \quad \text{and:} \quad k = k_1 k_2 = 0.40-0.45 \quad (33.6)$$

Estimation of coefficient k . The coefficient k which we have defined (eqn. (33.6)) varies between fairly wide limits, and it may be found difficult to fix its value. It is therefore of value to specify it particularly for the constructor of equipment for evaporators for vacuum pans. The coefficient may be split into the two sub-coefficients: $k = k_1 \times k_2$, k_1 being the ratio of area for passage of liquid in the tube to the area occupied by a tube on the tube plate. We have defined it (eqn. (32.14)) as:

$$k_1 = \frac{s_i}{\sigma} \quad k_1 = \frac{\pi}{2\sqrt{3}} \cdot \frac{d_i^2}{p^2} \quad (33.7)$$

d_i = interior diameter of the tubes
 p - pitch of tubes as defined above
 b = ligament.

The coefficient k_1 depends on the diameter d and pitch p ($p = d + b$). k_2 is the ratio of total cross-section of the tubes to the total area of the tube plate, thickness of side walls excluded: it depends on the areas lost in the neighbourhood of the side walls and various pipe connections. There is also more area lost when d_i is large compared with F (for floating calandria) or relative to $\Delta - D$ (pan with centre well); in other words, k_2 depends on the number of tubes which can be placed in the calandria.

For annular calandrias (with centre well), we may take according to the number of tubes:

$$k_2 = 0.97 - \frac{77}{n} \quad \text{or} \quad 0.97 - \frac{3.4}{\sqrt{n}} \quad (33.8)$$

We have thus for this case:

$$\text{If } n \geq 500: \quad k = \left(\frac{\pi}{2\sqrt{3}} \cdot \frac{d_i^2}{p^2} \right) \left(0.97 - \frac{77}{n} \right) \quad (33.9)$$

$$\text{If } 100 \leq n \leq 500: \quad k = \left(\frac{\pi}{2\sqrt{3}} \cdot \frac{d_i^2}{p^2} \right) \left(0.97 - \frac{3.4}{\sqrt{n}} \right) \quad (33.10)$$

Obviously:

$$k = k_1 k_2 = \frac{ns_i}{S} \quad (33.11)$$

The actual ligament is smaller than the theoretical thickness, on account of the tolerances

in manufacture of the tubes, which oblige the equipment manufacturer to drill in the plates holes which will admit the largest tubes. For steel tubes, for example, the tolerance is + 1.5%; for brass, + 1%.

Height of the pan

Construction firms generally furnish pans of a capacity calculated for a level of massecuite half-way up the highest window (or sight glass), *i.e.* a little below the top of the cylindrical portion of the pan.

In practice, the sugar boilers frequently exceed this mark and even the upper limit of the top sight glass, though this should be strictly forbidden by the foreman responsible for the supervision of the pan station.

Even if we consider only the limit fixed by the maker, this often gives a height of massecuite of 2.1 m (7 ft.) or more above the upper tube plate. Such heights, as we shall see shortly (p. 647), offer serious disadvantages owing to the effects of hydrostatic pressure. Hence the height of massecuite above the top tube plate is now limited, and care is taken not to exceed 1.5 m (5 ft.) for natural-circulation pans. With mechanical circulation, a height of 1.8–2 m (6–6.5 ft.) may be used without disadvantage.

To restrict entrainment, it is necessary to allow a certain height between the maximum massecuite level and the top of the pan. Tromp¹⁷ recommends a space equal to 85% of the maximal height of the massecuite above the top tube plate.

Bottom angle. The bottom of the pan generally takes the form of a truncated cone, but a segment of a sphere, or a "basket-handle" curve, or even a torus may be followed.

With the conical form, the angle of the cone with the horizontal is generally 17–25°. If the pan is of floating-calandria type, this angle should be 2–3° smaller than that of the bottom of the calandria. It is advantageous for graining volume that these angles should be small but, for rapid flow of the massecuite at the moment of discharge, an angle of less than 17° will not generally be used. According to the diameter of the pan, angles are adopted for pan bottom and calandria bottom of 25°/22°, 22°/19°, 20°/17°, the latter values applying to larger sizes of pan.

Graining volume

This term is used to denote the minimum volume of massecuite which must be introduced into the pan before turning on steam. It is obviously necessary to cover entirely the first coil or, in a calandria pan, the whole calandria, before steam is turned on. Otherwise, the ebullition would throw syrup on the exposed portions of heated metal; this would cause losses by caramelisation and would increase the colour of the sugar.

In practice, the massecuite should even be taken above the coil or the calandria by 50–75 mm (2–3 in.), so that the syrup can circulate above it and descend by the centre well. The graining volume, however, is calculated from the horizontal plane passing through the highest point of the coil or corresponding to the upper surface of the tube plate.

This volume is also termed the "footing volume" (*ped-de-cuite*).

It is generally expressed in per cent of the working capacity of the pan, and varies from 24 to 40%. It is obviously of advantage that it should be as low as possible, since the footing

is generally introduced in the form of virgin syrup or of high-purity magma; and it is only on the remainder of the strike that the purity can be reduced by the addition of first or second molasses. The graining volume, therefore, determines the possibilities of rapid exhaustion in the pan, a greater purity drop being obtained when a larger quantity of lower-grade molasses is used.

From this point of view, it is preferable not to go above 30% for the graining volume, or 35% as an extreme value.

For a coil pan, it is possible to remain much below these limits. For calandria pans, these values will generally not allow the strikes, particularly of low grade, to be completed in the one batch, in the one pan. In order to obtain the grain size and the crystal content required, it is necessary to "run up" the strike until the pan is full, then to "cut" by sending two-thirds of the charge into another pan or into a storage vessel, continuing the strike with the one-third remaining. The two-thirds thus cut over will be returned in two batches later in order to finish these strikes, or indeed they may be finished off in another pan.

EFFECTS OF HYDROSTATIC PRESSURE

When the syrup has just been introduced into the vacuum pan, the boiling is still somewhat similar to that which occurs in the last vessel of the multiple effect. But when the grain has been formed (or a pre-formed "footing" has been "cut in") and the charge is being built up, the material becomes increasingly viscous. Finally, at the moment of dropping, the massecuite is a glutinous, semi-solid mass, and circulation becomes a real problem.

The boiling massecuite, at a temperature of 80–90°C (175–195°F), has a specific gravity of about 1.47, (density of approximately 92 lb./cu.ft.). The pressure in the massecuite will therefore increase by about 0.15 kg/cm² per m (0.64 p.s.i. per foot) of depth.

Now, at each point, the temperature of the boiling massecuite is obviously equal to the boiling point of water under the pressure obtaining at that point, plus the boiling-point elevation corresponding to the brix of the massecuite at that pressure.

If there were no circulation, successive layers of massecuite would have temperatures increasing with the depth.

On the other hand, the degree of saturation of a sugar solution changes rapidly according to its temperature. If the uppermost layer is supersaturated, the following layers are increasingly less so, and there is a depth at which saturation temperature is reached. From there on, the lower layers are undersaturated, and a certain amount of re-resolution of the crystals occurs.

It is obvious that, in practice, matters are not as simple as this; the massecuite is circulating and its currents distort the isothermal planes which could otherwise have been recognised in the pan, change them to irregular surfaces, mix the whole material, and tend to bring it to a uniform temperature. It follows that some zones will consist of supersaturated massecuite at depths greater than the critical depth we have referred to, while inversely there will be under-saturated massecuite at shallower depths; and our theoretical view will, after all, give a fair average picture of the state of the material in the pan.

Webre¹⁸ has given an interesting study of this question. Assuming that the temperature of

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TABLE 33.3

EFFECTS OF HYDROSTATIC PRESSURE

Depth		Hydrostatic pressure		Local vacuum		B. p. of water at local vac.		B.P.E.		Theor. temp. of mass.		Saturation	Crystals
(m)	(ft.)	(cm Hg)	(in. Hg)	(cm Hg)	(in. Hg)	(°C)	(°F)	(°C)	(°F)	(°C)	(°F)		
(A) Vacuum 58.5 cm - brix 81.5													
0	0	0.0	0.0	58.4	23.0	63.7	146.8	9.3	16.7	73.1	163.5	1.33	Grow
0.6	2	6.6	2.6	51.8	20.4	70.8	159.5	9.4	17.0	80.3	176.5	1.25	Grow
1.2	4	12.9	5.1	45.5	17.9	76.4	169.6	9.6	17.3	86.1	187.0	1.18	Grow
1.8	6	19.6	7.7	38.9	15.3	81.2	178.2	9.8	17.6	41.0	195.8	1.14	Grow
2.4	8	25.9	10.2	32.5	12.8	85.3	185.5	9.9	17.8	45.2	203.3	1.10	Grow
3.0	10	32.5	12.8	25.9	10.2	88.7	191.8	9.9	17.9	98.7	209.7	1.08	Grow
3.6	12	39.1	15.4	19.3	7.6	92.0	197.6	10.1	18.1	102.1	215.7	1.05	Grow
(B) Vacuum 68.5 cm — brix 77.7													
0	0	0.0	0.0	68.6	27.0	46.1	115.0	6.7	12.1	52.9	127.2	1.33	Grow
0.6	2	6.6	2.6	62.0	24.4	58.9	138.0	7.1	12.8	66.0	150.8	1.13	Grow
1.2	4	12.9	5.1	55.6	21.9	67.2	153.0	7.3	13.2	74.6	166.2	1.04	Grow
1.8	6	19.6	7.7	49.0	19.3	73.4	164.2	7.5	13.5	80.9	177.7	0.98	Dissolve
2.4	8	25.9	10.2	42.7	16.8	78.6	173.4	7.6	13.7	86.2	187.1	0.94	Dissolve
3.0	10	32.5	12.8	36.1	14.2	83.1	181.6	7.8	14.0	90.9	195.6	0.91	Dissolve
3.6	12	39.1	15.4	29.5	11.6	86.8	188.2	7.9	14.2	94.7	202.4	0.88	Dissolve

the massecuite in each layer corresponded exactly to the theoretical boiling point which it should have in accordance with the hydrostatic pressure to which it is subjected, he has considered two massecuites of similar brix and purity, in two similar pans, but at two different vacua, chosen at the extremities of the usual scale of operations: (1) 58.5 cm (23 in.), (2) 68.5 cm (27 in.). (See Table 33.3.)

It should be recalled that, at lower vacuum, temperatures are higher and consequently:

- (1) The massecuite viscosity is lower
- (2) Crystal growth is more rapid.

Table 33.3 gives another reason for boiling at a low vacuum, and explains why "soft" or rounded crystals are obtained by working at a high vacuum, and "hard" sharp-edged crystals at a reduced vacuum. What happens is that, in the first case, the crystals are partly dissolved when they reach the bottom of the pan; this, to some extent, undoes the work which has been performed in the upper part.

To avoid this, it is preferable to:

(a) Use a reasonable vacuum: 61 – 66 cm (24-26 in.). Claasen¹⁹ stipulates that 66 cm should not be exceeded, and even recommends 58.5 cm (23 in.) for calandria pans. Considering risks of inversion, the author recommends 63.5 – 66 cm (25 – 26 in.).

(b) Use a pan in which the height of the massecuite is restricted to a reasonable value.

(c) Use a design of pan which favours the movement of the massecuite as much as possible, or, if necessary, actually provides a positive circulation.

MASSECUITE CIRCULATION IN VACUUM PANS

This is a much more important subject than is generally realised. In most factories where efficiency is not high, molasses exhaustion is the aspect in which there is most to be gained. Now, this exhaustion depends very much on the design of the vacuum pans and particularly on the extent to which they facilitate and speed up the circulation of the massecuite at the end of the strike.

Hence we intend to devote to this question a large section of this Chapter.

We proceed to examine the circulation in the two principal types of pan: the pan with central downtake and that with floating calandria.

What is it that promotes circulation? Putting aside pans with mechanical circulation, which we shall study separately, circulation in vacuum pans is promoted by the bubbles of vapour due to heating in the tubes: these bubbles tend to rise, growing as they do so, and agitate the mass and lift it towards the surface. The circulation is thus due to the heating, and for a given pan, a given massecuite level and the same type of massecuite, is approximately proportioned to the degree of heating. If heating is stopped, circulation also ceases.

The circulation is assisted, but to a slight extent only, by the difference in specific gravity of the hot massecuite rising in the tubes and the cooler massecuite descending in the centre or annular downtake after releasing its heat by evaporation at the surface. The vapour bubbles, however, are the main promoter of circulation.

While the circulation is caused and accelerated by this motive force, it is, on the other hand, slowed down by the resistance due to the shape and disposition of the calandria tubes, and the resultant pressure drop.

Descending passage. The return of massecuite from top to bottom of the pan takes place by the central **downtake**, or the annular space, depending on the type of pan.

Older pans had a very small centre well, for example, 20% of the inside diameter of the pan, hence a very poor circulation. About 1935, the disadvantage of this narrow restriction began to be realised. Webre²⁰ recommended a downtake of diameter 50% of the inside diameter of the pan; Tromp²¹ recommended 40%; in Hawaii²² a great improvement was obtained by installing pans with a centre well diameter of 45–50%. We discuss later the calculation of the optimal diameter of the downtake (p. 656).

The analogous problem for the floating-calandria pan is to determine the optimal interval between calandria and wall of the pan (p. 658). While modern pans with central downtake generally have calandria with flat upper and lower tube plates, and consequently tubes of equal lengths, the floating-calandria pan has a lower tube plate of truncated cone form and consequently tubes of unequal lengths. We shall consider two principal lengths:

- (a) The height at the outer edge of the calandria, L_e .
- (b) The mean length of the tubes, L . This mean length is equal to the height of a cylinder of the same diameter as the calandria and of equal volume. Generally:

$$l = \frac{L_e}{L} = 0.7-0.8 \quad (33.12)$$

Relation between number and diameter of tubes. The definition of the coefficient k (p. 645) gives us, for a pan with central downtake:

$$k = \frac{n \frac{\pi d^2}{4}}{\pi \left(\frac{\Delta^2 - D^2}{4} \right)}$$

Hence:

$$n = k \frac{\Delta^2 - D^2}{d^2} = k \frac{1 - m^2}{p^2} \quad (33.13)$$

k = ratio of the total interior cross-section of the tubes to the total plane surface of the upper tube plate

n = number of tubes

d = interior diameter of tubes

Δ = interior diameter of the pan at the level of the calandria, in the same unit

D = inside diameter of the downtake, in the same unit

$m = D/\Delta$

$p = d/\Delta$.

In the same way, for the floating-calandria pan:

$$k = \frac{n \frac{\pi d^2}{4}}{\frac{\pi F^2}{4}}$$

Hence:

$$n = k \frac{F^2}{d^2} = k \frac{f^2}{p^2} \tag{33.14}$$

$$f = \frac{F}{\Delta}$$

Relation between diameter and useful capacity of the pan. In modern pans, with a normal tube length and a maximal massecuite level of the order of 1.5 m (5 ft.) above the upper tube plate, the useful massecuite capacity may be taken as proportional to the square of the diameter:

$$v = \frac{V}{\Delta^2} = 1.9 \quad \left\{ v = \frac{V}{\Delta^2} = 6.2 \right\}, \tag{33.15}$$

v = ratio of useful capacity to the square of the pan diameter

V = useful capacity of the pan, in m³ (cu.ft.)

A = inside diameter of the pan, in m (ft.).

We have thus:

V (m ³)	A (m)	V (cu.ft.)	A (ft.)
20	3.25	500	9
30	4	1,000	12.7
40	4.6	1,500	15.5
50	5.1	2,000	18
60	5.6	2,500	20
80	6.5	3,000	22

Losses of head

The whole problem of massecuite circulation is governed by the losses of head suffered by the massecuite in the circuit which it describes in the pan. Its movement is relatively free in the upper part of the pan and even near the bottom; the principal resistances which it encounters are: (1) passing through the tubes, which causes a loss of head J; (2) its return by the downtake or annular space, which produces a loss of head I'. Other causes of head loss may be neglected relative to these two main factors.

We shall consider the circulation only in a pan filled with tight and heavy massecuite, near the end of the strike. This is in fact the only interesting case, since in any pan, even if badly designed, circulation is easy while the massecuite is light and mobile. It is only at the end of the strike that well-designed pans clearly exhibit their superiority and their fast boiling.

Now, as pointed out by Jenkins²³, the movement of the massecuite, in the final phase of the charge which we are considering, is in viscous or laminar flow in the tubes and the centre

well or the annular space. Turbulent flow occurs only in the early stages of the charge, before graining or when the grain is still small and insignificant in quantity. We may therefore apply Poiseuille's law which applies for laminar flow:

$$J_p = \frac{Q}{n} \cdot \frac{128 \mu}{\pi d^4} \cdot \frac{1}{10 g} \quad (33.16)$$

and:

$$J'_p = QL \frac{128 \mu}{\pi D^4} \cdot \frac{1}{10 g} \quad (33.17)$$

J_p = loss of head for the upward current passing through the calandria tubes, in kg/m²

J'_p = pressure drop for the descending current passing through the centre well, in kg/m²

Q = volume flow of each of these streams, in m³/s

n = number of tubes in the calandria

L = height of the calandria (assuming horizontal tube plates), in m

μ = viscosity of the massecuite, m poise

d = inside diameter of the tubes, in m

D = inside diameter of the centre well, in m

g = 9.81 m/s/s = acceleration due to gravity.

This equation also applies if all dimensions are in feet, viscosity in f.p.s. units, head losses in lb./sq.ft., volume flow in cu.ft./s, and g in ft./s/s.

We now require to know the corresponding values J_F and J'_F for the floating-calandria pan. J_F has a similar form to J_p , it being understood that L now represents the mean length of the calandria tubes (p. 650). It is now required to calculate J'_F .

We shall first derive an expression for the head loss as a function of the dimensions of the annular space, in other words, the equivalent of Poiseuille's law for an annular space. We shall retain the same notations as above, adding:

F = calandria diameter, in m (ft.)

$2E$ = width of the annular space = $(A - F)/2$.

We shall consider an annulus of massecuite, on each side of the median cylinder of average radius R situated in the middle of the annular space, equidistant from the calandria and the wall of the pan (Fig. 33.15).

The median annulus of massecuite, of thickness $2e$ (e on each side of the median cylinder), is in equilibrium under the effects of: (1) the viscosity μ of the medium; (2) the pressure difference or loss of head J'_F . We have

$$- 2\pi L_e (R + e + R - e) \mu \frac{du}{de} = J'_F \pi [(R + e)^2 - (R - e)^2]$$

whence:

$$\frac{du}{de} = - \frac{J'_F}{\mu L_e} e \quad u = - \frac{J'_F}{2\mu L_e} e^2 + \text{constant}$$

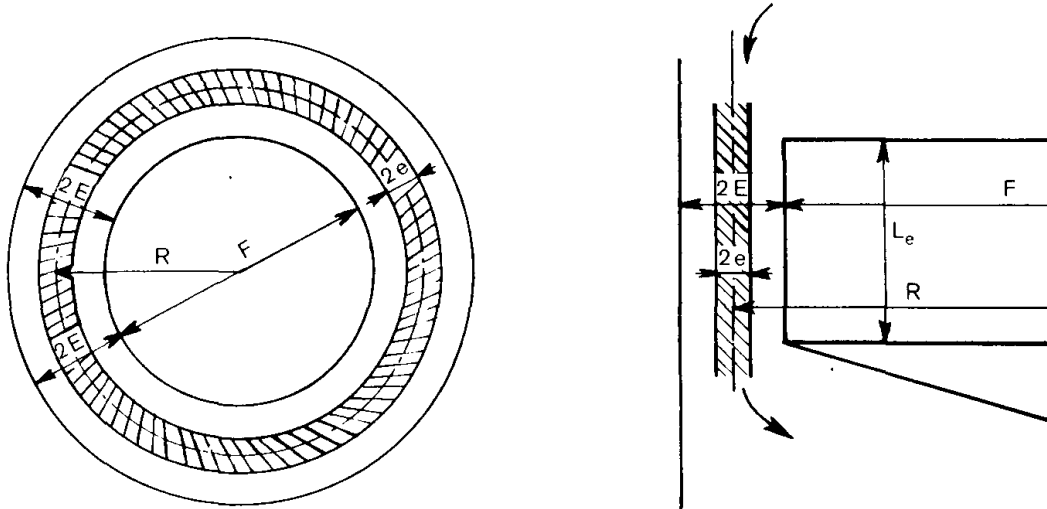


Fig. 33.15. Floating calandria. Calculation of head loss.

We shall determine the constant from the fact that, for $e = \pm E$, $u = 0$:

$$0 = - \frac{J'_r}{2\mu L_e} E^2 + \text{constant} \quad \text{constant} = \frac{E^2 J'_F}{2\mu L_e}$$

whence:

$$u = \frac{J'_r}{2\mu L_e} (E^2 - e^2)$$

Considering a very thin cylinder of massecuite of thickness de , the corresponding flow of massecuite has the value:

$$dQ = 2\pi(R + e) de \cdot u + 2\pi(R - e) de \cdot u = 4\pi R de \cdot u$$

or.

$$dQ = \frac{2\pi R J'_F}{\mu L_e} (E^2 - e^2) de$$

Hence:

$$Q = \frac{2\pi R J'_F}{\mu L_e} \int_{-E}^{+E} (E^2 - e^2) de = \frac{8\pi R E^3 J'_F}{3\mu L_e}$$

Finally, we have the value for the loss of head corresponding to the passage of massecuite along the annular space:

$$J'_F = \frac{3\mu Q L_e}{\pi R E^3}$$

In f.p.s. units:

$$J_i = QL_e \frac{\mu}{g} \cdot 1 \quad (33.18)$$

or, in m.k.s. units:

$$J'_F = QL_e \frac{3\mu}{10 g} \cdot 1$$

Since:

$$D = \frac{\Delta + F}{4} \quad \text{and:} \quad E = \frac{A - F}{4}$$

we may also write (m.k.s. units):

$$J'_F = QL_e \frac{0.75}{\pi (\Delta + F) (\Delta - F)^2} \cdot \frac{1}{10 g} \quad (33.19)$$

or again, since (eqn. (33.12)) $L_e = lL$:

$$J'_F = QL \frac{128\mu \times 0.75 l}{\pi (\Delta^2 - F^2) (A - F)^2} \cdot \frac{1}{10 g} \quad (33.20)$$

Order of magnitude. To assist the explanation, we shall ascertain the magnitude of J_p , J_F , and of J'_F . Eliminating with this object the common and very variable factors, we may write:

$$\lambda = \frac{128\mu QL}{10 g \pi \Delta^4} \quad (\text{m.k.s. units})$$

Thus we have:

$$J_p = X \frac{1}{np^4} = \lambda \frac{1}{k(1 - m^2)p^2} \quad J_F = \lambda \frac{1}{np^4} = X \frac{1}{kf^2 p^2}$$

$$J'_p = \lambda \frac{1}{m^4} \quad J'_F = \lambda \frac{0.75 l}{(1 - f^2)(1 - f)^2} \quad (33.21)$$

Taking values of:

$$p = 0.03 \quad m = 0.4 \quad / = 0.85 \quad / = 0.83 \quad (k = 0.5)$$

we have:

$$\begin{array}{r} J_p = 2,646 X \\ J'_p = 39 \lambda \\ \hline J_p + J'_p = 2,685 X \end{array} \quad \begin{array}{r} J_F = 3,076 X \\ J'_F = 100 X \\ \hline J_F + J'_F = 3,176 X \end{array}$$

This shows that, for typical values, the floating-calandria pan would theoretically give a total loss of head greater than that for a standard type of pan of the same diameter. In order to make these equal, it would be necessary, again theoretically, that the floating-calandria pan should have a diameter Δ' such that:

$$\left(\frac{\Delta'}{\Delta}\right)^4 = \frac{3,176}{2,685} \text{ or: } \Delta' = 1.043 \Delta$$

In fact it is found that for equal diameters, the floating-calandria pan gives a circulation at least as good as that of the pan with central downtake. This is probably due to the difficulty encountered by the massecuite descending from the surface in a downtake closely surrounded by a hot rising massecuite which encloses it and quickly encroaches upon it.

Equivalent diameter. In hydraulics, from the point of view of head losses, a non-circular cross-section may be treated as a circular one having the same ratio:

$$\frac{\text{Cross-section of passage}}{\text{Wetted perimeter}}$$

For a circular section, this ratio has a value:

$$\frac{\frac{\pi D^2}{4}}{\pi D} \sim \frac{D}{4}$$

For an annulus between two circles of respective diameters A and F , the equivalent diameter D will thus be given by:

$$\frac{D}{4} = \frac{\frac{\pi (\Delta^2 - F^2)}{4}}{\pi (A + F)} = \frac{A - F}{4}$$

Hence:

$$D = A - F \quad \text{or:} \quad m = 1 - f \tag{33.22}$$

For a narrow passage e between two plane walls of infinite or very great length z , we would have in the same way:

$$\frac{D}{4} \sim \frac{Ze}{2Z} \sim \frac{e}{2}$$

$$e = \frac{D}{2}$$

This gives a useful comparison between tubes and plate elements.

Dimensioning for minimal head losses

It is interesting to seek the relationship between the diameter d of the tubes and the diameter D of the centre well, or the width $(A - F)/2$ of the annulus, when the total head losses are minimal:

(1) **Pan with centre well.** We have (eqns. (33.13), (33.16) and (33.17)):

$$J_p + J'_p = \frac{128 \mu Q L}{10 g \pi r} \left(\frac{1}{nd^4} + \frac{1}{D^4} \right) \quad \text{with:} \quad n = k \frac{\Delta^2 - D^2}{d^2}$$

For a given pan and a given rate Q of massecuite in the circuit, we require to find the minimum value of:

$$z = \frac{1}{nd^4} + \frac{1}{D^4} = \frac{1}{k(\Delta^2 - D^2)d^2} + \frac{1}{D^4} = \frac{1}{\Delta^4} \left[\frac{1}{k(1 - m^2)p^2} + \frac{1}{m^4} \right]$$

or of:

$$y = \frac{1}{k(1 - m^2)p^2} + \frac{1}{m^4}$$

Differentiating this expression with respect to m :

$$\frac{dy}{dm} = \frac{2kp^2m}{k^2(1 - m^2)^2p^4} - \frac{4}{m^5}$$

This is equal to 0 for:

$$2kp^2m^6 = 4k^2p^4(1 - m^2)^2$$

or:

$$m^6 = 2kp^2(1 - m^2)^2 \quad (33.23)$$

Let $r = p\sqrt{2k}$. Expression (33.23) becomes:

$$m^3 + rm^2 - r = 0 \quad (33.24)$$

Let $m = x - r/3$; (33.24) then becomes:

$$x^3 - \frac{r}{3}x + r\left(\frac{2r^2}{27} - 1\right) = 0 \quad (33.25)$$

which has the form $x^3 + ax + b = 0$.

We can thus write:

$$\frac{b^2}{4} + \frac{a^3}{27} = \frac{r^2}{4} + \frac{2r^2}{27} - \frac{r^6}{27^2} + \frac{1-r^2}{27} > 0$$

The equation thus has a unique root: $x = A + B$. Writing:

$$A = \sqrt[3]{-\frac{b}{2} + \sqrt{\frac{b^2}{4} + \frac{a^3}{27}}} = \sqrt[3]{r \left(\frac{1-r^2}{2} + \sqrt{\frac{1-r^2}{4} - \frac{r^2}{27}} \right)}$$

Since $\sqrt{\frac{1}{4} - e} \approx 5 - e$, and since r is very small and r^2 negligible relative to $\frac{1}{2}$, we may write:

$$A = \sqrt[3]{r \left(\frac{1-r^2}{2} + \frac{1-r^2}{2} \right)} = \sqrt[3]{r}$$

In the same way:

$$B = \sqrt[3]{r \left(\frac{1-r^2}{2} - \frac{r^2}{27} \right)} - r \sqrt{\frac{1-r^2}{4} - \frac{r^2}{27}} = \sqrt[3]{r \left(\frac{1-r^2}{2} - \frac{r^2}{27} - \frac{1}{2} + \frac{r^2}{27} \right)} = 0$$

The root is thus:

$$x = A + B = \sqrt[3]{r}$$

Hence:

$$m = x - \frac{r}{3} = \sqrt[3]{r} - \frac{r}{3}$$

or:

$$m = \sqrt[3]{p \sqrt{2k}} - p \frac{\sqrt{2k}}{3} \tag{33.26}$$

We see that, for a pan of a given diameter A , the head loss in the circulation of masecuite will be minimal for a certain value of m , that is, of the diameter of the centre well, which depends on the relative diameter p of the tubes.

This conclusion is important and has never been stated before the present author drew attention to it²⁴. Most manufacturers do not take it into account, and adopt a fixed ratio m of the downtake, while using a standard tube diameter (hence a variable value of p) in all their pans from the smallest to the largest. Failure to recognise this is moreover more surprising since, without any calculation, the interdependence of the relative diameters m and p , from the point of view of head loss, is evident from first considerations.

Equation (33.26) gives the following table of values of m corresponding to various values of p :

TABLE 33.4

OPTIMAL THEORETICAL VALUES OF RELATIVE DIAMETERS m OF TUBES AND CENTRE WELL

k	P				
	0.020	0.025	0.030	0.035	0.040
0.40	0.2556	0.2743	0.2904	0.3047	0.3176
0.45	0.2604	0.2794	0.2958	0.3103	0.3234
0.50	0.2648	0.2841	0.3007	0.3154	0.3287
0.55	0.2688	0.2883	0.3052	0.3201	0.3335

It is realised that the theoretical values of m are at fault, on account of the zone of perturbation around and above the cylinder which forms the centre well. To obtain practical optimal values, which we designate by M' , we apply to them the correction given by the following formula:

$$M' = m + \frac{1 - m}{10} \quad (33.27)$$

- M' = practical optimal value of the relative diameter of the downtake = D/Δ
 m = theoretical optimal value of this diameter, given by eqn. (33.26) or Table 33.4
 D = diameter of centre well
 A = diameter of the pan at the calandria level.

Thus $M' = 0.37$ for $m = 0.30$. This gives us Table 33.5 of practical values of M' .

It will be noted that, for normal conditions, there is no advantage in exceeding 40% for the centre well, if the correction suggested by our formula (33.27) is sufficient.

TABLE 33.5

OPTIMAL PRACTICAL VALUES OF RELATIVE DIAMETER M' OF CENTRE WELL

k	P				
	0.020	0.025	0.030	0.035	0.040
0.40	0.330	0.347	0.361	0.374	0.386
0.45	0.334	0.351	0.366	0.379	0.391
0.50	0.338	0.356	0.371	0.384	0.396
0.55	0.342	0.359	0.375	0.388	0.400

(2) **Floating-calandria pan.** Following the same calculation as for the pan with centre well, the total head loss is (eqns. (33.16) and (33.20):

$$J_F + J'_F = \frac{128 \mu Q L}{10 g \pi} \left[\frac{1}{n d^4} + \frac{0.75 /}{(\Delta'^2 - F^2) (\Delta' - F)^2} \right]$$

and we require the minimum value of (cf. eqn. 33.14):

$$z = \frac{1}{kF^2 d^2} + \frac{0.75 l}{(\Delta'^2 - F^2) (\Delta' - F)^2} = \frac{1}{\Delta'^4} \left[\frac{1}{kf^2 p^2} + \frac{0.75 l}{(1 - f^2) (1 - f)^2} \right]$$

that is, of:

$$y = \frac{1}{kf^2 p^2} + \frac{0.75 l}{(1 - f^2) (1 - f)^2}$$

Differentiating this expression with relation to / we have:

$$\frac{dy}{df} \sim \frac{2}{f^3} + \dots - \frac{(1 + 2f)}{(1 - f^2)^2 (1 - f)^2}$$

We require the value of *f* for which

$$\frac{dy}{df} = 0 \quad \text{or:} \quad \frac{(1 - f^2)^2 (1 - f)^2}{(1 + 2f) f^3} = 0.75 kl p^2$$

We shall solve this equation by a graphical method, by drawing the graph representing the value of the 1st member as a function of *f*. We then give to the 2nd member the various values which we will take for various values of *p*. Taking the values of / on the graph corresponding to these diverse values of the 2nd term, we may construct the graphs of Fig. 33.16 which express between themselves the optimal values of *f* and of *p*, as a function of the commonest values of the product *kl*. These optimal values of *f* are all included between 0.86 and 0.92, most often between 0.87 and 0.91. As for the central downtake, we find that the relative interval (1 - *f*)/2 depends on the relative diameter of the tubes, *p*.

For a reason analogous to that quoted with relation to a centre well, it is realised that these optimal theoretical values of *f* are in error, being too high. Accordingly we shall take as the optimal dimension that furnished by the value *f'*:

$$f' = f - \frac{1 - f}{5} \tag{33.28}$$

Hence we obtain:

TABLE 33.6

OPTIMAL PRACTICAL VALUES OF RELATIVE DIAMETER *f'* OF A FLOATING CALANDRIA

<i>kl</i>	<i>P</i>				
	0.020	0.025	0.030	0.035	0.040
0.30	0.898	0.886	0.877	0.867	0.859
0.40	0.890	0.878	0.867	0.858	0.849
0.50	0.885	0.872	0.860	0.850	0.842

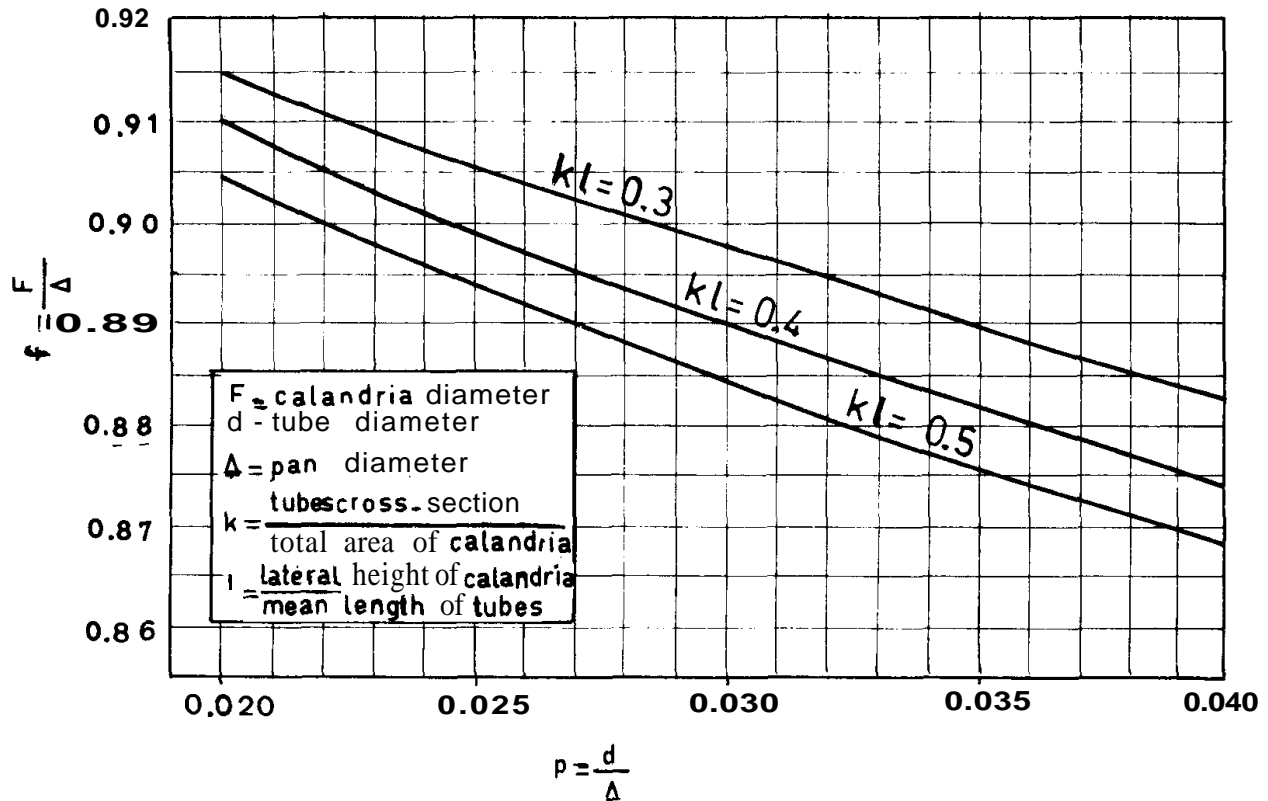


Fig. 33.16. Floating-calandria pans. Relation between f and p for minimal total head loss.

It is obvious that the use of different tube diameters would present disadvantages. However, it is necessary to remember the influence of the ratio of diameters of the tubes and the centre well, for example, if only to fix the area of the descending current of massecuite in terms of the standard diameter which should be adopted for the tubes.

Choice of centre well or floating calandria

The difference between the results recorded with these two types of pan is rather slight, and can vary according to the more or less careful design of the pan.

We have been concerned in fact with losses of head, but there are three other important characteristics:

- Graining volume
- Maximum level of massecuite
- Heating surface.

(a) The graining volume depends largely on the design of the bottom of the pan. With standard designs, the two types of pan do not differ much from this point of view, but it is possible to obtain a graining volume slightly more advantageous with the centre-well pan. The floating-calandria pan can with difficulty give a value below 30% and frequently reaches 33%.

(b) As regards maximum level of massecuite, there is very little difference between the two types of pan.

(c) Considering heating surface, we have seen (p. 655) that the floating-calandria pan would theoretically require a slightly greater diameter to avoid having a higher head loss for equal heating surface. Generally, it is preferred to have a calandria slightly smaller, and consequently a heating surface slightly lower; the results obtained are nevertheless equivalent.

Speed of circulation

Webre²⁵ has determined the velocity of circulation of massecuite in a calandria pan, of the following characteristics:

Diameter of pan:	3.6 m (12 ft.)
Diameter of centre well:	0.9 m (36 in.)
Height of calandria:	1.2 m (4 ft.)
Diameter of tubes:	0.127 m (5 in.)
Number of tubes in calandria:	418
Heating surface:	195 m ² (2,100 sq.ft.)

Maximum height of massecuite above upper tube plate:	2.1 m (7 ft.)
Working volume:	34 m ³ (1,200 cu.ft.)
Massecuite boiled during test:	C massecuite

The graph in Fig. 33.17 gives the mean circulation speeds found in the course of the strike. The velocity of circulation, which averaged 0.47 m/s (1.53 ft./s) during the first hour, fell to 0.0043 m/s (0.014 ft./s) during the sixth hour. It is seen that, 2 hours after the commence-

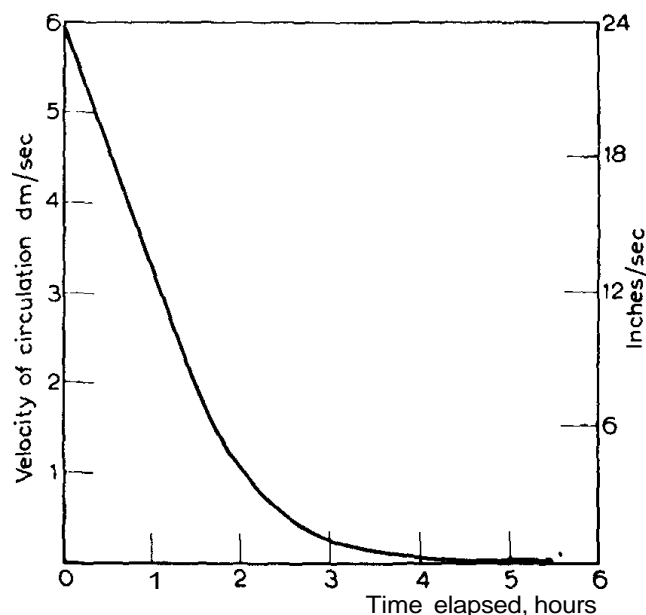


Fig. 33.17. Circulation speed during a strike.

ment of the strike, the speed of circulation has already fallen to a very low value. Near the end, it is negligible.

In the beet industry, Claassen²⁶ gives the following velocities:

	Purity	Velocity			
		at start		at finish	
A massecuite	96	1	-1.68 m/s	(3.3 -5.5 ft./s)	0.22 m/s (0.72 ft./s)
B massecuite	82	0.33	-0.22 m/s	(1.08 -0.72 ft./s)	0.006 m/s (0.02 ft./s)

With cane, Bosworth²⁷ indicates the following mean velocities of circulation:

Refinery massecuite: 20 cm/s (8 in./s)
 Raw factory A massecuite: 10 (4)
 Raw factory AB massecuite: 8 (3)
 Raw factory B massecuite: 4 cm/s (1.6 in./s)
 Raw factory C massecuite: 2 (0.8)

EVAPORATION COEFFICIENT

The evaporation coefficient of a pan is the weight of water evaporated from the massecuite per unit heating surface in unit of time. It is generally expressed as kg/m²/h (lb./sq.ft./h).

The variation in evaporation coefficient in the course of a strike is similar to the variation of the velocity of circulation, which we have just discussed. The same tests by Claassen in a beet factory gave:

	Purity	Evaporation coefficient			
		(kg/m ² /h)		(lb./sq.ft./h)	
		At start	At finish	At start	At finish
A massecuite	96	55	0.7	11	0.14
B massecuite	82	10	1.0	2.05	0.2

These figures correspond to the following heat-transfer coefficients:

	Purity	Heat-transfer coefficient			
		(kcal/m ² /°C/h)		(B.T. V./sq.ft./°F/h)	
		At start	At finish	At start	At finish
A massecuite	96	850 - 1,000	200	180-200	40
B massecuite	82	200- 300	50-100	40- 60	10 - 20

Webre²⁸ gives, for cane massecuites:

	Purity	Evaporation rate, in kg/m ² /h (lb./sq.ft./h)		
		At start	At finish	Average
B massecuite	12	34 (7)	1.2 (0.25)	11 (2.26)

the heat-transfer coefficient varying, in the course of the strike, from 490 to 24 kcal/m²/°C/h (100 – 4.88 B.T.U./sq.ft./°F/h).

For calandria pans in Cuba, the same author²⁹ gives finally the evaporation coefficients quoted in Table 33.7). We add some figures from South Africa:

TABLE 33.7 (cf. TABLE 40.8)

EVAPORATION RATES OF PANS, IN kg/m²/h (lb./sq.ft./h)

	Maximum	Minimum	Adopted in S. Africa
Footings	85 (17.5)	17 (3.5)	79 (16.1)
A massecuites	71 (14.6)	32 (6.6)	59 (12.0)
B massecuites	46 (9.4)	11 (2.2)	49 (10.0)
C massecuites	36 (7.5)	1.7 (0.34)	39 (8.0)

Honig³⁰ gives, for an A massecuite of purity 86, with a coil pan and heating steam at 1.25 – 1.4 kg/cm² absolute (18 – 20 p.s.i.a.), the following details of the values of evaporation coefficient:

Concentrating 60° brix syrup to saturation:	120 – 70 kg/m ² /h	(25-15 lb./sq.ft./h)
Introduction of magma and establishing the grain:	70-58	(15 – 12)
From footing to final volume of massecuite:	58-47	(12 – 10)
Heavying up the massecuite during the last 20 minutes:	47 – 23	(10- 4.7)

HEATING STEAM

Formerly, coil pans were heated with direct steam, generally reduced to 2 – 2.8 kg/cm² gauge (30 – 40 p.s.i.g). At present, all strikes are heated either with exhaust steam or, to a considerable extent, by Wed vapour. Good modern pans, however, are necessary to enable sugar boiling to be done with bled vapour, and the boiling is obviously slightly slower. On the other hand, there is less risk of decomposition of sucrose. Sometimes even vapour under slight

vacuum is used. Generally, vapour from the 1st effect is used; if the boiling becomes too slow or difficult at the end of the strike, a change is made to exhaust steam for heavying up.

When bled vapour is used, it is advisable to open the incondensable-gas valves more widely (*cf.* "Settings of Valves", p. 528).

Steam entry

For pans with centre well, it is preferable to distribute the steam through several entries placed around the calandria. A great quantity of steam surging against the outer tubes immediately in front of the steam entry produces a very rapid circulation at that point at the beginning of the strike, at the expense of other zones of the calandria. At the end of the strike, however, during heavying up, this zone near the steam entry produces false grain. From this point of view, it is also advisable to provide each steam entry with a conical baffle to distribute the steam, and avoid such direct impingement on the tubes facing the steam entry.

For floating-calandria pans, the best system of steam entry is to pass the steam inlet pipe through the save-all and bring it down vertically to the centre of the calandria. The pipe is furnished with a jacket leaving a space of 19 mm ($\frac{3}{4}$ in.) under vacuum between the steam pipe and jacket, to avoid splashes of massecuite becoming caramelised on the hot pipe.

STEAM CONSUMPTION AT THE PAN STAGE

In evaporation, we have seen (p. 563) that 1 kg of heating steam was sufficient to evaporate approximately 1 kg of water from the juice.

At the pan stage, if the steam consumption of the pans is expressed as a ratio of the theoretical quantity of water to be evaporated from the syrup, it is seen that this ratio is appreciably greater than 1. This is due to water used to dilute the molasses, water for washing out the pans, and water for dissolving false grain, all of which represent additional water which must be evaporated as well as that present in the syrup.

What then is the ratio k to be allowed for between the quantity of steam required and the quantity of water originally present in the syrup?

Deerr ³¹ estimates:	1.43
Tromp ³² :	1.74
Webre ³³ :	1.23

Webre³⁴, however, has more recently given the following more detailed values, relative to a juice quantity of 45,000 kg (100,000 lb.) per hour at 16° brix:

	Evaporation		Steam consumption		Coefficient k
	(kg/h)	(lb./h)	(kg/h)	(lb.h)	
Strikes A	2,760	6,090	3,035	6,700	1.10
B	1,015	2,240	1,165	2,575	1.15
C	890	1,970	1,070	2,365	1.20
Total	4,665	10,300	5,270	11,640	1.13

The coefficient in this case, however, takes into account **only** heat losses, as the water for diluting molasses is included in the 1st column; this gives a mean coefficient $k_m = 1.30$:

$$\begin{array}{r} k_A = 1.27 \\ k_B = 1.32 \\ k_C = 1.38 \\ \hline k_m = 1.30 \end{array}$$

At Pleystowe, in **Australia**³⁵, and in the **Philippines**³⁶, the quantities of water, in addition to concentration requirements, have been measured. Table 33.8 gives the figures obtained:

TABLE 33.8

QUANTITY OF WATER TO BE EVAPORATED AT PANS, IN kg/t.C. (lb./t.c.)		
	<i>Australia</i>	<i>Philippines</i>
Movement or balancing water	13.7 (30.2)	26.0 (57.5)
Pan washout water	2.9 (6.5)	
Molasses dilution water	13.2 (29.1)	6.5 (14.3)
Wash water at centrifugals	5.5 (12.2)	1.9 (4.2)
Total added water	35.3 (78.0)	34.4 (76.0)
Original water from syrup		49.8 (110.0)
Total water to be evaporated		84.2 (186.0)

The figures for the Philippines correspond to an evaporation 1.7 times the theoretical quantity of water to be evaporated from the syrup. However, the quantities added may show great variations between one country and another; in **Taiwan**^{36a}, for example, figures have been reported of 1.3 with defecation and 1.6 for carbonation.

We may determine this coefficient k by a calculation similar to that of p. 562 applied to a vacuum pan for the whole period of a strike. The heat lost is obviously very important, on account of the time necessary to effect the necessary heat exchange. It may be estimated at about 20% of the total heat transferred during the strike. The result depends to a great extent on the quantity of water added to the molasses and returned to the pan, expressed as a percentage of the water originally present in the syrup. This varies greatly according to the methods used. In the French sugar factories which we are acquainted with, it should scarcely exceed 10–20%.

In these conditions, we find $k = 1.5$. FCB admits $k_m = 1.52$.

For the individual steam consumption of a vacuum pan, we may take as a mean³⁷ for the duration of the charge 0.22 t/h of steam per m³ (14 lb./h/cu.ft.) of nominal capacity for A masseuite, or 0.18 (11.5) for C masseuite.

Values generally accepted. Widely diverging figures are given in the technical literature for the steam consumption of vacuum pans:

	<i>Steam consumption</i>	
	<i>(kg/t.c.)</i>	<i>(lb./t.c.)</i>
Lopez and Clark ^{37a} :	123	275
Paturau ^{37b} :	125	280
Fletcher and Stewart, vapour at 104°C:	140	314
Jenkins ^{37c} : 115 – 170 kg/t.c.; mean:	150	336
In Australia ³⁷ :	160	358
In Brazil:		
ordinary pan:	165	370
pan with circulator:	128	287

Calculation of steam consumption. For each of the three massecuites, we may reckon:

$$\text{Steam consumption} = Qkm \text{ kg/h per t.c.} \quad (33.29)$$

Q = kg of massecuite per t.c.h.

Massecuite A: $k = 0.275$	Batch operation: $m = 1.5$
Massecuite B: $k = 0.300$	Continuous operation: $m = 1.2$
Massecuite C: $k = 0.360$	

Example. Assuming A massecuite is boiled in batch operation, and B and C in continuous operation:

Volume of A massecuite = 140 l/t.c.	Weight = 140 x 1.47 = 206 kg/t.c. (460 lb./t.c.)
Volume of B massecuite = 54 l/t.c.	Weight = 54 x 1.47 = 80 kg/t.c. (180 lb./t.c.)
Volume of C massecuite = 36 l/t.c.	Weight = 36 X 1.47 = 53 kg/t.c. (119 lb./t.c.)

Steam consumption:

A strike: 206 x 0.275 X 1.5 =	85 kg (190 lb.) steam per t.c.
B strike: 80 x 0.300 x 1.2 =	29 kg (65 lb.) steam per t.c.
C strike: 53 X 0.360 X 1.2 =	23 kg (50 lb.) steam per t.c.

	137 kg (305 lb.) steam per t.c.

Steam for agitation. Due to the high density of the massecuite at the end of the charge, it circulates very slowly at that stage and remains almost motionless in certain zones of the pan. Mechanical circulation contributes strongly to improve the situation, but one remedy adopted is admission of live steam into the massecuite by means of a circular perforated pipe at the bottom of the old type of pans, about one-third of the radial width of the calandria, reckoned from the outer wall of the pan, that is, at a distance from the wall of:

$$\frac{1}{3} \left(\frac{\Delta}{2} - \frac{D}{2} \right)$$

In South Africa, this agitating steam is given the name of "jigger steam" (perhaps implying steam which makes the massecuite dance a jig). The quantity of additional steam thus consumed may attain 50% of the evaporation produced as a whole in the pan. The coefficient m indicated above can thus extend, for example, from 1.5 to 2.25. For economy, Fives Cail-Babcock, in its continuous pans, uses for such agitation the incondensable gases escaping from the pan.

HEATING SURFACE AND RATIO OF HEATING SURFACE TO VOLUME

A very important factor in the design and performance of vacuum pans is the ratio of their heating surface to their working capacity. We have just seen (p. 598) that this ratio in itself is not sufficient to characterise the evaporative capacity or the speed of a pan; but, for the majority of pans with tube diameters as used at present, it constitutes a dominating factor.

For a convenient expression of this ratio, the working volume is expressed in cubic metres, with heating surface in square metres (in cubic feet and square feet in the British system).

We have then:

$$\frac{S}{V} = \frac{\text{Heating surface of pan}}{\text{Working capacity}} \quad (35.30)$$

A well-designed pan should have a definite ratio S/V . However, this ratio should depend:

- (a) On the steam used for heating
- (b) On the massecuite to be handled.

(a) **Steam.** Coil pans, utilising steam at pressures between 0.5 and 4 kg/cm² (7 and 60 p.s.i.), generally have a ratio S/V of 3.3-4.9 m²/m³ (1-1.5 sq.ft./cu.ft.), preferably 3.9-4.6 (1.2-1.4).

Calandria pans, utilising exhaust steam at a pressure of 0.5-1 kg/cm² (7-15 p.s.i.), should have a ratio S/V of 4.9-6.6 m²/m³ (1.5-2.0 sq.ft./cu.ft.).

If these pans are intended to use normally bled vapour from effects, at a pressure of 0-0.42 kg/cm² (0-6 p.s.i.), it will be advisable to increase their heating surface so as to raise their ratio S/V to 6.6-7.2 m²/m³ (2-2.2 sq.ft./cu.ft.).

The optimum proportions of heating surface are summarised in Table 33.9.

In other words, the heating surface will be increased all the more according as the steam used is at lower temperature.

TABLE 33.9

RATIO OF HEATING SURFACE TO VOLUME

Type of H.S.	Steam used	Steam press.		Optimum S/V ratio	
		(kg/cm ²)	(P.s.i.g.)	(m ² /m ³)	(sq.ft./cu.ft.)
Coils	Live steam	0.5-4	7-60	3.9-4.6	1.2-1.4
Calandria	Exhaust	0.5-1	7-15	4.9-6.6	1.5-2.0
Calandria	Bled vapour	0-0.4	0-6	6.6-7.2	2.0-2.2

(b) **Massecuite.** The influence of the massecuite to be treated is less important than that of the steam. However, it is desirable to take it into account, since the massecuite becomes more viscous as its purity decreases. Now it is necessary to heat a viscous, badly circulating massecuite more carefully and more gently, on account of the risk of caramelisation and formation of false grain. Hence it is desirable to decrease the heating surface and also the steam pressure, when low-grade material is being treated, so that the rate of boiling may be reduced as required by the viscosity of the material.

Table 33.10 gives, for example, suitable figures for different grades of massecuite, using exhaust steam.

TABLE 33.10

OPTIMUM VALUE OF RATIO S/V FOR DIFFERENT STRIKES

	<i>Time for strike</i> (hours)	<i>Ratio S/V</i> (m^2/m^3)	<i>Ratio S/V</i> (sq.ft./cu.ft.)
Refined sugars	2	9.8	3
Special white sugars	$2\frac{1}{2}$ - 3	8.2	2.5
1st massecuites	3 - 4	5.9	1.8
Low-grade massecuites	4 - 8	4.9	1.5

HEIGHT OF MASSECUITE, CIRCULATION, HEATING SURFACE AND GRAINING VOLUME

It must not be supposed that a desired ratio of heating surface to volume and a certain grain- ing volume can be adopted independently. There are in effect four important factors to which one should strive to give the optimum value when designing a pan of a given capacity:

(a) The maximum height of the massecuite, which should be as small as possible, to avoid re-resolution of crystals.

(b) The circulation, which should be as rapid as possible, in order to give rapid working and a good exhaustion.

(c) The graining volume, which should be as small as possible, in order to permit the maximum exhaustion with the minimum volume of massecuite per tonne of cane.

(d) The heating surface required to give the desired ratio S/V . For A and B massecuites, the higher the heating surface, the faster will be the pan; for low-grade pans, where such fast boiling is not required, the greater the heating surface, the lower will be the pressure and temperature of the heating steam required, hence lower heat losses and less colour formation.

It is readily seen that these four factors are contradictory:

(a) If the height of the massecuite is decreased, circulation will be improved, but the grain- ing volume will be increased. Or, if the same graining volume is maintained, the heating sur- face will be decreased.

(b) If the circulation is to be improved, for a given type of pan, it is often necessary to increase the centre well or the annular space, and thus to decrease the heating surface. The graining volume at the same time is increased. In the same way, increasing the tube diameter would improve the circulation, but at the expense of heating surface, in two ways: firstly,

because, for a given calandria, the heating surface is inversely proportional to tube diameter; secondly (see Tables 33.5 and 33.6), since, when the tube diameter is increased, the diameter of the centre well or the width of the annular space must be increased in order to conserve optimal proportions.

Finally, we see that, for a given type of pan, a given diameter of tubes, and a given circulation space (central or other downtake), a compromise must be reached between height of massecuite, graining volume and heating-surface - volume ratio. It is not possible to do more than choose the optimum compromise for the duty desired.

These dictates of design render it difficult to design a pan having, for example, more than 5.9 m^2 heating surface per m^3 (1.8 sq.ft./cu.ft.) of useful capacity, without exceeding a graining volume of 30%.

Only the adoption of mechanical circulation would render solutions possible which otherwise are absolutely incompatible. In particular, it permits of much greater massecuite heights, of the order of 2.1 m (7 ft.), without serious risk as regards re-solution of grain.

CAPACITY OF PANS

The capacity of a vacuum pan is generally measured by the maximum volume of massecuite which it can contain. This maximum level is generally (*cf.* p. 646) in the middle of the top sight glass; which obviously is a way of saying that the designer places the top sight glass in such a position that its centre coincides with the maximum level which he has fixed.

However, the volume of a pan is only superficially the principal factor in its capacity in terms of tonnes cane per hour. The determining factor is in reality its heating surface, which determines the quantity of water which it can evaporate in the course of the season from the syrups, molasses and massecuite which it will have to handle. Many people reckon the capacity of a factory as regards its pans, according to the heating surface of the latter.

There exists moreover a certain proportion between heating surface and working volume (*cf.* p. 667). If a comparison is made between various pans, some with coils and some calandrias, and if allowance be made for the fact that some of these are heated with exhaust steam, others by bled vapour, and if it be assumed that all these pans are correctly proportioned (that is to say, they have the desired heating-surface - volume ratio), it will then be preferable to refer to them in terms of volume, because the heating surface has been modified according to the steam or vapour used for heating. In other words, three pans each of 28 m^3 (1,000 cu.ft.):

the first a coil pan, using reduced live steam, with heating surface $s = 130 \text{ m}^2$ (1,400 sq.ft.),

the second a calandria pan, using exhaust steam, $s = 165 \text{ m}^2$ (1,800 sq.ft.),

the third a calandria, heated by vapour from the 1st vessel, $s = 185 \text{ m}^2$ (2,000 sq.ft.), will have practically the same working capacity, that is to say, they would boil massecuites of the same category (A, B or C) in the same time.

For this reason it remains logical to reckon pan capacities in terms of volume. It will be of interest to indicate also the heating surface.

We consider that in raw sugar manufacture and for calandria pans, a pan capacity of 1.6

m³ (57 cu.ft.) with a heating surface of 9- 10 m²/t.c.h. (100 – 110 sq.ft./t.c.h.) should be provided. The distribution of this will be indicated later (p. 676). With horizontal pans with plate elements of the F.L.C. type, these figures could be very readily reduced to 1.5 m³ (54 cu.ft.) and 9 mVt.c.h. (100 sq.ft./t.c.h.). In South Africa and Mauritius^{36b}, 1.7 mVt.c.h. (60 cu.ft./t.c.h.) is generally available as a mean volume.

SUGAR-BOILING PROCEDURES

When concentration of the syrup coming from the evaporators is continued in a vacuum pan, when the massecuite has been "grained" and "built up", and when this massecuite has been subsequently passed to the centrifugals, the sugar remains in the centrifugal basket, and the mother liquor passes through the gauze. The mother liquor so separated is termed the "molasses" from the centrifuged massecuite. The first massecuite obtained from virgin syrup (to which sometimes a certain proportion of first molasses is "boiled in" to the pan) is called A (or first) massecuite, and the mother liquor separated from it in the centrifugals is called A molasses, sometimes first molasses or A syrup.

However, this A molasses still contains a high proportion of crystallisable sugar. It is therefore put aside, and is used, for example, to "build up" further massecuites on a suitable footing. Such a massecuite is then called B or second massecuite, and the mother liquor which is separated from it in the centrifugals is termed B (or second) molasses.

This operation may be repeated several times, but the number of massecuites is soon limited, because:

- (1) The molasses becomes more and more exhausted of sugar
- (2) All the sugar which it contains is not crystallisable, since the non-sugars immobilise a certain proportion of sugar
- (3) The decrease of purity consequent on the removal of sugar, together with the repeated boiling, mixing and centrifuging, give rise to an increase in the viscosity of the successive molasses obtained, rendering the last massecuites very difficult to treat and to keep in circulation in the pans.

The last molasses obtained, which is considered to be practically exhausted, or from which no further recovery is attempted, is termed "final molasses", or in common parlance simply "molasses". It is sent out from the factory to the distillery, where unfortunately it takes with it the considerable proportion of sugar which it still contains; though the attempt to recover this sugar has been given up, it is very far from being negligible in quantity, since it generally constitutes the greatest loss suffered in the manufacture.

The procedure employed in order to attain the desired recovery of sugar is obviously of great importance. Accordingly we shall study the processes generally employed.

Purities. All processes of sugar boiling are based on purity. Indeed, purity is the simplest criterion of exhaustion. Unfortunately, it is not the best: one molasses at 36 apparent purity may be well exhausted; another, at 34, may result from defective work and could have been easily reduced to 32.

There are several ways of calculating the purity p . The purity has the value:

$$p = \frac{s}{B} \times 100 \quad (33.31)$$

p = purity

s = sugar % in the material analysed

B = total material in solution, as % of this material.

Now, the quantity of sugar s may be estimated in two ways:

(a) As *sucrose*: for this it is necessary to use the Clerget method of analysis, which is time-consuming.

(b) As *polarisation*, for which a single reading in the polariscope is sufficient. Since this method is much simpler and more rapid, it is the one most frequently used. However, on account of the presence of reducing sugars, the rotatory power of which is opposite to that of sucrose (levo-rotary instead of dextro-rotary), the polarisation always gives a figure for sugar content lower than the true quantity of sucrose contained in the solution.

In the same way, the total solids in solution (sugars plus non-sugars) are determined in three different ways:

(a) *Dry substance*, or solids by drying.

(b) *Refractometric dry substance*, obtained by the reading of a refractometer. This figure generally differs only slightly from the true dry substances, but is always higher.

(c) *Gravimetric solids*, or "brix", obtained by the reading of a Brix saccharometer or a hydrometer. This figure differs from the true dry substance in the same direction but more markedly than the refractometric brix, thus giving a figure appreciably higher. Although the use of the refractometer has increased greatly, many sugar countries still use the hydrometer, which means that brix and purity figures are not comparable between the two groups of countries.

According to the analytical methods employed, we thus distinguish:

(a) *True purity*:

$$p = \frac{\text{sucrose}}{\text{dry substance}} \quad (33.32)$$

(b) *Gravity purity*:

$$p = \frac{\text{sucrose}}{\text{brix}} \quad (33.33)$$

(c) *Apparent purity*:

$$p = \frac{\text{mol}}{\text{brix}} \quad (33.34)$$

Certain American countries still conduct factory control on the basis of apparent purities. There is little disadvantage in this for control of the clarification or for that of molasses exhaustion. It is useful, however, to determine the sucrose (Clerget) for materials which influence the sucrose balance to a substantial extent, that is, at least for mixed juice and for final molasses. The question of exhaustion will be discussed in terms of apparent purities, with special comment when the question of Clerget purity is involved for molasses.

An approximate calculation is given by

$$\text{True purity} = 21.5 + 0.785 \times \text{Apparent purity} \tag{33.35}$$

In the discussion which follows we shall assume that all the purities are apparent figures.

The boiling process commences with syrup the purity of which is of the order of 85, and generally finishes in the cane sugar factory with a molasses the apparent purity of which is approximately:

Mediocre exhaustion (or unfavourable conditions):	40
Good exhaustion (or normal conditions):	36
Very good exhaustion (or good conditions):	32
Exceptional exhaustion (or very favourable conditions):	28

(1) Two-massecuite system

This term describes the process of operating in two stages. It is little used except in temperate countries where the syrup purity is very low. It consists (Fig. 33.18) of forming two types of massecuite:

(a) A first massecuite, of purity 75, formed from a footing of straight syrup, with return of high-purity first molasses and, if necessary, lower-purity first molasses (see below)

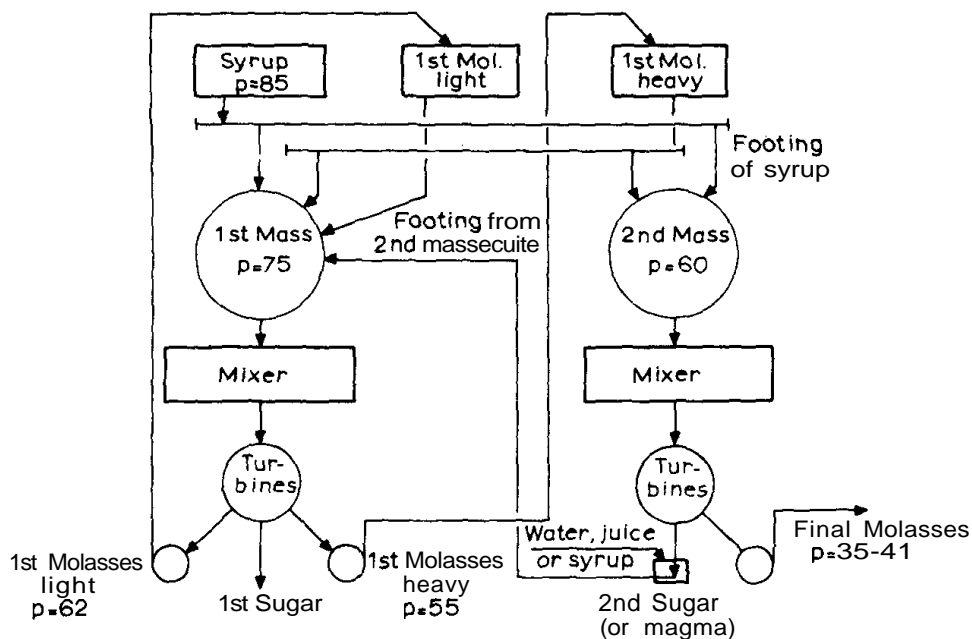


Fig. 33.18. Two-massecuite system.

(b) A second massecuite, of 60 purity, built up on a footing of straight syrup, with return exclusively of lower-purity molasses.

At the centrifugals the first massecuite may be treated by double purging, or by single purging with separation of heavy molasses (fugalling without washing) and light molasses of higher purity (corresponding to the period of washing with water and steam during which a certain quantity of sugar dissolved by the water or steam goes out with the molasses).

The purities which should be obtained are indicated in the diagram given (Fig. 33.18), which is self-explanatory. The sugar from the second massecuite may be either despatched as it is, for sale, or made into magma in the mixer below the centrifugals for use as footing for the first massecuites.

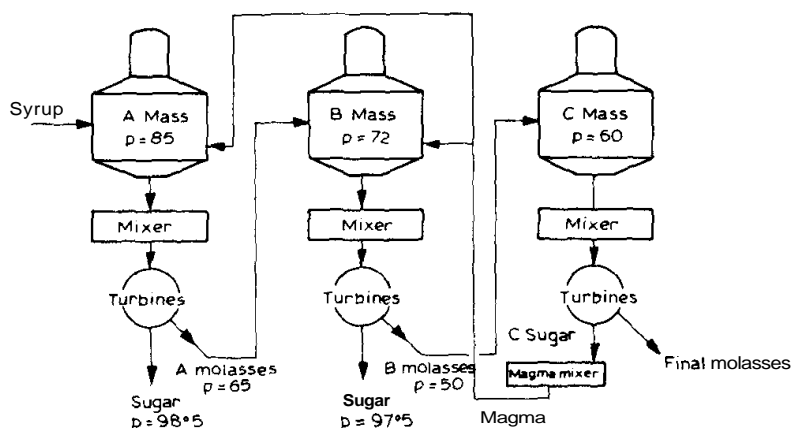


Fig. 33.19. Three-massecuite system.

(2) Three-massecuite system (Fig. 33.19)

This is the system most frequently adopted. Three types of massecuite are boiled:

(a) A massecuite of purity 80 - 85, formed on a footing of C magma and built up, according to the syrup purity, either entirely of straight syrup or on syrup with more or less substantial additions of A molasses near the end of the strike

(b) B massecuite of purity 68 - 72, formed on a magma footing and built up entirely of A molasses

(c) C massecuite of purity 56 - 60, grained on a blend of syrup and A molasses of purity 72-75, and built up with B molasses, after establishing and growing the grain.

The A and B sugars are despatched for sale, being centrifuged generally with light washing with water. In this case a light-coloured raw sugar is obtained, of polarisation about 98.5 for the A and 97.5 for the B sugar. The amount of washing may be varied according to the quality of sugar required. More often a more thorough washing with steam is used. Generally the rich and poor molasses are not separated, to avoid complicating the operation, since these two molasses differ only slightly according to the degree of washing employed.

The C sugar is pasty (due to adhering molasses); it may be remelted, but the general and most economic method consists of remixing it below the low-grade centrifugals with water, juice or syrup. The best solution is perhaps the use of juice, taken from the clear-juice box of the clarifier, stored and cooled in a small unlagged tank.

If there is an excess of C sugar relative to the requirements for A and B footings, it is remelted; but this excess can be avoided by controlling the grain size. For example, it may be **planned** to produce a C sugar of 0.3 mm, a B sugar of 0.7 and an A sugar of 1 mm. The size of the C grain depends on the quantity of ground-sugar slurry introduced in the graining, and on the fineness of the mixture.

Remelting the C sugar, though it is less economical, has at least the advantage of helping to give a purer A sugar crystal and one of better filterability.

The purity drops are generally of the order of:

- 18 – 20 points between A massecuite and A molasses
- 21 – 23 points between B massecuite and B molasses
- 22 — 30 points between C massecuite and C molasses.

(3) Four-massecuite system

As far as we know, this system is utilised only in a few factories in Hawaii and the Antilles and to a diminishing extent in Australia. The author has used it only in exceptional cases.

Four types of massecuite are boiled³⁸:

- (a) A massecuite, formed from the straight syrup
- (b) AB massecuite, formed from one-third of syrup and two-thirds of A molasses
- (c) B massecuite, formed from one-quarter of syrup and three-quarters of AB molasses
- (d) C massecuite, formed from one-eighth syrup and seven-eighths of B molasses.

The purities obtained are, according to the mutual control of Queensland factories^{38a} (averages for 1940), as given in Table 33.11.

The sugar from the first three massecuites forms the shipment sugar, with purities of 99.5, 99 and 98, respectively.

This system, like the preceding one, may equally well be used with magma or grained footings.

TABLE 33.11

PURITIES IN 4-MASSECUITE SYSTEM

<i>Massecuite</i>	<i>Brix</i>	<i>Massecuite purity</i>	<i>Mol. purity</i>	<i>Purity drop</i>	<i>Crystal % brix</i>
A	91.8	86.9	68.8	18.1	57.8
AB	92.4	79.4	59.5	19.9	49.1
B	92.9	72.9	53.4	19.5	41.8
C	93.5	61.2	36.8	24.4	38.6

This procedure increases the quantity of molasses reboiled, and consequently tends to increase their viscosity, to an extent which risks losing the benefits obtained from the additional complications of the system.

Double-magma system. In the 3-massecuite process, which is by far the commonest, it is possible to arrange to produce only A sugar, by utilising C sugar, mixed into magma as footing for the B massecuites, and using B sugar again in the form of magma as footing for

the A strikes. Even in this case, it may be arranged so that there is no excess of C sugar; but there is unavoidably an excess of B sugar. The neatest way of operating then consists of separating the B centrifugals into two equal groups: one which supplies the sugar to be mixed as magma to serve as footing for the A massecuites, the other furnishing commercial sugar. The latter receives some water wash, while the other half receives no wash.

This system has the advantage that only about 13% of the commercial sugar consists of B sugar, instead of about 27%. It allows of more rapid fugalling of the portion of B sugar which is used as magma, with a better exhaustion since it is not washed; and, contrary to what may be expected, it requires hardly any extra capacity in pans or A centrifugals than does the single magma process.

Alternative procedures. There are three main systems of operating the three-masseccuite system:

(a) CB — CA. The C massecuite is seeded and its grain is used as magma to serve as footing for the A and B strikes.

(b) CBA. The C massecuite is seeded and its grain is used as magma to serve as footing for the B strike, the grain of which serves as footing for the A strike. The excess of crystal is remelted.

(c) System of footing for B and A. A syrup strike is seeded and this grain serves as footing for the B and C strikes. The C sugar is remelted, the B sugar serves as footing for the A massecuite.

Quantities of massecuites

Taking into account the molasses returned to the various charges, the total volume of massecuite will be of the order given in Table 33.12.

These volumes of massecuite per ton of cane will obviously vary with the brix and the purity of the mixed juice, above or below the mean values given.

TABLE 33.12

MASSECUITE VOLUME, IN litres/t.c.(cu. ft./t.c.)

	<i>System</i>			
	<i>2-mass.</i>	<i>Single-magma</i>	<i>Double-magma</i>	<i>4-mass.</i>
A massecuite	140 (5)	95 (3.4)	125 (4.5)	85 (3)
AB massecuite				55 (2)
B massecuite	30 (1)	50 (1.8)	50 (1.8)	35 (1.3)
C massecuite		37 (1.3)	37 (1.3)	25 (0.9)
Totals	170 (6)	182 (6.5)	212 (7.6)	200 (7.2)

Influence of purity on boiling time

Purity has a marked influence on velocity of crystallisation. Van Hook³⁹ indicates that, for

certain syrups, this crystallisation velocity falls by half when the purity drops from 100 to 95, and to one-ninth of its value when the purity drops from 100 to 80. Hence there are considerable differences between the times necessary for boiling A, B and C massecuites.

For a given massecuite, it is considered in Jamaica⁴⁰ that, provided the purity does not differ too greatly from normal, the time for boiling a strike increases by 2% when the purity falls by one unit (83 - 82, for example).

Distribution of pans between the various massecuites

The time necessary for a strike increases rapidly in proportion as the purity of the charge decreases (*cf.* Fig. 33.3 and Table 33.10). It follows that the proportions of the total pan volume to be reserved for each of these massecuites vary greatly from the proportions of the respective volumes of the massecuites.

With modern and well-designed pans, the following times of boiling are typical:

	<i>S. Africa</i>	<i>Fives Cail - Babcock</i>
A massecuite: $2\frac{1}{2}$ - 3 h	4½ h	3.3 h
B massecuite: 3 - 4 h	6 h	4.5 h
C massecuite: 4 - 8 h	9 h	8 h

This leads to the following distribution of pans in terms of useful capacity:

	<i>(hl/t.c.h.)</i>	<i>(cu.ft./t.c.h.)</i>		
A massecuite:	6	(21)	or:	37.5%
B massecuite:	4	(14)	or:	25 %
C massecuite:	6	(21)	or:	37.5%
Total	16	(56)		100

For the 4-massecuite formula, Tromp⁴¹ suggests;

A massecuite:	8 =	33%
AB massecuite:	7 =	29%
B massecuite:	5 =	21%
C massecuite:	4 =	17%
	24	100%

The first column simply gives the relative proportions.

Relative location of pans

The order in which pans are arranged or aligned on the pan stage is of some importance. In a 3-massecuite system, for example, there will be two (single-magma) or three (double-magma) sizes of grain, and it is important to avoid mixing grain of different sizes. Now the

panns are connected by a "cut-over" pipe of large diameter, since massecuites and magmas are difficult materials to transfer through a pipe, and there always remains a considerable volume of crystals in this piping after each "cut over" or transfer. The row or rows of pans should accordingly be located so that the normal transfers of the different types of massecuite or magma do not use the same portion of this piping system. This is easily avoided by a judicious arrangement of the pans; if it is impossible, in an old factory, two cut-over pipes may be provided.

Crystal yield from massecuites

The crystal yield from a massecuite (sometimes termed the exhaustion of the massecuite) is the proportion of crystal recovered from the massecuite, expressed as percentage of its sucrose content. It has the value:

$$r = \frac{100 (j - m)}{j (100 - m)} \quad (33.36)$$

j = purity of massecuite

m = purity of molasses from that massecuite.

The following figures have been reported from South Africa⁴² for average crystal yield:

A massecuite:	62.5%
B massecuite:	60.3%
C massecuite:	56.6%

The recovery from the C massecuite is by far the most important, since it determines the purity of the final molasses and consequently the overall recovery. It may vary between 42 and 60.

MOLASSES

Exhausted molasses

There is no absolute criterion to indicate whether the molasses obtained is really exhausted. According to Deerr⁴³, experience indicates that, in an exhausted molasses, for unit of non-sugar, 0.455 water is required to maintain the non-sugar in solution, and for unit of water, there is approximately 1.8 of sugar dissolved in this water. Hence about 0.82 of sugar for 1 of non-sugar.

Deerr states that sugar will not crystallise from such a molasses when further water is removed.

The composition of such molasses would be as follows:

Dry substance:	80%
Polarisation:	27
Sucrose:	36%
Non-sucrose:	44%
Brix:	90
True purity:	45
Gravity purity:	40
Apparent purity:	30

This composition obviously is given as typical only, and Deerr comments that in practice better figures are often obtained.

Exhaustibility of molasses

Lacking a better criterion, the degree of exhaustion of a molasses has generally been judged until recently by its purity, the Clerget purity being a better index than the apparent purity. However, it has been long realised that the purity reached depended on the composition of the impurities contained in the juice and consequently in the molasses. In particular, the purity attainable was:

- (a) Lower as the proportion of reducing sugars was higher
- (b) Higher as the proportion of ash was higher.

Many formulae have been proposed, of which several have been given in the second edition. The more important are the following:

- (a) The Douwes Dekker⁴⁴ formula:

$$P = 36 - 0.08r + 0.26c \quad (33.37)$$

P = true purity attainable with normal exhaustion of the molasses (sometimes termed target purity)

r = reducing sugars % of non-sucrose in the molasses

c = ash (sulphated) % non-sucrose in the molasses with:

$$\text{Non-sucrose} = \text{dry substance} - \text{Clerget sucrose}$$

and⁴⁵:

$$\text{Carbonated ash} = \text{sulphated ash} \times 0.752$$

The Douwes Dekker formula applied only to defecation factories was:

$$P = 33 - 0.055r + 0.31c \quad (33.38)$$

Dry substance, difficult to determine accurately, may be obtained by the formula of Sijlmans for defecation factories:

$$\text{Dry substance} = \text{brix by dilution } 1 : 9 \text{ by weight} - (0.72 \times \text{sulphated ash } \%)$$

The value for dry substance varies greatly depending on the method of analysis. The following are figures determined in Mauritius⁴⁶ for the same product the ash content of which was 14.36%:

Brix by dilution $\frac{1}{10}$ (wt./vol.):	93.34
Brix by dilution $\frac{1}{5}$ (wt./wt.):	93.09
Brix by dilution $\frac{1}{2}$ (wt./wt.):	88.60
Refractometric brix by dilution $\frac{1}{1}$ (wt./wt.):	83.98
By drying (under vacuum):	81.37
By drying (at 105°C):	79.94

It will be noticed that the refractometric brix approaches the true figure more closely than the gravimetric figure.

(b) In Queensland⁴⁷, the formula of the Sugar Research Institute is:

$$P = 40.67 - 17.8 \log \frac{R}{C} \quad (33.39)$$

P = true purity obtainable

R = reducing sugars% molasses

C = sulphated ash %.

(c) In South Africa⁴⁸:

$$P = 39.94 - 19.60 \log \frac{R}{C} \quad (33.40)$$

The similarity with the Australian formula suggests that a more general formula could be:

$$P = 40 - 18.5 \log \frac{R}{C} \quad (33.41)$$

(d) On account of the differences in these various formulae, Reunion uses the very simple formula:

$$P = 40 - 4 \frac{R}{C} \quad (33.42)$$

P = refractometric - Clerget purity obtainable, R and C as above.

Quantity of molasses per tonne of cane

If the molasses is not weighed, or if it is desired to check the result given by the materials balance of the factory, it is easy to calculate the weight of molasses per tonne of cane which the factory should produce.

Let:

Q = weight of mixed juice % cane

B = brix of mixed juice

P = purity of mixed juice

q = weight of molasses % on cane

B_m = brix of the molasses

P_m = purity of the molasses

ϵ = loss in muds + undetermined losses, expressed as a ratio of the sugar contained in mixed juice

r - boiling-house recovery (*cf.* p. 838).

The sugar contained in the mixed juice has the value QBP . Of this, the factory recovers $QBPr$.

The remaining sugar:

$$QBP(1 - r)$$

should be found in the molasses, muds and undetermined losses, if losses by inversion or caramelisation are neglected. Now the sugar lost in the molasses has the value:

$$qB_mP_m$$

We should therefore have:

$$QBP(1 - r) = qB_mP_m + QBPe$$

Hence:

$$q = \frac{QBP(1 - r - \epsilon)}{B_mP_m} \quad (33.43)$$

Example:

$$\begin{array}{rcl} Q = 98 & B = 15 & P = 85 \\ & B_m = 95 & P_m = 40 \\ & r = 0.90 & \epsilon = 0.01 \end{array}$$

This gives:

$$q \sim \frac{98 \times 15 \times 85 (1 - 0.90 - 0.01)}{95 \times 40} = 2.96\% \text{ on cane}$$

CONDUCT OF THE SUGAR BOILING

The job of the sugar boiler is certainly the most critical of all those in the factory. Although it tends to become more and more simplified and controlled by instruments, sugar boiling is still certainly a matter of skill, and this skill and this craft on the part of the sugar boilers still have a profound influence on the quality and yield of sugar.

A strike consists of four principal phases; these will be described with particular reference to C massecuites, which are the most critical.

(a) Concentration

The sugar boiler introduces into the pan a certain quantity of either syrup or a mixture of syrup and molasses of about 75 purity. This quantity is determined by the graining volume. It should be as small as possible, while being sufficient to cover the calandria, or the bottom coil; otherwise there would be caramelisation of the splashes and small drops of syrup which would make contact with the exposed portions of the heating surface.

The smaller the volume of syrup drawn in, or the graining volume, the smaller will be the number of crystals in the final massecuite, and the larger will be the crystals.

The sugar boiler should add further syrup as the volume in the pan is decreased owing to the concentration of the charge, so as to maintain the liquid level in the pan always high enough to cover the heating surface in use.

(b) Graining

As the syrup becomes concentrated, it becomes more viscous. When the supersaturation has attained the required value, graining is carried out. This was formerly done by the "waiting method" or by shocking, *i.e.* by sudden introduction of a "drink" of cold syrup. It is now done (with few exceptions) only by seeding.

In the absence of instruments which will be described later (p. 686), the right moment corresponds to a brix of the order of 80. The sugar boiler recognised this condition by the viscosity of the syrup, the drops of which flow down more and more slowly on the sight glasses. He also used the "string proof"; he took a sample of syrup by means of the proof-stick and, taking a small portion between thumb and index finger, he separated the fingers quickly; the right condition for graining corresponds to a string which breaks at a length of about one inch. If it is too soon, the string breaks as soon as the fingers are separated; if too late, it does not break.

Graining by the waiting method took place in the labile zone (*cf.* p. 631). Graining by shocking was done in the intermediate zone, the shock itself giving a rapid passage into the labile zone ($S = 1.60 - 1.70$). Graining by seeding takes place in the metastable zone, at about $S = 1.10 - 1.20$ for high-purity strikes, $1.15 - 1.18$ for lower-purity material. At the right moment the vacuum is reduced to 58.5 - 61 cm (23 - 24 in.), and the operator waits until the instruments (or the string proof) indicate afresh the correct concentration at this new temperature. The slurry of ground sugar is then introduced, generally by a cock provided for the purpose, which must of course discharge into the pan below the syrup level. This cock and attached tube should be of rather small diameter, in order that the quantity of air taken in with the slurry should not take the fine seed grain with it towards the condenser, through

the massecuite, in the form of fine bubbles. The cock will be opened cautiously to avoid this.

After the grain has been introduced, steam is shut off completely to avoid possible solution of the fine crystals by contact with the heating surface. After waiting a few minutes, boiling is then continued very gently, so as to allow the grain to grow. Further additions of syrup are not made until 30 - 60 min after seeding, and the vacuum is then gradually brought back to normal, in a further 15 - 30 min.

Since graining for a C massecuite is a long and critical operation, with risk of forming false grain, it is recommended that the operation be carried out in a small pan and in a medium of relatively high purity; this is best obtained by a mixture of syrup with a high-grade molasses. The B molasses is not introduced until the crystals have reached a reasonable size (40 - 50 microns). To avoid at that stage an excess supersaturation, the molasses is diluted by addition of water: 5 - 10 min in A and B massecuites, 2 - 8 min with C strike. This may also be done by reduction of the vacuum.

Preparation of the seed. The grain of the ground sugar slurry should be as uniform as possible, the aim being a size of 10 - 25 microns. **Knovl and Moller**⁴⁹ recommend a time of grinding of 8 hours at least, followed by a period in rotating containers of 24 hours, which reduces the grain to 5 or 8 microns.

Since the icing sugar used for preparing the grain has a great tendency to settle as a compact mass, this disadvantage can be avoided by adding 1% of tricalcium phosphate. The prepared slurry on its part has a tendency to settle, so it is advisable to use it promptly, or to keep only the necessary quantity, in a small bottle and for a short time only.

Quantity of seed to introduce. **Honig and Alewijn**⁵⁰ estimate (and we share their view) that best results are obtained when seeding is effected by means of a slurry of very finely powdered refined sugar. This slurry is prepared in a small ball-mill where the sugar is suspended in an organic liquid which does not dissolve sucrose, such as isopropyl alcohol, ethyl alcohol, petrol, etc. They recommend 160 g of sugar per 100 m³ (4.5 g/100 cu.ft.) of final C massecuite. The quantity obviously depends on the fineness of grinding of the seed grain and the size of final grain required. **Roger Bax**⁵¹, in Mauritius, reckons 530 - 565 g of sugar per 100 m³ (15 - 16 g/100 cu.ft.) of final C massecuite. **Belle Glade in Florida**⁵² gives 1.6 kg per 100 m³ /1 lb./1,000 cu.ft.).

The sugar for preparing the seed should be very dry. If moist, it forms small lumps. It is necessary to keep it in demijohns, in airtight tins or in stoppered bottles. The ball-mill is left to run for 3, 6, 12 or even 24 hours, in order to obtain the desired fineness of seed crystal.

(c) Building up the grain

After introducing the seed slurry into the pan, supersaturation continues to increase for 3 - 5 min. As soon as this time has elapsed it is necessary to commence the use of movement water, in order to maintain an evaporation rate sufficient to give good circulation. Use of movement water will be continued for 15 - 20 min until the crystals become visible to the naked eye; this practice avoids an increase of supersaturation which would involve formation of secondary grain.

As soon as the grain is large enough to be observed in a proof, the experienced sugar boiler

will know whether he has enough grain or too much. If he has too much he proceeds to dissolve a portion of it by the rapid introduction of a large charge of syrup. If he has insufficient grain, there is little remedy; he will be obliged to boil a large grain, which is not a great disadvantage for high-grade massecuites, but will lead to unsatisfactory exhaustion in a final massecuite. To avoid having an irregular grain, it is necessary to take particular care not to form a further crop of grain, otherwise crystals of two different sizes will be formed, which will be very detrimental to the fugalling.

The remainder of the strike consists particularly of developing the existing grain without forming false grain (that is to say, new grains which may form either crystals of a different size, rendering the fugalling very difficult, or a fine crystal which is liable to pass through the holes in the centrifugal gauzes and so enrich the molasses which is separated).

With this object, it is necessary to strive to maintain the greatest possible uniformity of operation, and to maintain constant both the vacuum and the steam pressure.

Any sudden increase in vacuum or any sudden fall in steam pressure causes a risk of forming secondary grain, due to the reduction in boiling temperature in the pan (causing a sudden passage into the labile or intermediate zone).

Apart from these two common causes, false grain may also be due to:

- (1) Rate of evaporation forced too high
- (2) Introduction of syrup or molasses at too low a temperature
- (3) Entry of air by leakage at the bottom door of the pan, if it is not tightly closed.

If false grain does form, the sugar boiler should notice this promptly, and proceed to dissolve it by:

- (a) Raising the temperature in the pan, by decreasing the vacuum
- (b) Introducing a large quantity of syrup or of water so as to dissolve the fine grain.

When boiling large-grain sugar, it may even be difficult to avoid false grain unless a very dilute syrup or even juice is available for use as pan feed.

If false grain is formed near the end of the strike, the high viscosity and the lack of circulation in the massecuite render it almost impossible to redissolve it (*cf.* Fig. 33.17).

The sugar boiler will not resume feeding the pan with syrup (A massecuite) or will not commence feeding molasses (B and C massecuites) until the crystals have attained a size such that they represent about one-third of the volume of the massecuite.

(d) Tightening the massecuite

Guided by experience, the sugar boiler introduces syrup (or molasses) into the pan at a rate sufficient (1) to compensate for the quantity of water evaporated, (2) to increase slowly and progressively the volume of the massecuite in the pan, in such a way that the concentration increases with the level of the massecuite, and so that the brix will approximate to the maximum value desired when the massecuite has reached the level corresponding to the full pan.

During the last phase of the strike, we have therefore in the pan a very tight and very viscous massecuite, circulation of which is very laborious. Concentration is continued to "tighten" the material to the maximum obtainable.

In order to maintain an acceptable circulation during this final phase of the strike, movement water will be introduced as was done after graining; this will be done during the last half hour or hour of the strike. As well as giving some improvement in the circulation, this

practice at the same time achieves a reduction in the supersaturation and so reduces the risk of formation of very fine false grain in this last critical period of the strike.

When it is observed that the crystals occupy the maximum space, and that the mother liquor occupies only the void volume remaining between the adjacent crystals, which are touching each other, the strike is "dropped". That is, the vacuum connection is shut off, the cock communicating to the atmosphere is opened, which breaks the vacuum with a characteristic whistle, and then the discharge door of the pan is opened to the crystallisers or mixers below.

When the pan is empty, steaming-out is commenced: a steam valve is opened to admit steam to the interior of the pan, leaving the bottom door open. The steam dissolves the sugar and cleans the vessel.

Exhaust steam will be used, as it is more effective and more economical than live steam. A more complete washing is obtained by using water sprays arranged round the upper part of the pan; the water should be hot and under pressure, well directed on the sides of the pan.

Actually, only the upper portion of the pan need be steamed out, since the part corresponding to the graining volume is automatically washed by the thin syrup introduced at the beginning of the following charge, while that syrup is being concentrated.

Feeding. There are two principal methods of feeding the syrup or molasses to the pan:

(a) Discontinuously by "drinks"

(b) Continuously by leaving the inlet valve set at a reduced but constant opening.

The latter method permits a constant supersaturation to be maintained, or allows the supersaturation to be increased in a progressive and regular fashion. It is much better than the former method, which gives poor or inferior results; it also gives appreciably faster boiling.

Automatic feeders give continuous feeding, the actuating valve opening more or less according to requirements.

Supersaturation. It is not possible to indicate precisely the supersaturation to be maintained, since it depends on the purity of the syrup or mother liquor, which itself varies in the course of the strike. However, since graining is carried out practically always on a syrup of 80–85 purity, it is possible to indicate typical figures. Although graining may be carried out by seeding in the metastable zone, it is generally carried out, whether by seeding or by shocking or by the waiting method, in the labile zone. The following supersaturations may therefore be recommended:

$$\begin{aligned} \text{(a) Graining:} & \quad 1.25 < S < 1.40 \\ \text{(b) Building up the grain:} & \quad 1.08 < S < 1.20 \end{aligned} \tag{33.44}$$

Webre⁵³ considers that, for C masseccutes, a normal supersaturation is 1.325, and may vary from 1.25 to 1.40.

Tests carried out in Australia⁵⁴ have shown that, for growing the crystals in A masseccutes, it is necessary to keep between 1.07 and 1.17, according to the zone of the pan involved, if it is desired to avoid formation of false grain.

There are two conditions which permit these values to be more easily maintained. These are:

- (1) Individual vacuum
- (2) Use of the cuitometer or other instruments, and automatic feed.

CENTRAL AND INDIVIDUAL VACUUM

For reasons of simplicity and economy, many factories used to concentrate the vacuum system of the factory in one part of the building, generally known as the "vacuum house", and the single installation was known as *central vacuum*.

The factory then has only one vacuum pump, only one barometric condenser, one pump for cold water and one pump for hot water from the condensers. The multiple effect and all the vacuum pans were then connected by vapour piping to this central condenser.

This installation was very practical from the mechanical point of view, and conducive to minimum cost. However, it had a serious disadvantage. If the multiple effect does not produce appreciable disturbances in the system, it is by no means the same with the vacuum pans. When syrup is introduced into one of the pans, and concentration is commenced, evaporation is very rapid (*cf.* p. 663); the evaporation and condensation of a quantity of vapour so much greater than normal constitutes such an additional load on the central vacuum system that the whole installation cannot but feel the effects. Sometimes the vacuum falls abruptly in this way from 66 to 61 cm (26 to 24 in.), for example. When by chance two pans commence a strike at the same time, the trouble is obviously more serious. Such sudden changes are obviously very detrimental to strikes in process in the other pans; we have just seen (*cf.* p. 683) to what extent it is important for sugar boiling and for fugalling to maintain a very uniform vacuum. A variation of 13 mm ($\frac{1}{2}$ in.) in vacuum, under certain conditions, is sufficient to cause formation of false grain.

A partial remedy may be obtained by furnishing the vacuum valve of each pan with a small pipe 50–75 mm (1–2 in.) in diameter by-passing the valve, and furnished itself with a small valve of the appropriate diameter. When a pan is restarted, the large vacuum valve is then left closed, and the vacuum is raised by means of the small by-pass. Thus the quantity of air to be handled and consequently the overload imposed on the central vacuum system is limited. When the vacuum in the pan has nearly reached its normal value, the main valve is reopened.

Webre⁵⁵ recommends the installation of a small vacuum system reserved specially for use when restarting pans.

These devices provide a solution for the pan connection, which, however, is incomplete, since when the vacuum valve is opened the pan is certainly at the same pressure as the vacuum system, but is full of air, and this is sufficient to exceed the capacity of the air pump. Further, when steam is opened into the calandria of the pan, the excessive evaporation load at the start of the strike remains too high and still has its influence on the whole vacuum system.

A slight improvement may be obtained by providing at each pan a separate condenser connected to the central air pump.

However, the only complete solution is the *individual vacuum system*. This consists of hav-

ing each pan, and similarly the multiple effect, entirely separate and providing for each a small individual vacuum system.

This solution obviously presents some disadvantages:

(a) Multiplicity of units (condensers and pumps).

(b) High cost. Not only are more units required, but further, the sum of the capacities of the individual condensers and pumps must be much greater than the capacity of the central condenser or pump, since each individual installation must be capable of evacuating the maximum quantity of air and of steam from the pan which it serves, whereas the central vacuum system is designed on the mean output of several pans, on the assumption that only one of these will be working under peak conditions, that is, at the beginning of a strike.

(c) Power consumption will be appreciably higher, since each pump is designed for maximum duty, and works during most of the time far below its nominal power consumption; this always causes a higher power consumption, especially for electric motors.

(d) Consumption of cooling water will be greater.

On the other hand, it has several advantages:

(a) It eliminates the enormous and cumbersome vacuum piping connecting the central condenser to the multiple effect and to all the vacuum pans. Each small condenser may be placed in immediate proximity to the pan which it serves.

(b) A breakdown of one unit involves a shutdown of only the pan concerned and can generally be repaired without shutting down the factory.

(c) Finally, the very great gain, which is the main object of the system, is that the operator of each pan can regulate its vacuum at will without affecting the operation of the other units.

When low-speed reciprocating air pumps were used, individual vacuum was really hardly practical, but with the introduction of jet condensers and particularly rotary vacuum pumps using a liquid annulus (such as the Nash type), the value of the system cannot *be* overemphasised. All of the A and B pans could be grouped under the same vacuum system, but the C pans at least should have individual vacuum equipment. Individual vacuum constitutes an essential condition for good operation of the factory; its installation is possible without serious alterations, and its higher cost is rapidly recovered by the improvement in efficiency of the factory and in quality of the sugar.

However, there is one case where central vacuum could regain preference. It is where the factory has continuous pans only for A, B and C strikes. At present such cases are very few, but the extension of continuous pans will inevitably render them more and more common. There is then no longer anything to disturb the vacuum, which can remain strictly uniform; and there is nothing against installing a central vacuum system, which is simpler and more economical.

INSTRUMENTS FOR PAN CONTROL (G.H.J.)

Since degree of supersaturation is the most important variable in sugar boiling, some type of instrument to indicate degree of supersaturation is a necessity for efficient operation. Instruments used are of two main classes: one uses boiling-point elevation of the material, while the other used electrical conductivity, as an indication of supersaturation. In either case, the

relationship between the property measured and the supersaturation of the mother liquor must first be established for the particular class of material concerned; and, with either method, the detecting elements must be correctly located to ensure accurate results.

Instruments based on boiling-point rise

These instruments measure the boiling-point rise (BPR) of the massecuite relative to that of pure water at the same absolute pressure. Since the boiling-point rise of a solution, for a given solute, is directly proportional to the concentration of solute, the supersaturation coefficient is given by the ratio of the boiling-point rise of supersaturated solution to that of a saturated solution at the same temperature. With pure sucrose, the relationship of boiling-point rise to degree of saturation is readily established; with impure solutions such as those treated in the raw sugar factory, the effect of the impurities must be ascertained for each type of material.

Attempts at using BPR as a measure of supersaturation were unsuccessful until Smith⁵⁶ in the 1930's established the conditions necessary for dependable readings. The main requirements are

- (a) Correct location of the thermometer elements in the massecuite
- (b) A dependable arrangement for measuring the corresponding vapour temperature.

Due to the effects of hydrostatic head, the temperature of the massecuite at any depth below the boiling surface is considerably higher than that at the surface; the latter is the true boiling point corresponding to the working vacuum. Smith accordingly placed the thermometer element in the downtake, on the reasoning that the temperature of the descending massecuite is the same as that at the surface, after the superheat due to hydrostatic head has been released by flash.

Measurement of the vapour temperature is best obtained by attaching a small pilot pan to the vacuum pan; water is boiled in the small pan, the vapour discharging into the vapour space of the main pan. The temperature in the small pan thus gives the boiling point of water at the vapour pressure in the pan, and the difference between the readings of the massecuite thermometer and the thermometer in the small pan thus gives the boiling-point elevation.

It is found in practice that the thermometer element in the downtake does not give an accurate measure of the temperature at the surface of the boiling massecuite. Since the massecuite circulation does not follow the ideal pattern depicted in Fig. 33.9, the superheat due to hydrostatic head is not entirely released at the surface, and the temperature of the descending massecuite is higher than that at the boiling surface. The resulting inflation of massecuite temperature thus increases as the level in the pan builds up; hence, to maintain a uniform supersaturation, an increasing BPR (as read by the instrument) must be used as the pan fills. With automatic control, a suitable cam can be used to set the apparent BPR to an increasing figure as the pan builds up.

Generally, supersaturation coefficients are not calculated for normal operation, but BPR itself is used as the controlling figure, *i.e.* the boiling is carried on at a specified boiling-point rise.

BPR - purity - saturation relationships. In order to determine supersaturation coefficient, it is necessary to establish the value of the BPR for a saturated solution of the composition

concerned and at the temperature in use. Holven⁵⁷ has established that the BPR at saturation increases with increasing temperature of the solution, and increases with decreasing purity. The increased BPR at lower purities is a result of the increased total solids content of a saturated solution of lower purity, relative to pure sucrose under similar conditions. Figure 33.20 shows the relationships for one type of material as given by Holven. Such graphs of course apply only for non-sugars of a particular composition, and the actual values will vary not only with purity of the mother liquor at a given factory, but with the nature of the non-sugars present.

Having established such relationships for the particular composition of material concerned, the supersaturation coefficient for such material in the pan is readily determined from the boiling-point rise relative to the established figure for that material at saturation.

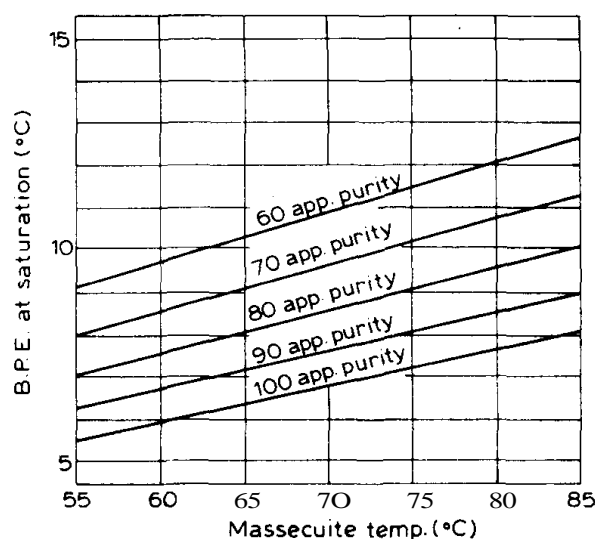


Fig. 33.20. Boiling-point elevation of mother liquor as a function of temperature. (Reproduced by permission of John Wiley and Sons.)

Effect of temperature variations. It is readily seen that the BPR is a function of the *concentration* of dissolved solids, and gives a measure of supersaturation only if temperature (and hence solubility) remains constant. If temperature varies, the BPR will remain essentially constant, but the solubility will alter and the supersaturation coefficient will change. Hence, if the BPR is to give a *reliable* measure of supersaturation, the absolute pressure or vacuum in the pan must remain constant; and it is found that automatic control of vacuum is an essential for satisfactory use of the BPR method.

Instruments. Resistance thermometers are generally used in modern installations as the most suitable type for continuous recording. The instruments are thus of **Wheatstone-bridge** type, registering the temperature difference in terms of the resistances of the thermometer elements. Indicating, recording and controlling instruments are used; and the feed of syrup or molasses to the pan is controlled, manually or automatically, to maintain the set value of BPR.

Conductivity instruments

Honig and Alewijn in Java⁵⁸, about 1927, considered using viscosity as an index of supersaturation, and chose electrical conductivity as the most convenient indication of viscosity. The method depends on the reduction of conductivity with increasing viscosity, due to the retarding effect of viscosity on the mobility of ions in solution. It was found that conductivity followed the supersaturation more closely than did refractive index, which was used to some extent at the time. With the development of robust electrodes for use in a pan, the method was adapted to practical use in the factory, and is now very widely used.

As with the BPR method, position of electrodes in the pan is important. They are generally placed below the calandria, or in the downtake, to obtain a sample of the material corresponding approximately to its temperature and concentration at the surface after it has released its superheat. The electrodes must be placed at a suitable distance from the wall of the pan, and from any other metal bodies such as coils or calandria, to avoid short-circuiting of the current to such metal. Dimensions of electrodes have been generally standardised, *e.g.* in Queensland, cylindrical electrodes of 16 mm (1 in.) diameter and 44 mm (1½ in.) long are used, spaced 75 mm (3 in.) apart between centres. The electrodes are suitably mounted for ready removal; this is important, as frequent cleaning is necessary to maintain consistent readings.

As with the BPR method, supersaturation coefficients are not generally calculated for ordinary operation; conductivity figures (often expressed simply as current in mA at a standard voltage) are used as the basis for control. Establishment of the correct values, of course, requires prior calibration for each type of massecuite, and for differing compositions of the non-sugars concerned.

Conductivity — viscosity — supersaturation relationships. The relationship of conductivity to supersaturation may be summarised in general terms by stating that the viscosity of a molasses follows the supersaturation fairly closely for variations in either concentration or temperature within the normal range of pan operations. As specific resistance is approximately proportional to viscosity, the resistance in turn follows the supersaturation coefficient fairly closely, *i.e.* there is an inverse relationship between conductivity and supersaturation coefficient. It will thus be seen that, by fortunate relationships between different properties, the conductivity gives a good indication of supersaturation even with slight variations in temperature. For this reason, the conductivity method, unlike the BPR method, is not seriously affected by moderate changes in vacuum.

The conductivity reading of a massecuite is influenced by the presence of crystal, since the crystal is essentially a non-conductor; the volume occupied by the crystals causes a decrease in the volume of conducting material between the electrodes. Thus, with an increasing crystal content as the charge progresses, a constant conductivity reading means a lower supersaturation coefficient; on the other hand, the decreasing purity of the mother liquor means a higher concentration of electrolytes, with some compensating effect on the conductivity.

Instruments. Conductivity can be measured by a very simple instrument; for example, by connecting an a.c. supply (at 30 V for normal raw sugar massecuites) to the electrodes, and reading the current by a milliammeter. Provided the voltage is steady, this gives reliable readings. Mains frequency of 50 cycles is satisfactory, though higher-frequency supply would

give greater accuracy. The British "Cuitometer" (Fig. 33.21) is a refinement of this arrangement, with a stabilised voltage supply and a choice of voltage over a considerable range, for use with different massecuites. More sophisticated instruments such as the Kent Multelec use a Wheatstone bridge and recording potentiometer, and give a reading in resistance rather than conductivity; these are extensively used to operate automatic control of the syrup or molasses, thus controlling to a set conductivity (or resistance) figure. Kelso⁵⁵ has developed a simple and effective controller, consisting of an extension of the simple instrument reading current in mA for a given applied voltage; this instrument, with suitable proportioning of feed pipes and valves, gives satisfactory control with a simple on-off method.

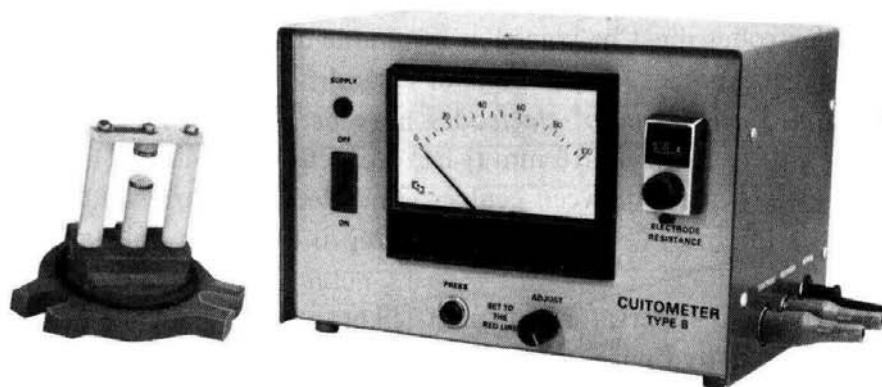


Fig. 33.21. Cuitometer (Sumasuco).

Other instruments. Ditmar Jansse⁶⁰ has developed an instrument based on heat transfer as an indication of supersaturation. Since heat transfer depends mainly on the viscosity of the massecuite, this instrument also uses viscosity indirectly as a measure of supersaturation.

Refractive index, read by a refractometer specially designed for use on the pan, was one of the earlier properties used to indicate supersaturation, and has been revived in recent years to a limited extent. Since refractive index gives a measure of concentration, it indicates supersaturation only if temperature is kept constant, and in this respect it resembles the BPR method.

Importance of instruments

The use of instruments to indicate supersaturation has provided a scientific basis for control of the sugar-boiling process, in place of the earlier "mysterious art" of the sugar boiler. This in turn has led to a vast improvement in sugar quality (particularly in uniformity of crystal size), and in effective capacity of pans and centrifugals; and such improvements have resulted in a substantial improvement in sugar recovery. Modern standards of performance would indeed be impossible without such instruments.

Recent developments

In a review of recent developments, Wright^{60c} comments that conductivity measurements are particularly suitable for control of raw sugar strikes with purities below 95, provided the

inorganic ash level of the raw material feed does not fluctuate rapidly. Improved instruments for conductivity measurement have been developed and conductivity methods are generally preferred for raw sugar boilings. Boiling point elevation methods are less used because of superheat interference with measurements, while consistency (rheology) measurements are used mainly for high-purity boilings.

Automatic control of boilings, and of pan stage operation in general, is used to an increasing extent (see chapter 46).

OTHER FACTORS

Grain size

The size of the crystals should be inversely related to the viscosity of the mother liquor.

The growth of the crystal takes place by attachment of sugar from solution on the surface of the crystal. Now:

(7) The smaller the grain, the greater is the total surface area of the crystals.

(2) The smaller the grain, the smaller is the average distance to be travelled by any molecule to reach the surface of the nearest crystal. Now, the more viscous the massecuite, the more difficult it will be for each molecule to reach the immediate neighbourhood of a crystal. For this reason it is necessary to strive to increase the crystal surface, and to decrease the average distance between crystals.

The object, therefore, will be to boil C massecuite with the smallest grain size possible, consistent with requirements of fugalling. In Hawaii, the general grain size for C massecuites is 0.2–0.3 mm; generally, the grain size is kept at 0.3 mm or between 0.25 and 0.35 mm, and it is not desirable to exceed 0.35 mm.

For the A and B massecuites, the grain size is less important. For B massecuite it is generally kept about 0.6 mm, and between 0.8 and 1 mm for A sugar.

The grain size also depends on the system of boiling. In general, the following sizes (mm) are sought at the discharge of the pan:

	<i>Footing</i>	<i>A</i>	<i>B</i>	<i>C</i>
CB/CA system:		0.50-0.60	0.50-0.60	0.30-0.40
CBA system:		0.55-0.65	0.40-0.45	0.25-0.30
Footing C and BA:	0.35	0.70-0.80	0.45-0.50	0.40-0.45

Brix

In the absence of false grain, the purity of the mother liquor varies in inverse relationship with the brix of the massecuite. Thieme⁶¹ has found, for example, the relative variations shown in Table 33.13.

It is necessary therefore to strive to boil the massecuites as heavy as possible, and particularly so for the final massecuite. A brix of 96° is easily attained, sometimes 98° and even 101° (97–98° refractometric).

However, the "heaving" or "tightening" cannot be obtained when the pan is already full or nearly full. A good sugar boiler should maintain a maximum brix and maximum super-

saturation (the cuitometer allows this to be done without risk), and not wait until the pan is full to tighten up the massecuite; on the contrary, he should conduct the boiling in such a way that the brix progresses with the filling, and that the brix reaches the desired value at the time when the level of the massecuite reaches the middle of the top sight glass.

The brix values generally obtained are:

	<i>Densimetric brix</i> (1 : 5 dilution)	<i>Refractometric brix</i>
A massecuite:	92- 96	91 – 93
B massecuite:	94- 97	92 – 95
C massecuite:	97 – 101	95 – 97

Chen⁶² recommends a brix of 97 for C massecuite. Below that figure, the purity of the molasses increases by 0.72 per unit of brix below 97.

TABLE 33.13

VARIATION OF MOLASSES PURITY WITH BRIX OF MASSECUIE

<i>Brix of final massecuite</i>		<i>Purity of mother liquor when dropped</i>
<i>Densimetric</i>	<i>Refractometric</i>	
97	93	44
98	94	42
99	95	40
100	96	38

Vacuum. Caramelisation, inversion and colour formation are proportionally more marked with the higher temperatures. It follows that, when making white sugar, it will be necessary to utilise a correspondingly lower steam pressure and to maintain in the pan a correspondingly higher vacuum when a better colour of sugar is required.

Vacuum is the factor which has the greatest influence on the massecuite temperature in the pan. Ditmar Jansse⁶³ considers that the most favourable temperatures for A and B massecuites are between 65 and 74°C (150 and 165°F); colour formation and undesirable reactions causing decomposition are then minimal and the crystallisation rate is satisfactory. For C massecuites, the temperature will be kept preferably in the neighbourhood of the lower limit (65 °C).

Crystal surface. The crystal surface area is given as a function of the length of the crystal, measured under the microscope, by the equation⁶⁴:

$$s = \frac{0.00421}{l} \quad (33.45)$$

s = surface area of crystals, in m²/g

l = length of crystal, in mm.

This equation enables an estimate of the crystal surface to be obtained, at a given moment in the strike, as a function of the linear dimension of the crystal.

Crystal content of massecuite

The fluidity or inversely the consistency of a tight massecuite depends essentially on its crystal content. This content may be expressed in weight, either as per cent of massecuite or per cent of brix; in the latter case it is advisable to make the meaning clear by specifying “% on brix”, since these two values are very close together when the brix is in the neighbourhood of 100.

Gillett and Kenda⁶⁵ consider that tightening the massecuite scarcely influences the fluidity until the crystal content reaches 43% by weight on brix, and consider that, for most factories in Hawaii, the optimal concentration of the C massecuites should be between 45 and 48% on brix. They consider that a difference of 1% around this optimum, assumed determined accurately (*e.g.* 47%), leads to an increase of 1.3 – 1.4 in the purity of the molasses.

Honig⁶⁶ considers that the crystal content of massecuite generally remains in the neighbourhood of 30 – 36% in C massecuites at the moment of dropping. This content obviously increases in the crystalliser and reaches 40 – 45% as a final figure. Payne⁶⁷ considers that a good compromise corresponds to 35 – 40% and that 45% is a maximum, the viscosity becoming absolutely prohibitive above that value.

In Australia, for A massecuites, the crystal content is maintained in the neighbourhood of 35 – 40% during the strike, and increased quickly to about 50% before dropping. It is considered that, during the strike, a crystal content of 50% is excessive and conducive to false grain⁶⁸. 30 – 35% is considered as a light massecuite, 40 – 50% as a tight one.

Kelly⁶⁹ has shown that the viscosity increases sharply when the crystal content reaches 44%, a value at which the crystals can no longer move independently, and that at 48% the massecuite would completely lose fluidity. It is thus necessary to keep a practical value of 40-45%.

Bauserman⁷⁰ recalls that the void volume, between sugar crystals in random arrangement, represents about 75% of the true crystal volume. He considers that a massecuite containing 46% of crystals by volume is still of a practical consistency. This would have therefore:

Crystal volume:		46%
Volume of voids between crystals:	$46 \times 0.75 =$	34%

Apparent volume of crystals:		80%
Surplus liquid for centrifuging:	$100 - 80 =$	20%

and he considers that it is found in practice that an optimal massecuite is one having a surplus of 20% when supplied to the centrifugals, or as measured in Gillett's laboratory apparatus, which is a very useful aid to correct practice in sugar boiling. He accordingly recommends a massecuite of 46% crystal before centrifuging.

In Mauritius, Lamusse and Wiehe⁷¹ reckon crystal contents as percentage on brix of the order of:

	<i>As dropped</i>	<i>Leaving crystalliser</i>
A massecuite:	40-60%, average 48%	45-65%, average 56%
B massecuite:	30-45%, average 36%	35-50%, average 42%
C massecuite:	25-40%, average 32%	30-45%, average 37%

or about 116 leaving the crystalliser for 100 leaving the pan.

Calculation of crystal content of massecuite. The crystal content is readily determined in the laboratory. It may also be calculated in the following manner. Let:

$$\begin{array}{ll}
 B_m = \text{weight of solids \% massecuite} & B_e = \text{weight of solids \% mother liquor} \\
 S_m = \text{weight of sugar \% massecuite} & S_e = \text{weight of sugar \% mother liquor} \\
 P_m = \text{purity of massecuite} = \frac{S_m}{B_m} \times 100 & P_e = \text{purity of mother liquor} = \frac{S_e}{B_e} \times 100 \\
 x = \% \text{ crystal on massecuite} & P_s = \text{purity of sugar.}
 \end{array}$$

We have (*cf.* p. 839):

$$\frac{x}{S_m} = \frac{P_s (P_m - P_e)}{P_m (P_s - P_e)}$$

Hence:

$$x = P_s \frac{B_m}{100} \frac{P_m - P_e}{P_s - P_e} \quad (33.46)$$

This is the same expression as the S.J.M. formula.

If it is assumed that the crystals consist of pure sucrose ($P_s = 100$) and if the crystal content is required not as % massecuite but as % on brix, say x_0 , we will have:

$$x_0 = \frac{x}{B_m} = \frac{P_m - P_e}{100 - P_e} \quad (33.47)$$

which could be calculated even more conveniently by writing that, % brix of mother liquor, the purity of the massecuite is equal to the sucrose content of the molasses plus that of the crystals, divided by the total solids:

$$P_m = \frac{P_e + x_0}{100 + x_0} \quad \text{hence: } x_0 = \frac{P_m - P_e}{100 - P_e}$$

Inversion losses

Experiments by King and Jison⁷² have shown that inversion losses were very low at 50 - 60°C (122- 140°F), became perceptible at 70° C and considerable above 80° C (176°F). These tests were made with A molasses of 54 apparent purity and 6.0 pH.

Cleaning vacuum pans

Vacuum pans accumulate much less incrustation than do evaporators. Many factories operate for a whole season without cleaning the pans. However, an excellent practice in certain countries is to boil out the pans with water for a quarter of an hour at the end and at the middle of the week, and so maintain them in a very high state of cleanliness⁷³.

Disposition of C sugar

The sugar obtained from the low-grade centrifugals handling the C massecuite is generally brown, pasty and impregnated with molasses, since the last strike is not washed. It cannot be sold in this state. We have therefore the choice of two ways of using it:

(1) **Remelting.** Firstly the sugar may be remelted. For this purpose some clarified juice is pumped to the mixer which receives the C sugar below the centrifugals. This mixture may then be taken by means of a rota pump and sent to a mixing tank situated near the effect supply tank, where it is redissolved in clarified juice. It is thus sent back into manufacture, between the clarifiers and the multiple effect.

This method has the disadvantages:

(a) Of placing an extra load on the whole boiling house, by re-circulating through it in the form of solution sugar which had already been crystallised.

(b) Of increasing the brix and sometimes decreasing the purity of the clarified juice entering the evaporator; and this adversely affects the average heat-transfer coefficient in the evaporators.

(2) **Utilisation as footing.** The best practice is that devised by Spencer⁷⁴ and explained by Solano and Villalobos⁷⁵ in an excellent article which we recommend to the reader. It consists of using the C sugar as grain or footing for A and B massecuites. The procedure is thus as follows: the C sugar is remixed in the magma mixer placed below the C centrifugals by adding syrup, clear juice, or even water. The magma thus formed is taken up by a pump and transferred to a crystalliser – mixer, preferably under vacuum, situated on the pan floor; this serves as a storage vessel and also as a means of keeping the magma mixed to uniform consistency. The A and B footings are then drawn from this mixer.

This method presents several advantages:

(a) It eliminates all recirculation at evaporators and pans, since the C sugar when once crystallised is not remelted in juice. Hence the capacity of the multiple effect, the A and B pans, the B mixers and the A and B centrifugals, is increased.

(b) The C sugar, consisting of small crystals, provides an excellent grain on which to build up the A and B massecuites.

(c) The sugar boilers handling A and B massecuites are relieved of the delicate operation of graining. There is less risk of producing false grain or uneven grain. They start, on the contrary, with a good grain, relatively uniform. Hence an improvement in the pan work, on A and B massecuites.

(d) The A and B strikes are boiled more rapidly, since they commence after the graining phase with a grain already comparatively large. Hence there is a further gain in capacity for the A and B pans.

In this case the C masseuites must be boiled more carefully; they must be grained by seeding and care must be taken to obtain a good homogeneous and uniform C sugar.

This "simple magma system" has, however, one disadvantage, *i.e.* it leads to a commercial sugar of inferior refining qualities, since the C grain always contains some impurities which remain included in the nucleus of the commercial sugar crystals produced from it. Also, C sugar from continuous centrifugals suffers breakage and does not form good grain.

Magma

The magma purity should be watched carefully: if too low, of the order of 80, it causes an undesirable return of molasses to the process. On the other hand, if a high purity is obtained at the expense of the C sugar recovery in the centrifugals, it is too high. Generally, a purity about 85 is a good compromise where single purging is employed, or about 88 with double purging.

Moreover, there need be no objection to a higher purity, if this purity is obtained without washing the sugar, and if the C masseuite has been boiled good and tight (*i.e.* with high crystal content). There is good reason to be well satisfied with a C sugar of higher purity, and thus of a higher-purity magma. When circumstances are favourable (low viscosity and regular grain), a magma purity of 90 may be attained or exceeded. Stevenson⁷⁶ in Australia considers that the highest purity practically obtainable is 92 - 93 (pol/refractometric brix), and points out that a high magma purity has a favourable influence on sugar quality (reduction of ash content) and filterability.

Proportion of crystal in the magma. When mixing magma, it is desirable to add the quantity of syrup necessary to render it sufficiently fluid for pumping, but no more. This is well judged by eye, according to the consistency of the magma. For calculation purposes the quantity may be reckoned on about half syrup (or molasses) and half crystals. In other words, the aim should be to reconstitute the masseuite which has just been fugged by replacing its molasses content by syrup, but using the minimum quantity of syrup compatible with satisfactory operation of the magma. It is particularly advisable to arrange automatic addition of syrup or molasses⁷⁷.

TABLE 33.14

MOLASSES RECYCLED AS A FUNCTION OF C SUGAR PURITY

<i>Purity of C sugar</i>	<i>Crystal % sugar</i>	<i>Mo/asses % sugar</i>	<i>Molasses % crystal</i>	<i>Molasses recycled % mass. at 45% crystal</i>
80.1	66.3	33.7	50.9	$0.45 \times 50.9 = 22.9$
88.1	82.5	17.5	21.2	$0.45 \times 21.2 = 9.5$
92.2	90.3	9.7	10.7	$0.45 \times 10.7 = 4.8$

Hessey and Manning⁷⁸ and Jensen⁷⁹ have made some interesting comments on the composition of magma and C sugar. In the Australian conditions considered, the proportion of molasses was related to the purity of the C sugar as shown in Table 33.14.

By interpolating these values, and remembering that massecuite with 45% of crystal obviously contains 55% of molasses and that, for a given massecuite and a given molasses, the quantity of molasses eliminated should be the same, we obtain the figures of Table 33.15.

TABLE 33.15

MOLASSES RECYCLED AS A FUNCTION OF C SUGAR PURITY

<i>Purity of C sugar</i>	<i>Crystal % sugar</i>	<i>Molasses % sugar</i>	<i>Molasses % crystal</i>	<i>Molasses recycled % mass. at 45% crystal</i>	<i>Molasses eliminated % mass.</i>	<i>Relative quant. of C mass. produced</i>
80	66.1	33.9	51.3	23.1	55 - 23.1 = 21.9	188
85	76.4	23.6	30.9	13.9	55 - 13.9 = 41.1	100
90	86.2	13.8	16	7.2	55 - 7.2 = 47.8	86
92	90	10	11.1	5	55 - 5 = 50	82

This table demonstrates the influence of C sugar purity (if the magma is made with water) or of magma (if made with juice, syrup or molasses) on the quantity of C massecuites. It also shows the importance of ensuring that the magma purity does not descend below 85.

Double-magma process. The boiling system just described was devised by Spencer and is described as the single-magma process: the C sugar, remixed as magma, serves as footing for the A and B massecuites, fuggalling of which furnishes the commercial sugar, which is a mixture of A and B sugars.

This concept may be pursued further by operating the double-magma process, which has developed logically from the single-magma system and has been formerly described formally by Carsolio⁸⁰. It consists of forming a C magma by remixing the C sugar with high-grade molasses, syrup, clear juice or water, and using this magma as footing for the B massecuites. A B magma is formed in the same way by remixing the B sugar with syrup, juice or water, and using this magma as footing for the A massecuites. The end result is that the commercial sugar is A sugar only (Fig. 33.22); the advantages are:

- (a) A better-quality sugar, since it originates only from A massecuites.
- (b) A minimum recycling of A and B molasses. This is the great advantage of the process: the massecuite viscosities are minimal, the massecuites are freer, and there is a favourable effect on recovery.
- (c) Reduction of graining to a minimum. Only C massecuites are seeded (an advantage in common with the single-magma process).
- (d) In the same way, some economy of steam. Since there is no recycling, heating and boiling the same molasses twice is avoided.
- (e) There is obviously more A massecuite, and slightly increased capacity in pans, mixers and A centrifugals is required, but the difference is of little magnitude, since the strikes build up better and more rapidly and are fuggalled more readily. Capacity of pans, mixers and B centrifugals is improved. We may reckon:

A massecuite: 127 – 153 dm³/t.c. (4.5-5.4 cu.ft./t.c.)

B massecuite: 51 – 62 dm³/t.c. (1.8-2.2 cu.ft./t.c.)

C massecuite: 28- 42 dm³/t.c. (1 – 1.5 cu.ft./t.c.)

Generally, there will be an excess of C and B sugars. The excess C sugar could be remelted or utilised for part of the A massecuites. The B centrifugals may be divided so as to produce only the quantity of sugar necessary for the A footings, thus remixing part under the B centrifugals, and diverting the remaining B centrifugals to the conveyor, so that this sugar is mixed with the A sugar and goes out as commercial sugar. The first portion of the B centrifugals would not receive any washing; the second could be washed.

This combination furnishes an economic and elegant method of operating the sugar boiling.

Formation of magmas. The following is a summary of the CBA double-magma process:
 A massecuite run up with standard liquor (syrup plus remelt of excess of B and C magmas)
 Remelt of B and C magmas carried out with clarified juice
 B massecuite run up with A molasses plus molasses from affination of C sugar
 C massecuite grained from syrup and run up on B molasses
 B magma made up with standard liquor
 C magma before affination carried out with syrup
 Affined C sugar remixed with syrup
 Size of crystals: C footing, 0.17 mm; A sugar, 0.8 mm.

A and B massecuites. In the single-magma system, the A and B sugars are generally despatched for sale. For these commercial sugars the objective will be a moderate grain size of 0.7 - 1 mm. In order to obtain a grain of this size, starting from a footing of C sugar of 0.35 mm, the final volume of massecuite should be about 9 times the volume of magma made from C sugar, which is used as footing:

$$0.35 \sqrt[3]{9} = 0.35 \times 2.08 = 0.73 \text{ mm}$$

To achieve this, the following simple procedure may be followed. Assuming an A or B pan of 30 m³ (1,050 cu.ft.), the graining volume of which is 10 m³ (350 cu.ft.):

- (a) Take from the magma mixer (see above, "utilisation as footing") a footing of 10 m³
- (b) Build up this footing to 30 m³ with syrup
- (c) Transfer 20 m³ of the massecuite obtained to a seed storage vessel for A and B massecuites
- (d) On the remaining 10 m³ in the pan, boil an A or B massecuite
- (e) In the same way, utilise the two other footings each of 10 m³ which have been stored in the seed vessel, either in the pan concerned or in another pan
- (f) If there is too much magma, remelt it and send it to the syrup tanks; if, on the contrary, there is not enough, decrease the initial grain size of the C sugar, and increase the quantity cut over to the seed vessel.

Graining on molasses

We have seen ("Concentration", p. 681) that graining can be done either on syrup or on a mixture of syrup and an appropriate grade of molasses. The latter method has been introduced by Webre⁸¹ and consists of graining on a mixture of syrup and molasses proportioned to

obtain a purity of 70 – 74; Honig and Alewijn⁸² suggest 72 – 78. In Queensland, a mixture of 4 parts A molasses to 1 of syrup is used, which in that country corresponds to a purity of 70 – 75. The grain obtained is more uniform, conglomerates do not form at those purities, and the final purity drop is somewhat greater, since the strike commences from a footing of lower purity.

Syrup inlet to the pan. Previously, discharge of syrup into the pan was made by a pipe terminating abruptly in the centre near the bottom. The syrup, with a density of about 1.30, entering a massequite of density about 1.47, tended to rise in large globules to the surface where it spread as a sheet, mixing imperfectly and slowly with the massequite. To avoid this disadvantage, modern pans are provided with a distributing pipe in the form of a star, or a ring pipe provided with several outlets or small radial branches.

Storage tanks for syrups and molasses

The final molasses leaving the centrifugals is sent direct to the distillery or for sale elsewhere. The A and B molasses, on the other hand, must be collected, mixed, and stored in tanks while waiting to be returned into the B and C massequites.

Similarly, a certain syrup storage is required between the evaporators and pans. Capacities recommended by Salinas⁸³ in Cuba and average figures reported for Puerto Rico⁸⁴ are:

	<i>Puerto Rico</i>	<i>Cuba</i>
For syrup:	615 l/t.c.h. (135 Imp.gal./t.c.h.)	820 (180)
For A and B molasses:	1,225 l/t.c.h. (270 Imp.gal./t.c.h.)	1,000 (220)

Treatment of A and B molasses. The storage tanks for these molasses should be provided with:

- (1) Either a coil for compressed air, or preferably a stirrer such as a ship's propeller

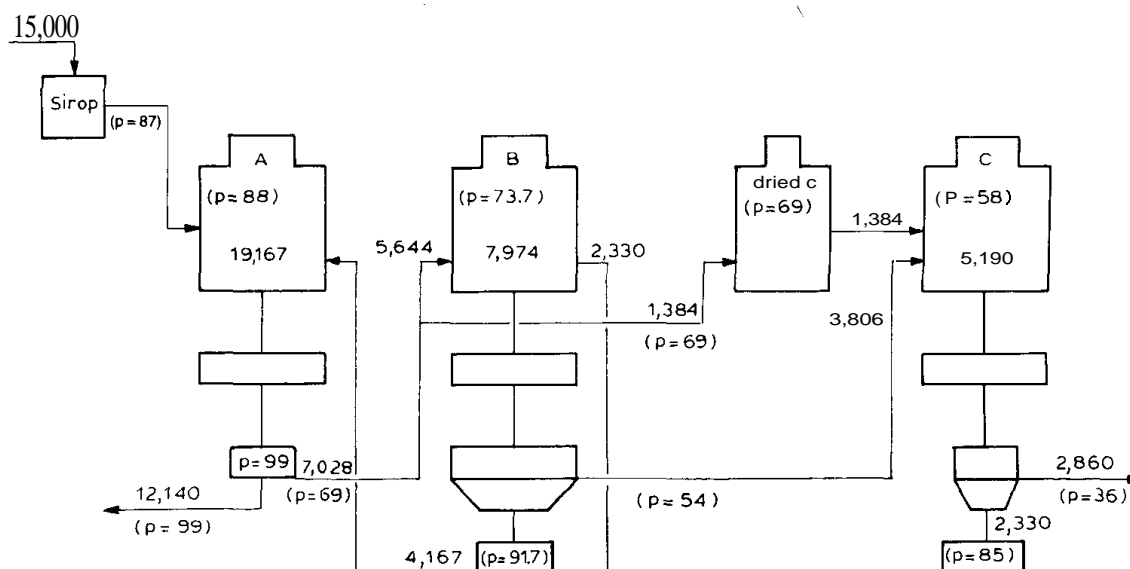


Fig. 33.22. Exhaustion diagram, CBA process.

(2) A steam coil, or other means of heating.

This permits of:

(a) Diluting these molasses by means of hot water, which will preferably be condensate from the later vessels of the multiple effect

(b) Dissolving completely any crystals which may be present in the molasses and which if introduced into the pan would cause irregular grain size

(c) Mixing thoroughly, homogenising and controlling the diluted material to a very uniform brix and temperature, which will assist and improve the strike.

As brix of these diluted molasses a figure of 70–75° will be adopted, and a temperature a few degrees above that of the massecuite in the pan, say 75°C (165°F) as a minimum. (Hence a mnemonic for remembering easily the suitable values of these two quantities: 75/75.) These storage tanks will be furnished with an indicating thermometer with a large dial; the tanks will be lagged, and also the piping connecting them to the pans.

Specific heat of molasses. The specific heat of molasses may be obtained with somewhat better precision than the simple formula (30.1), by means of the formula of Janovsky and Archangelsky⁸⁵:

$$c = 1 - [0.6 - 0.0018t + 0.0008 (100 - P)] \frac{M}{100} \quad (33.48)$$

$$(c = 1 - [0.6 - 0.001 (t - 32) + 0.0008 (100 - P)] \frac{M}{100})$$

c = specific heat of a molasses, in kcal/kg (B.T.U./lb.)

t = temperature of the molasses, in °C (°F)

P = true purity (sucrose/dry substance) of the molasses

M = dry substance % in the molasses.

Specific heat of crystal sugar

Between 40 and 80°C, this is given to a close approximation by $c = 0.283$ (cf. eqn (36.3)).

Proportions of syrup and molasses

The fabrication superintendent has continually to solve the following problem:

Problem. It is desired to obtain a massecuite of purity P_m by means of a footing (syrup or massecuite) of purity P_s and a returned molasses of purity P_q . What are the respective proportions of syrup and molasses to use?

Solution. Let:

Q_s = weight of syrup required

Q_e = weight of molasses required

Q_m = weight of massecuite obtained

B_s = weight of dry substance % of syrup

B_e = weight of dry substance % of molasses

B_m = weight of dry substance % of massecuite.

Equating the quantities of dry substance (d.s.) and of sugar contained in the syrup and in molasses to the quantities in the massecuite:

$$\begin{aligned} Q_s B_s + Q_e B_e &= Q_m B_m \\ Q_s B_s P_s + Q_e B_e P_e &= Q_m B_m P_m \end{aligned}$$

Eliminating $Q_e B_e$ between these two equations, we have:

$$\begin{aligned} Q_e B_e &= Q_m B_m - Q_s B_s \\ Q_s B_s P_s + (Q_m B_m - Q_s B_s) P_e &= Q_m B_m P_m \\ \frac{Q_s B_s}{Q_m B_m} &= \frac{P_m - P_e}{P_s - P_e} \end{aligned} \quad (33.49)$$

where the first member represents the d.s. of the syrup % of the d.s. of the massecuite. Hence:

$$\frac{Q_s}{Q_m} = \frac{D_m}{B_s} \cdot \frac{P_m - P_e}{P_s - P_e} \quad (33.50)$$

where the first term represents the weight of syrup % of weight of massecuite.

By volume, we shall have:

$$\frac{V_s}{V_m} = \frac{d_m B_m}{d_s B_s} \cdot \frac{(P_m - P_e)}{(P_s - P_e)} \quad (33.51)$$

In the same way, we have:

$$\frac{V_e}{V_m} = \frac{B_m d_m}{B_e d_e} \cdot \frac{(P_s - P_m)}{(P_s - P_e)} \quad (33.52)$$

V_s = volume of syrup

V_e = volume of molasses

V_m = volume of massecuite

d_s = density of syrup

d_e = density of molasses

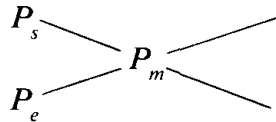
d_m = density of massecuite

Equations (33.51) and (33.52) give:

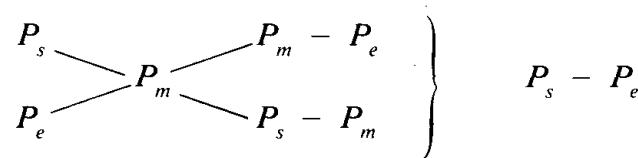
$$\frac{V_s}{V_e} = \frac{B d}{B_s d_s} \cdot \frac{(P_m - P_e)}{(P_s - P_m)} \quad (33.53)$$

We may comment that the volumes given by these last three equations do not indicate the level at which feeding of the respective materials into the pan is commenced, since the materials become concentrated as the charge proceeds. The figures correspond to the true volumes of each product before introduction into the pan.

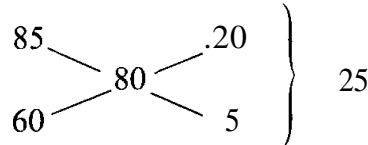
Calculation of eqn. (33.49) may be made very rapidly by the following device. The three quantities for the data P , P_e and P_m are placed in the form of a cross as shown:



and the differences are inserted in the diagonal directions. We thus have



and addition of the two differences shown on the right gives the total ($P_s - P_e$) of the proportions sought, the proportion of syrup ($P_m - P_e$) being shown on the level with P_s , and below, on the same line as P_e , the proportion of molasses ($P_s - P_m$). If, for example, we have: $P_s = 85$, $P_e = 60$, $P_m = 80$, we obtain



or 20/25 syrup and 5/25 of molasses (in terms of dry solids).

Quantitative scheme for pan station

To demonstrate the method of calculation for the pan station as a whole, we give a very simple example: the three-massecuite system, CBA. The method of calculation and the reasoning used apply to other, more complex cases which may be encountered.

We assume a factory crushing 100 t.c.h., in the following conditions:

	<i>Brix</i>	<i>Purity</i>
Clarified juice:	15	87
Syrup:	68	87
Molasses:	85	36
Commercial sugar:	99.2	99
B sugar:	96	92
C sugar:	94	85

	<i>Massecuities</i>		
	<i>A</i>	<i>B</i>	<i>C</i>
Brix	93	94	97
Purity	88		58
Purity drop	19	20	22
Purity of molasses	69		36
Litres per t.c.h.	140	58	37

We assume the conditions for the A and C massecuities, but not those for the B, which if assumed may be incompatible with the others. To proceed to the calculation commencing from the syrup and the molasses; from the molasses we pass to the C massecuite and its footing, then to the C magma, and revert to the A massecuite. We then assume a probable purity for the B massecuite, and repeat the calculation, modifying the figure if this leads to a quantity or purity drop which is abnormal. Undetermined losses in the manufacture are neglected.

The whole calculation is made in terms of dry substance. We do not pass to normal quantities by means of brix until the table is complete and satisfactory. We calculate previously the quantities of dry substance in massecuite, including B massecuite, to ensure that it will be approximately normal.

The calculation is made with the aid of the Cobenze diagram (p. 702), to resolve the problems of mixtures and proportions.

Mixtures. If we take two products A and B of respective purities P and p , and mix q_A of A and $1 - q_A$ of B, the purity P_m of the mixture is given by:

$$q_A = \frac{P_m - p}{P - p} \quad \text{and:} \quad q_B = \frac{P - P_m}{P - p}$$

or:

$$q_A P + (1 - q_A) p = 1 \times P_m \quad \text{and:} \quad (1 - q_B) P + q_B p = 1 \times P_m$$

Massecuite dry substance. The following are the normal values which we aim to maintain approximately:

C massecuite: 37 l/t.c.h. = 37 x 1.47 or 54.4 kg/t.c.h. = 54.4 x 0.97 = 52.8 kg d.s./t.c.h.
 B massecuite: 57 l/t.c.h. = 57 x 1.47 or 85.3 kg/t.c.h. = 85.3 x 0.94 = 80.1 kg d.s./t.c.h.
 A massecuite: 140 l/t.c.h. = 140 x 1.47 or 205.8 kg/t.c.h. = 205.8 x 0.93 = 191.4 kg d.s./t.c.h.

Sugar and molasses. The syrup contains: 100,000 x 0.15 = 15,000 kg of d.s., which gives (eqn. (39.4)) (Fig. 33.22):

$$15,000 \times \frac{87 - 36}{99 - 36} = 15,000 \times 0.807 = 12,140 \text{ kg of d.s. in sugar}$$

and:

$$15,000 \times 0.193 = 2,860 \text{ kg of d.s. in the molasses}$$

C massecuite. Quantity of d.s. in C sugar:

$$\begin{array}{ccc} 85 & \diagdown & 22 & \text{C sugar} \\ & & 58 & \\ 36 & \diagup & \frac{27}{49} & \text{molasses} \end{array}$$

$$2,860 \times \frac{27}{49} = 2,330 \text{ kg/h}$$

Total d.s. in C massecuite:

$$2,860 \times \frac{49}{27} = 5,190 \text{ kg/h}$$

Composition of C massecuite. Dry substance in C footing:

$$\begin{array}{ccc} 69 & \diagdown & 4 & \text{A molasses} \\ & & 58 & \\ 54 & \diagup & \frac{11}{13} & \text{B molasses} \end{array}$$

$$5,190 \times \frac{4}{13} = 1,384 \text{ kg/h}$$

Dry substance in B molasses:

$$5,190 \times \frac{11}{13} = 3,806 \text{ kg/h}$$

A massecuite. Dry substance in A molasses:

$$\begin{array}{ccc} 99 & \diagdown & 19 & \text{A sugar} \\ & & 88 & \\ 69 & \diagup & \frac{11}{30} & \text{A molasses} \end{array}$$

$$12,140 \times \frac{11}{30} = 7,028 \text{ kg/h}$$

Remaining in A molasses after taking for C footing:

$$7,028 - 1,384 = 5,644 \text{ kg/h}$$

Total quantity of B massecuite:

$$2,330 + 5,644 = 7,974 \text{ kg/h}$$

We now check that 2,330 is an acceptable proportion for footing:

$$2,330 = 29\% \text{ of } 7,974$$

which is acceptable.

C magma. The magmas are mixed with water in order to retain their purity and to simplify operations. There is thus 2,330 kg/h of d.s. which serves to go to the B massecutes.

Purity of the B massecutes. This will be:

$$P_B = \frac{5,644 \times 69 + 2,330 \times 85}{7,974} = 73.7$$

B molasses. We should find:

$$\begin{array}{ccc} 91.7 & \begin{array}{c} \diagdown \\ \diagup \end{array} & 19.7 & \text{B sugar} \\ & 73.7 & & \\ 54 & \begin{array}{c} \diagup \\ \diagdown \end{array} & \frac{18}{37.7} & \text{B molasses} \end{array}$$

$$7,974 \times \frac{18}{37.7} = 3,807 \text{ kg/h d.s.}$$

All is correct. B sugar:

$$7,974 \times \frac{19.7}{37.7} = 4,167 \text{ kg/h d.s.}$$

A massequite. Dry substance in the commercial sugar:

$$4,167 + 15,000 = 19,167 \text{ kg/h}$$

and we check that:

$$19,167 \times \frac{19}{30} = 12,139 \text{ kg/h d.s.}$$

The true quantity of commercial sugar is:

$$\frac{12,140}{0.992} = 12,238 \text{ kg/h}$$

and the molasses:

$$\frac{2,860}{0.85} = 3,365 \text{ kg/h}$$

Calculations for the boiling house

Throughout the whole boiling-house process, and particularly for sugar boiling and fugalling, it is very interesting and practical to relate the calculations to unit of non-sugar. In effect, water is evaporated and sugar withdrawn, all the way from syrup to final molasses. Calculations thus cannot be based on the weight of either brix or sucrose, or on purities. The non-sugar, on the contrary, is theoretically constant and does not vary from beginning to end of the process. The non-sugar entering with the syrup should be recovered in the final molasses, apart from the small proportion left in the commercial sugar, and in the absence of accidental losses of material in the course of the process.

Assuming, for example, a B syrup of purity 42 and a final molasses of purity 35, a recovery as crystal from this B molasses in the course of the C strike will be:

$$p = \frac{\frac{42}{100 - 42} - \frac{35}{100 - 35}}{\frac{42}{100 - 42}} = \frac{0.724 - 0.538}{0.724} = 0.257 = 25.7\%$$

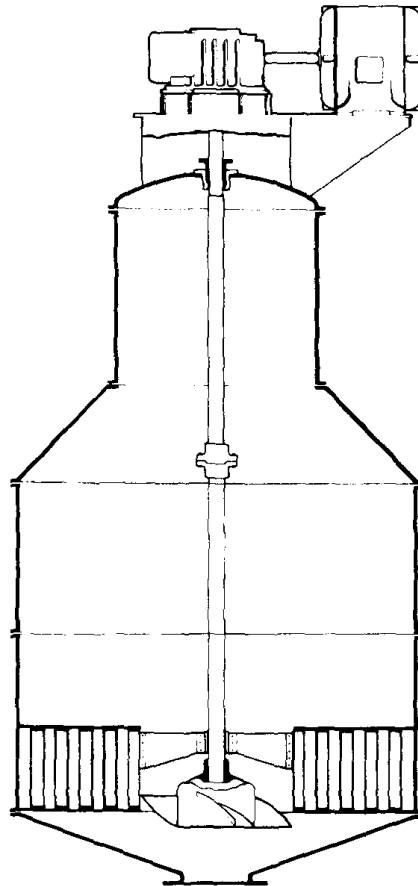


Fig. 33.23. Pan with mechanical circulation (Fletcher & Stewart).

In the same way, an expression for the exhaustion process may be advantageously calculated in terms of units of non-sugar. The true quantities are readily obtained with the aid of the purity figure, by relating the non-sugar to the total product (*i.e.* sugar + non-sugar + water). The calculations are thus simplified.

PANS WITH MECHANICAL CIRCULATION

Mechanical circulation has found in America an ardent promoter and supporter in A.L. Webre; it becomes increasingly popular.

Pans with mechanical circulation are operated as for other pans. However, they present some important differences compared with the latter. (Fig. 33.23).

(a) **Speed of circulation.** Webre⁸⁶ designs his circulators for a velocity of massecuite in the tubes of 45 cm/s (18 in./s), which allows only about 3 s for contact with the heating surface, reducing substantially the maximum temperature. Actually, however, the velocity given to the massecuite is far from attaining that value. It has been measured⁸⁷ and found to vary from 20 cm/s (8 in./s) at the beginning of the strike to 5- 10 cm/s (2- 4 in./s) at the end of the heavying-up in the case of the C massecuite. The latter values are nevertheless much superior to those obtained with natural circulation (*cf.* p. 661).

(b) **Evaporation coefficients.** Evaporation coefficients found by Webre for natural circulpans have already been given (p. 663). Completing the table, we recall these values below and add those found by the same author in the same series of experiments on the same massecuite (B) and at the same purity (72):

	Heat-transfer coefficient, in kcal/m ² /°C/h (B.T.U./sq.ft./°F/h)		Evaporation coefficient, in kg/m ² /h (lb./sq.ft./h)		
	Start	Finish	Start	Finish	Mean
Natural circulation	490 (100)	24 (5)	34 (6.98)	1.2 (0.25)	11 (2.25)
Mechanical circulation	550 (113)	193 (40)	35 (7.2)	12 (2.5)	20 (4.1)

It is seen that mechanical circulation does not alter performance greatly at the beginning of the charge, but modifies it substantially at the heavying-up period. In this case, the evaporation coefficient at the end was ten times greater for mechanical circulation, but in general Webre reckons *only* six times greater.

(c) **Duration of strike.** It follows that the duration of the strike is substantially reduced. Webre⁸⁸ reckons a saving in time of:

A massecuites: 15- 20%	hence duration 80- 85% of previous
B massecuites: 25- 30%	hence duration 70- 75% of previous
C massecuites: 40- 50%	hence duration 50- 60% of previous

(d) Steam pressure. In the same way, low-pressure vapour may readily be used, for example, vapour bled from the second effect.

(e) Height of massecuite. Finally, the height of the massecuite above the calandria may be pushed considerably above what is reasonable with natural circulation. A height of 2 or even 2.4 m (7 or 8 ft.) may readily be attained without difficulty. Hence a gain in capacity and a decrease in graining volume.

In short, mechanical circulation permits the use of bled vapours at low pressure, and consequently economy in fuel, while improving the quality of the sugar, the purity drop, and consequently the yield. It is in C massecuites that it gives the greatest improvement.

(f) Power consumption. The power consumed by the circulator increases very rapidly as the massecuite is tightened up near the end of the strike.

Spengler and Horn⁸⁹ have measured the power necessary to circulate the massecuite in a small pan. Table 33.16 gives the results obtained, as a function of the temperature and the crystal content⁹⁰. The figures in the table correspond to the power consumed in watts.

TABLE 33.16

POWER ABSORBED BY MECHANICAL CIRCULATION, IN Watts

Crystal % massecuite	Temperature, in °C (°F)				
	30 (86)	40 (104)	60 (140)	70 (158)	80° (176°)
0	83.8	82.5	81.5	80.5	80.0
5	86.8	85.5	84.5	83.5	82.3
10	88.8	87	86	85	84.1
15	90.6	88	86.5	85.6	84.3
20	91	88.5	87.5	86	85
25	93	89	88.5	87.1	86
30	95	90	89	88	86.2
35	100.8	92.6	92	91	88.3
40	106.9	100	98.5	94.2	92.7
45	125	119	107.5	102.3	97.6
50	204	183.3	142	126	106.6
55	335	300	250	190	129.1

For a pan of 250 hl (880 cu.ft.), Duus⁹¹ gives an average power of 22.5 kW (30 h.p.), with a peak at the end of the charge of 45 - 48 kW (60 - 65 h.p.). Webre⁹² estimates 55 - 75 kW (75 - 100 h.p.) for the peak power required by a pan of standard type 4.25 m (14 ft.) in diameter (approximately 400 hl (1,400 cu.ft.)).

Approximately, we may estimate:

Nominal power to be installed: 2.3 kW/m³ (0.087 h.p./cu.ft.) of working capacity
 Peak power consumption: 1.9 kW/m³ (0.07 h.p./cu.ft.) of working capacity
 Average power consumption: 1.0 kW/m³ (0.04 h.p./cu.ft.) of working capacity

The efficiency of the stirrer is very low, of the order of 7%. It has been found in

Australia⁹³ that a circulator of semi-axial and semi-radial flow gave a much better efficiency, of the order of 11%. For an impeller of the Webre type, the position giving the best result corresponded to the bottom of the centre well, with the impeller having 40% of its thickness (*i.e.* its height) in the interior of the downtake and 60% below the lower tube plate. The circulation remains good⁹⁴ as long as the fraction of the height of the impeller situated within the downtake is between 100% (bottom of the impeller flush with the tube plate) and 35%. The optimal diameter of the impeller is $0.94D$ (D = inside diameter of downtake). Its pitch should be $1.5A$ and its horizontal projection should cover 4 sections of 60° in the case of a 4-blade impeller or 2 sections of 90° in the case of 2 blades.

The angle of attack of the blades is generally 37° . It is advantageous if the bottom of the pan has a central conical section, as shown in Fig. 33.9. This assists in directing the flow of the massecuite towards the outer walls of the calandria.

On account of the large increase in power consumption as the tightening up of the massecuite proceeds, it is of advantage to have 2-speed agitators. The higher speed is then used at the start of the strike, and the low speed from the time when the massecuite level reaches about three-quarters of the working capacity. In this way, excessive power consumption is avoided, while still realising, at any period of the strike, a circulation very superior to what would be obtained with natural circulation, which indeed is practically nothing at the finish.

Jenkins⁹⁵ considers that it would be necessary to reduce the speed to 30% below the maximum.

The 2-speed drive may be realised by a motor with two sets of windings, or by a 2-speed gearbox, but the latter solution is expensive.

Webre⁹⁶ recommends stopping the circulator at the moment steam is shut off, as the power load on the motor increases abruptly when boiling ceases.

(g) Speed of the circulator. The circulator located in the centre well takes up a large proportion of its cross-section. If it is not to obstruct the circulation more than it assists it, it is necessary that it should rotate at a certain minimum speed.

Webre⁹⁷ has adopted the following speeds for circulators of 1.2 m (4 ft.) diameter in pans of 3.65 m (12 ft.) diameter:

For footings, A and B massecuites:	80 r.p.m.
C massecuites of brix up to 97:	60 r.p.m.
C massecuites of brix greater than 97:	50 r.p.m.

Venton⁹⁸ considers that it would be of advantage to retain a speed of about 50 r.p.m. for all heavy massecuites.

The impeller may have a large central hub (Fig. 33.23) or a narrow hub. It may be a ship's propeller with 3 blades, and in this case it is advantageous to fix on the wall of the downtake 4 vertical plates arranged radially, of radial width about 20% of the radius of the downtake, with the object of guiding the descent of the massecuite vertically and avoiding horizontal turbulence (Fig. 33.24). The impeller may also be formed of blades fixed on a conical boss with an apical angle of about 26° , the blades being almost normal to the cone⁹⁹.

The peripheral speed at the extremity of the blades is generally between 5 and 7.5 m/s

(16—25 ft./s, more often about 6.5 m/s for massecuite of very high purity, or between 3.5 and 5 m/s (11—16 ft./s) for low-grade massecuite. A ship's propellor with a large boss consumes less power than one with a narrow hub.

In pans with mechanical circulation, a shorter tube length is generally adopted than for pans without agitator⁴⁹, for example, 600—700 mm (24—28 in.).

Advantages of mechanical circulation. Summarising and supplementing the advantages already indicated, Webre indicates the following:

- (a) Saving in time for the strike (*cf.* p. 707)
- (b) Maximum temperature reached by the massecuite substantially lower
- (c) Crystals better formed, more shapely and more regular
- (d) Less colour in the sugar and reduced risk of sugar losses by local overheating

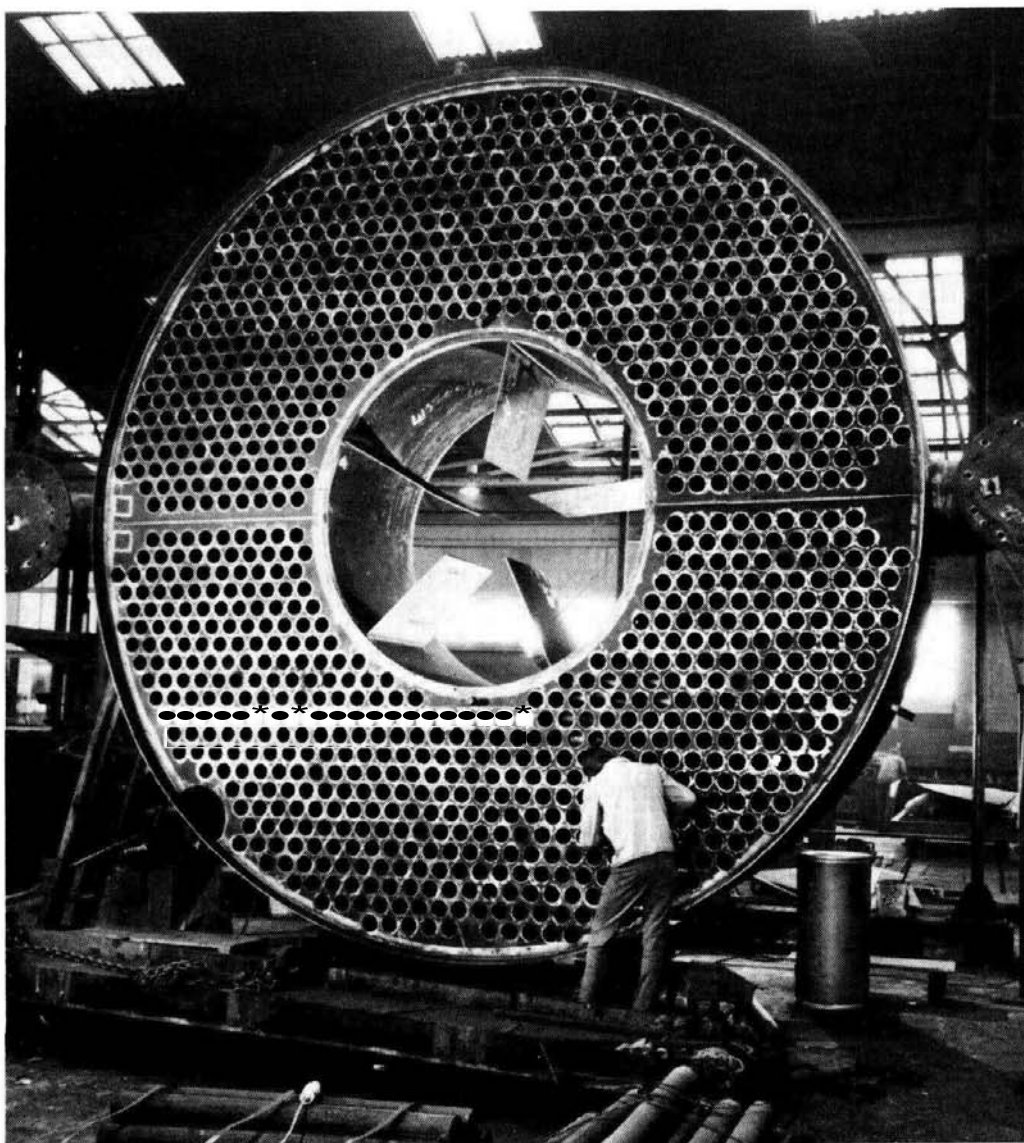


Fig. 33.24. Calandria for mechanical circulation pan with radial blades.

(e) Possible ready application of automatic feed controlled by the power consumption of the motor; when power increases, the feed valve is opened; when it decreases, the valve is closed

(f) Possibility of feeding the pan with syrup and molasses only slightly undersaturated, without risk of remelting crystal

(g) Less deposit on the tubes, due to the abrasion effect by friction of the circulating crystals

(h) Possibility of utilising a lower pressure of steam for heating.

The disadvantage of mechanical circulation is the power consumption.

Application. Pans with mechanical circulation give a marked improvement in the boiling, and improve the quality of the product. They permit a valuable gain in the purity of molasses and render possible work with tighter masseccutes. They are thus strongly recommended, particularly with C masseccutes. It may even be justifiable for a new factory to equip all pans with mechanical circulators. The only alternative to compete with this would be equipment with continuous pans. A series of pans offered by Fives Cail - Babcock with mechanical agitators is identical with that of their pans with flat fixed calandria, the only difference being the addition of the agitator, which can in fact be installed without difficulty in their ordinary pans (Fig. 33.25).

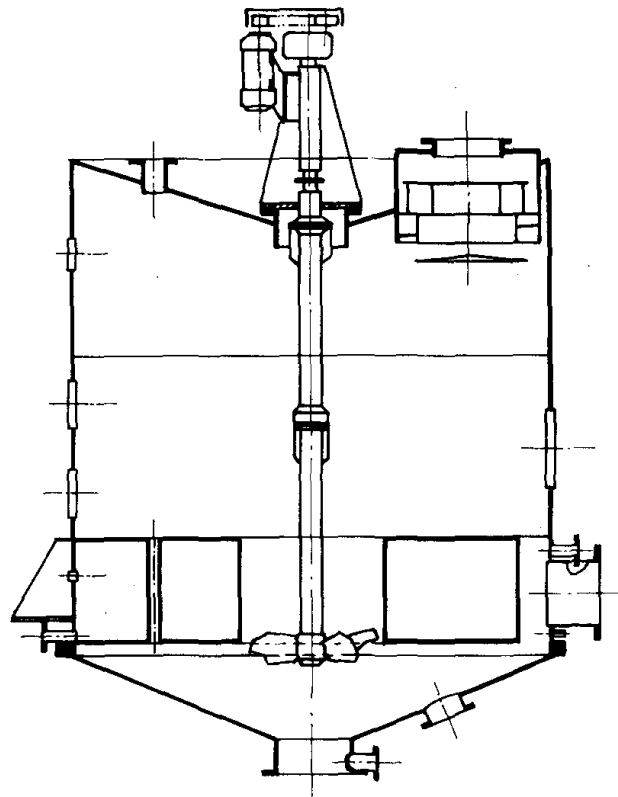


Fig. 33.25. Fives Cail - Babcock pan with circulator, type CTM.

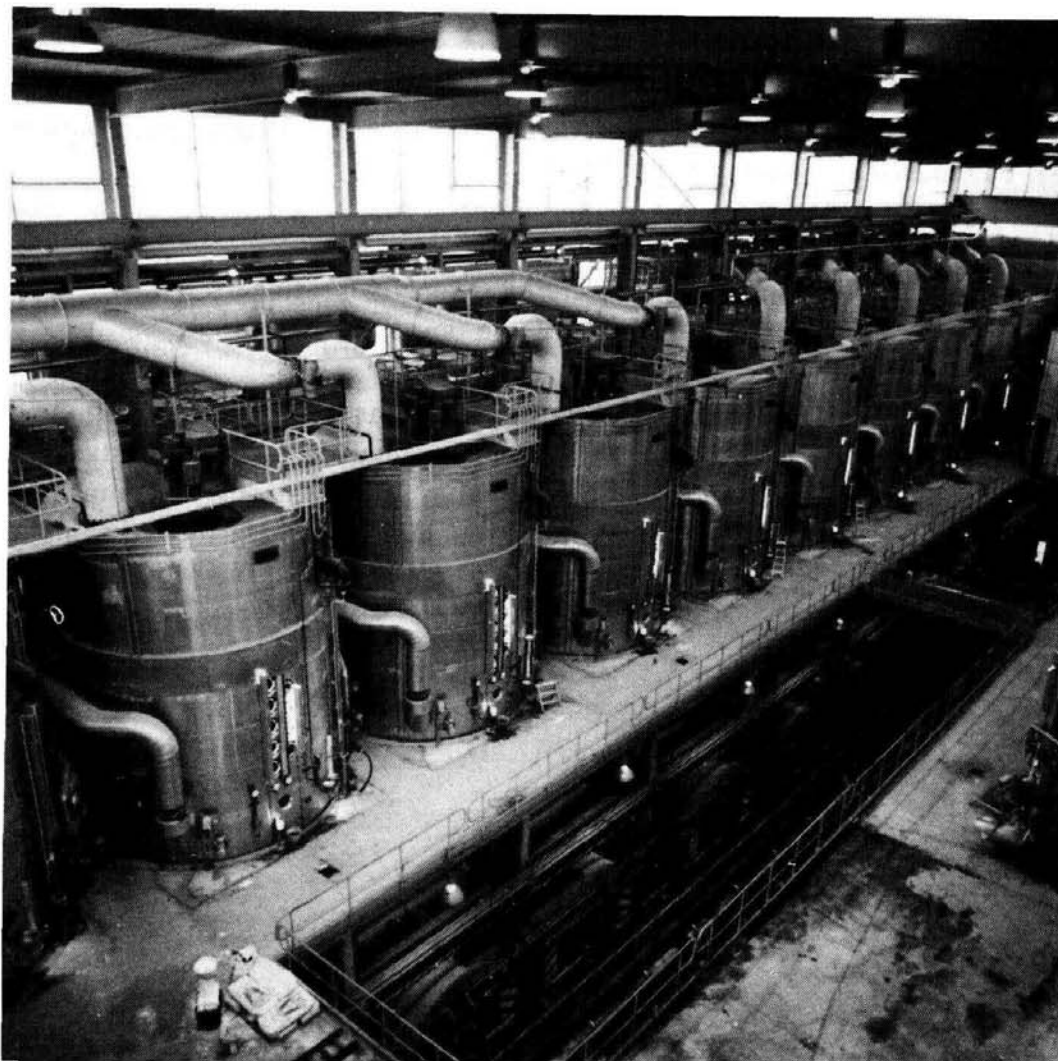


Fig. 33.26. Boiling station with mechanical circulation pans (B.M.A.)

CONTINUOUS VACUUM PANS

Continuous pans have appeared in the course of the 1970's, and are expanding rapidly. For the same space occupied, they accomplish the same work as ordinary pans in the same time. They present numerous advantages in space occupied, cost, simplicity and economy of operation, and their initial difficulties have been the only reason delaying their more general use. In the present situation, the continuous pan should become general.

Fives Cail—Babcock continuous pans

The best-known and practically the only type in wide use at the moment is the continuous pan of Fives Cail—Babcock (Figs. 33.27 and 33.28).

The pan consists of a cylindrical body with horizontal axis, and with reinforced end plates. In the lower half of the pan, there is an assembly of longitudinal tubes arranged in horizontal rows; the upper row is flat, while the bottom rows are shaped to follow the curve of the cylin-

drical body of the pan, leaving a space at the sides between the tubes and the pan wall for downward circulation of the massecuite. The tubes are free and not surrounded by any metal sheets. Being disposed in a rectangular pattern and not staggered, they are in vertical rows also, thus facilitating the ascending circulation in the interior of the tube assembly.

The tubes are of 35 x 38 mm, and this exterior diameter permits general standardisation of tube sizes throughout the factory. They may also be of 27 x 30 mm. The thickness is 1.5 mm. They are fixed at the ends by expansion joints to allow for expansion.

Vertical baffles divide the pan into a number of compartments, which may vary between 10 and 16, but will be standardised at 13. These compartments communicate with each other by an opening located at the base of the partitions. The first compartment occupies the full width of the pan; it delivers on the left into the second compartment which, with the third, occupies the space between the first and second transverse partitions; the second delivers across the vessel into the third, which delivers longitudinally on the right into number 4, and so on; the intermediate compartments are in pairs across the pan.

In the beet sugar industry there are pans carrying out all the sugar-boiling operations, including seeding and graining. With cane sugar, assuming that the CBA or CB/CA systems are followed, the footing for a C massecuite is formed in an ordinary pan and the continuous pan is fed with this footing; with A and B strikes, the continuous pan is fed directly with the magma which is fed into the first compartment.

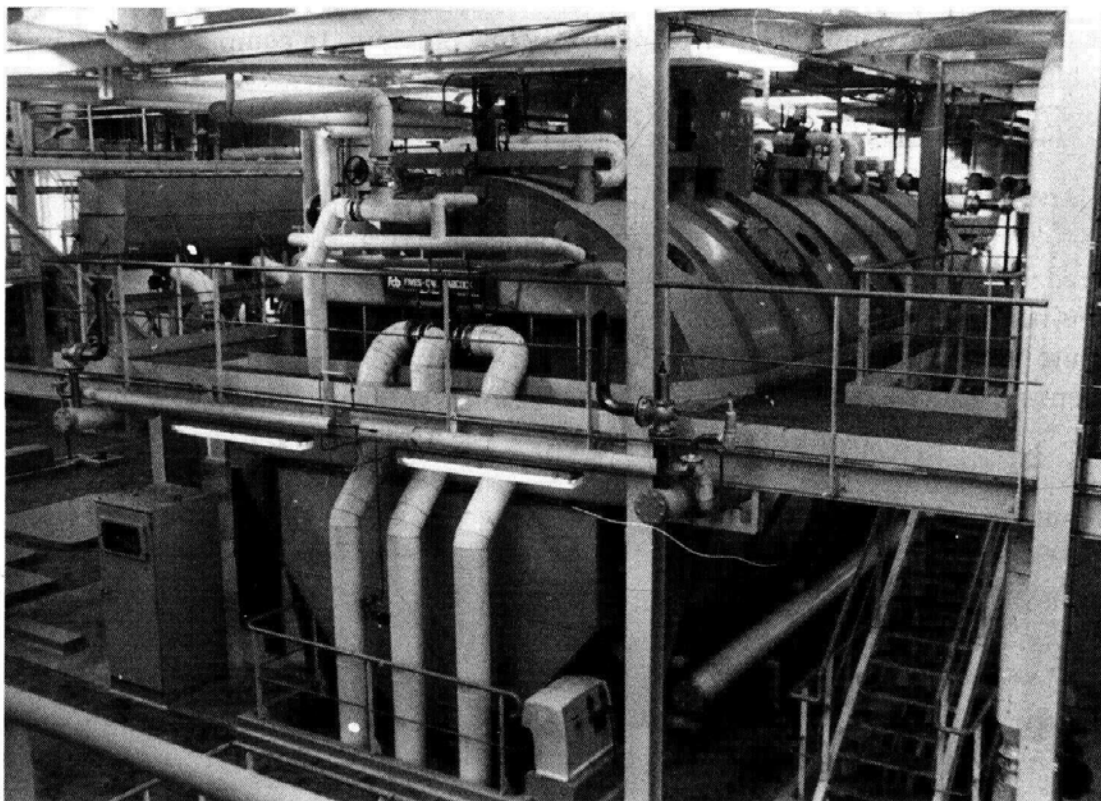


Fig. 33.27. Continuous pan (Fives Cail - Babcock).

Controls. For the three-massecuite system, continuous pans are equipped with four systems of control designed to maintain at a constant level:

- (a) Steam pressure in the tubes
- (b) Vacuum in the pan
- (c) Pressure of agitating steam
- (d) Massecuite level in the pan.

In addition, Fives Cail – Babcock supply:

(7) For A and B massecuities: (d) a control adjusting the flow of syrup or molasses Q admitted into the later compartments as a function of the density of the massecuite delivered; (b) a control maintaining an input of syrup or molasses into the early compartments proportional to Q .

(2) For C massecuities: the compartments are separated into three groups each having its equipment for regulating the temperature or the conductivity of the massecuite controlling the feed of molasses into the preceding compartments.

A flowmeter registers the total feed of syrup or molasses introduced into the pan.

The incondensable gases from the steam are used to supplement the mixing and agitation produced by the boiling in the pan, by injecting the gases along the lower portion of the pan, thus serving to homogenise the mass and prevent stagnation of the bottom layer. The designer states that this supplementary agitation avoids formation of false grain and contributes to production of a regular crystal. The incondensable gases are evacuated with the vapour from the massecuite from the top of the pan. This agitation is thus necessary. Moreover, with cane, it is necessary that the pan should be lagged; this contributes to avoiding deposit of crystal on the walls of the pan. With the same object, a heating jacket or double enclosure may be provided and supplied with steam at very low pressure. In equipment for massecuities of very high purity, the formation of such deposits is also prevented by means of small swinging nozzles which have received the nickname of "whirling dervishes".

Range of sizes. Fives Cail - Babcock make these pans in four diameters D with four or six different lengths L available, giving 13 different capacities V . In the medium sizes, a choice is offered between short and long models. All pans have a ratio S/V of $10 \text{ m}^2/\text{m}^3$ (3 sq.ft./cu.ft.). Table 33.17 gives the pan dimensions and in each case the volume V of massecuite which can be treated per hour, also the capacity A in t.c.h. of the factory which this equipment should serve according to which massecuite (A, B and C) it will be required to handle.

Other continuous pans

A pan very similar to the Fives Cail – Babcock was installed at Mossman in Australia in 1976. It has been the object of numerous studies by Broadfoot and Wright¹⁰⁰, after which they have made the following recommendations:

- (a) To limit possible entrainment, they consider the volume of vapour space above the massecuite level should represent 80 – 86% of the massecuite volume in the pan.
- (b) The magma introduced in a C pan should have at least 20% of crystal, and the growth of crystals should go from 0.18 mm at entry to 0.30 at the exit, for a ratio weight of massecuite to weight of magma of approximately 3.5 – 4.

(c) The coefficient of variation of crystal should range from 0.25 in the magma to 0.30 for the final massecuite.

(d) They suggest that the successive capacities of the compartments of a pan of 120 m³ of 7 compartments should be: 6, 10, 16, 20, 20, 24 and 24 m³ (or 5, 8, 13, 17, 17, 20 and 20%, respectively).

TABLE 33.17

SERIES OF CONTINUOUS PANS (FIVES CAIL - BABCOCK)

Cap. (m ³)	Diam. (m)	Length (m)	Diam. (m)	Length (m)	Massecuite A		Massecuite B		Massecuite C	
					v (m ³ /h)	A (t.c.h.)	v (m ³ /h)	A (t.c.h.)	v (m ³ /h)	A (t.c.h.)
16	3.1	3.715			8.1	60	6.4	85	3.2	70
19	3.1	4.455			9.5	70	7.5	100	3.8	85
22.5	3.1	5.135			11.5	85	9	120	4.5	100
27	3.1	5.935	3.5	5.195	13.5	100	10.9	145	5.4	120
32	3.1	7.415	3.5	5.935	16.2	120	12.8	170	6.3	140
38	3.1	8.900	3.5	7.415	18.9	140	15	200	7.6	170
45	3.5	8.900	4.0	5.935	22.3	165	18	240	9	200
54	3.5	10.385	4.0	7.415	27	200	21.8	290	10.8	240
64	3.5	11.870	4.0	8.900	32.4	240	25.5	340	12.6	280
76	4.0	10.385	4.6	7.415	37.8	280	30.4	405	15.3	340
90	4.0	11.870	4.6	8.900	42.6	330	36	480	18	400
104	4.6	10.385			51.3	380	41.6	555	20.7	460
120	4.6	11.870			59.4	442	48	640	23.8	530

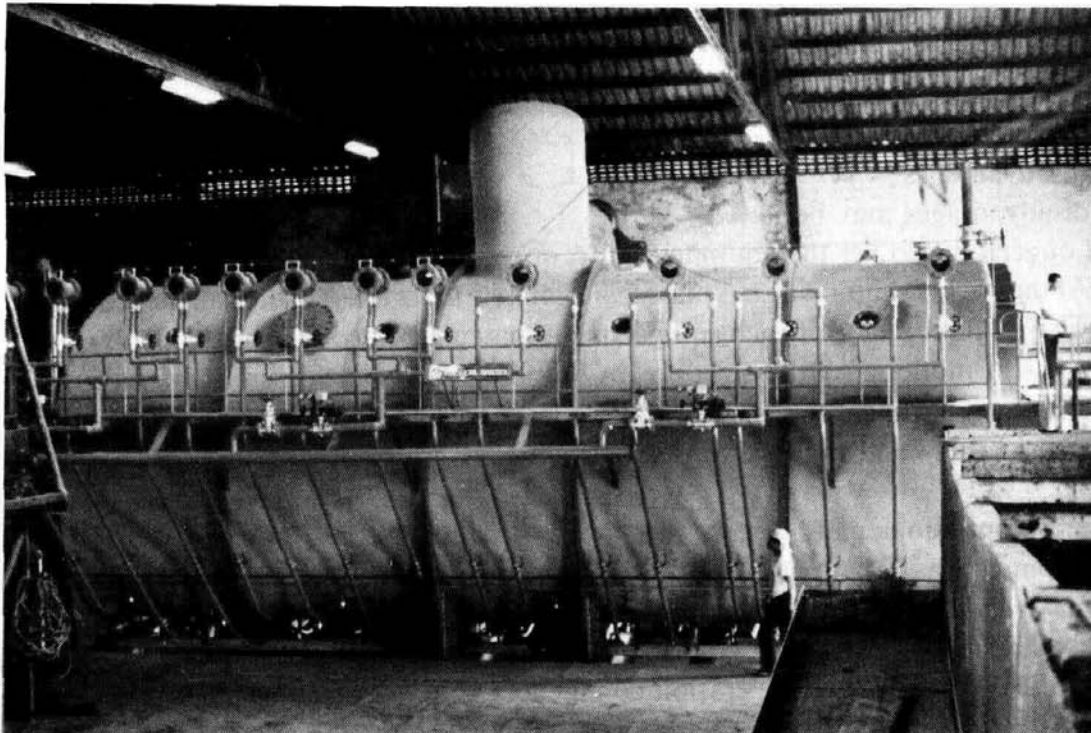


Fig. 33.28. Continuous pan (Fives Cail - Babcock).

(e) The partitions of the equipment should be coated with epoxy resin to prevent the sugar crystals from adhering and falling later into the massecuite, tending to block the passages.

(f) It is considered in Australia¹⁰¹ that the ratio of total area of passage between the tubes to the corresponding downtake area for massecuite should be less than 2.5 and as low as possible. We suggest approximately 2.

Advantages

Continuous vacuum pans are destined to replace discontinuous pans which are still almost universal. They present great advantages:

(a) They can function at very low steam pressures on account of the low height of massecuite above the calandria, and the agitation of the massecuite below.

(b) Steam consumption is uniform, in contrast to the high peak demands at the start for batch pans.

(c) No discharge or steaming-out is involved as is the case after each batch boiling, with consequent dilution of material and losses of heat.

(d) Total automation is possible, simplifying the operation of the factory.

(e) Space occupied is much less, first because the useful volume of the equipment is much less for equal production capacity, and also on account of the gain in capacity due to the uninterrupted operation.

(f) There is economy in manpower: one man can operate the whole of the pan stage.

(g) The number of pans is reduced: the factory can operate with 3 continuous pans and one for graining.

(h) Investment is lower: supporting structures are lighter, the large vacuum pump for the condenser is perhaps not required.

(i) Maintenance is reduced.

(j) There is the possibility of standardising tubes with those for other units in the factory.

(k) It is possible to boil the massecuite to a higher brix.

Some disadvantages

Some disadvantages may be noted:

(a) Longer time to get the equipment into operation, when first installed.

(b) At the weekly shutdown an A massecuite is likely to set if left in the pan, hence the necessity to transfer the contents of each compartment to a mixer (also in compartments) where it is maintained in motion. At the start-up on Monday, each portion must be returned to its appropriate compartment to resume normal operation. Otherwise it is necessary to empty the equipment at the weekly shutdown.

Langrenney continuous pans

This equipment was designed about 1970 by Francois Langrenney. It depends on the idea that, if the massecuite is confined in a narrow corridor, there is no need for separate compartments. As the massecuite is a viscous material, it has no tendency to mix as a mobile liquid would do. Provided that the operation commences with a magma of sufficient consistency and this is provided with a direct path in a narrow corridor, there is no necessity to divide the equipment into separate compartments. The object of these is only to prevent the

massecuite from mixing with that of neighbouring stages, so that each stage corresponds to a portion of the required crystal size, and all the crystals follow approximately the same path as they pass along the equipment. Any crystals which lag behind, or which move faster than the main bulk, will give larger or smaller grains than required, thus resulting in a poor quality of sugar of uneven grain size.

The massecuite is heated in vertical tubes, and on leaving the tubes it returns down the narrow corridor and re-enters the calandria further downstream. Thus the height of the equipment must allow sufficient space above and below the tubes for the massecuite circulation, for release of vapour above the tubes, and for satisfactory entry of the massecuite at the bottom of the tubes. The forward movement of the material is maintained by the flow of magma and molasses entering. The material enters at one end of the corridor and leaves at the far end (Fig. 33.29).

Thus the basic equipment designed on this principle consisted of a long rectangular tunnel having a narrow tubular calandria, for example, of 4 to 6 rows of tubes, while on the other side there was an even narrower canal or corridor. The bottom portion of this allows a transverse area, per unit of length, greater than that of the calandria tubes for the same length, without giving excess area where the massecuite could stagnate. Vertical plates above the calandria support a roof formed of a horizontal plate to arrest splashes of the massecuite. The syrup or molasses feed to the boiling mass is introduced below the calandria at numerous points, by suitable distributing pipes.

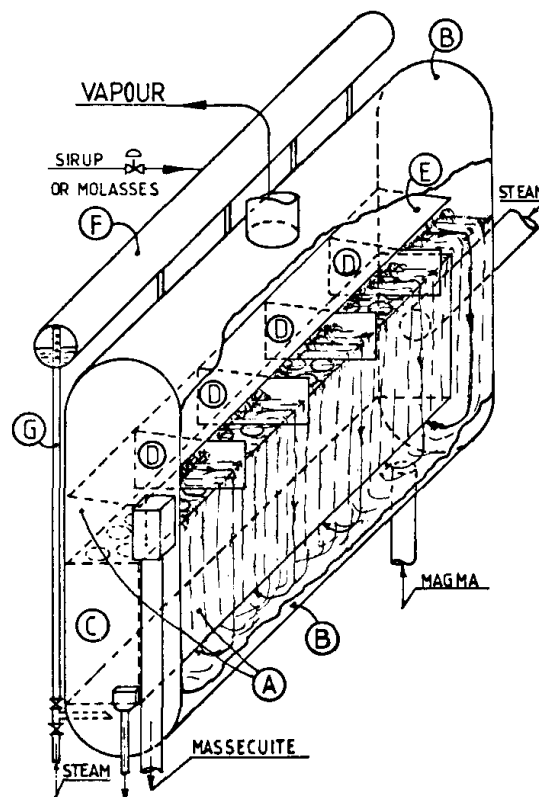


Fig. 33.29. Principle of the Langreny pan, massecuite circulation.

Since an excessively long apparatus would not be acceptable, Langreny first considered dividing this tunnel into two sections, one each side of the calandria, so that there would be one calandria with a channel on each side of it. The magma was introduced at one end on the one side, proceeding to the opposite end of the channel and continuing from one side to the other and returning to the initial end, where the massecuite left by overflow and descended by a barometric column to be collected 6 or 7 m (20 – 23 ft.) below. This was the first pilot equipment, the functioning of which was satisfactory and encouraged the inventor.

However, even divided into two, the length of flow was insufficient. Langreny then had the idea of changing the straight channel to a circular one; at the same time, he gained in strength of the material by avoiding the necessity for reinforcing a flat plate. Taking a diameter equal to the length of the straight equipment, he thus multiplied the length by a factor π , and to avoid loss of the double-channel concept, he again provided two channels, one on each side of the ring calandria. This latter was common to the two channels, and was provided with a vertical partition separating the portion serving the interior channel from that serving the exterior one. For a diameter of 7 m (23 ft.), he thus attained a length of 30 m (98 ft.). This is the equipment which he designated under the name of “crystalliser”.

Description. In the present models (1982), the equipment has the same exterior appearance as a normal vacuum pan. Inside, a radial baffle separates the beginning and the end of the circular channels. The magma is introduced below, in the interior channel, against this radial baffle. As the equipment is designed for a level of the boiling massecuite about 25 cm (10 in.) above the upper plate of the calandria, vertical baffles up to 1 m (3 ft.) above avoid lateral projection of drops and splashes. The interior circuit is separated from the exterior by a cylindrical baffle rising above the calandria and dividing it into two sections corresponding to the respective heating surfaces. This baffle extends below the calandria and is thicker at this position, since it serves as a support for the calandria. The massecuite passes from the interior to the exterior circuit by a large opening situated in the supporting plate and extending from the bottom of that plate, against the radial baffle.

The syrup or molasses feed is piped to two circular distributing boxes situated above the conical roof of the equipment, and descends by feedpipes distributed around the apparatus and feeding into the two circuits. There is one distributing box for each circuit. A sight glass is interposed in the vertical portion of the pipe, permitting visual inspection of the flow, and a hand-operated butterfly valve is provided.

The transfer at the bottom from vertical to horizontal flow is made by a T fitting, permitting cleaning of the pipe.

Discharge of the massecuite is made near the radial baffle, by means of an overflow with an adjustable weir plate, with sight glasses showing the overflow level. The circuit described above may be followed in the opposite sense (Fig. 33.30): entry by the exterior channel and discharge from the interior one. This arrangement is in fact necessary for the CA model which is described later.

The calandria is in two portions welded together to form a single body. There is one steam entry or, if necessary for large apparatus, two entries at 180° apart. Vacuum is maintained constant. There are at least 10 conductivity meters along the travel of the massecuite, three being in the first circuit, and seven in the second. The indications from electrodes 2 to 9 are

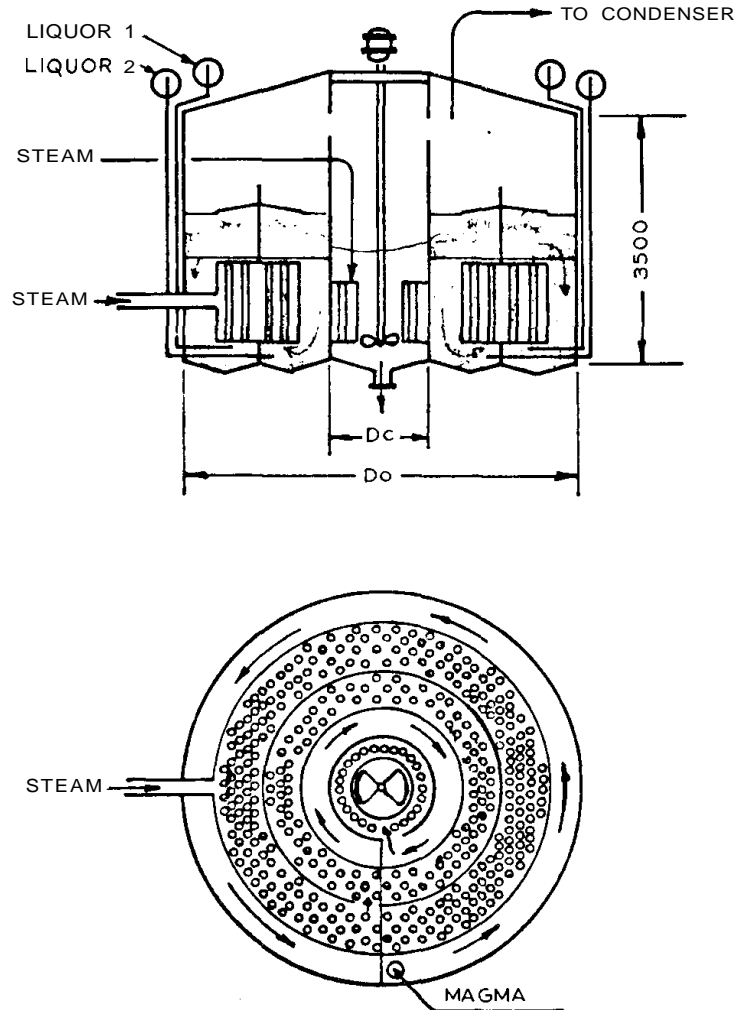


Fig. 33.30. Langreny continuous pan, type CA.

connected in a current - pressure converter and the result is transmitted to the molasses valve or steam valve.

Two systems of automatic control are employed at present. One consists of regulating the input of molasses according to the integrated indication of conductivity, the two circuits being controlled separately: the first by electrodes 2 and 3, the second by electrodes 5 to 9. The steam pressure is thus maintained constant. The other system consists of regulating the feed valve for molasses in relation to a flowmeter placed on the feed of magma, so as to establish proportionality between these two feeds, and regulating the steam valve in relation to the integration of the 8 conductivity readings. The flow meter is graduated in cubic meters per hour, with an adjustable setting.

The electrodes must be maintained very clean. For this purpose they are located at several decimetres from the wall of the equipment and are surrounded by a sheath which directs a jet of hot water under pressure on the electrodes, opened every quarter hour (with A massequite) or hourly with C massequite. All the instruments for these automatic controls are assembled on a control panel.

The steam from the boiling massequite is evacuated to the condenser through the central dome of the pan. The ratio S/V is approximately $9.2 \text{ m}^2/\text{m}^3$ (2.8 sq.ft/cu.ft). The calandria

tubes are of steel, 98.6 X 101.6 mm ($3\frac{7}{8}$ x 4 in.) and 1.20 m (4 ft.) in length. By limiting them to 1 m (3.3 ft.), the heat-transfer coefficient would be improved, but to maintain the same S/V , the diameter of the equipment would have to be increased by 10%.

The equipment functions well, and the coefficient of variation of the crystals is comparable with that from discontinuous pans. Its simplicity is an attractive feature, its cylindrical form and smaller overall size facilitating its installation in the pan station. The length of travel of the massecuite is a favourable factor.

There are 3 principal types:

- (a) The 2L5, for capacities below 30 m³, with straight element
- (b) The CN, for capacities 25 to 80 m³, cylindrical element
- (c) The CA, for capacities 15 to 80 m³, cylindrical with an agitator at the end of the boiling (Fig. 33.27).

With the CN model, the space inside the interior circuit is unfortunately wasted, since it is difficult of access: but it increases the diameter of the whole vessel by only 5% for a given capacity. The CA model resolves this difficulty very elegantly by finding an excellent use for this space. By adopting the reverse flow, the massecuite entering the exterior channel first and then the interior channel, the massecuite coming to the end of the latter passes through an opening in the central cylinder which thus forms a small vacuum pan with central downtake, heated by a calandria slightly lower than the principal calandria of the equipment and fitted with an agitator which ensures circulation of the massecuite at the moment when it is most useful and most effective, *i.e.* at the end of the strike. The level in this inside pan is maintained slightly lower than in the main circuits, and the massecuite is withdrawn at a point in the periphery by a small opening opposite the point of entry to this third circuit.

This model makes the Langreny the only continuous pan with an agitator. Thus modified, this equipment, already original in its conception, marks an interesting stage in the history of the continuous vacuum pan (Fig. 33.30).

Table 33.18 gives dimensions of the various models of the CA pan.

TABLE 33.18

SERIES OF LANGRENEY CONTINUOUS PANS TYPE CA

No.	Diameter D		Volume V		Heating surface S	
	(m)	(ft./in.)	(m ³)	lcu.ft.)	(m ²)	(sq.ft.)
150CA	3,710	(12 ft. 2 in.)	15	530	138	1,488
200CA	4,298	(14 ft. 1 in.)	20	706	184	1,984
250CA	4,673	(15 ft. 8 in.)	25	883	230	2,480
300CA	5,233	(17 ft. 2 in.)	30	1,059	276	2,976
350CA	5,639	(18 ft. 6 in.)	35	1,236	323	3,472
400CA	6,045	(19 ft. 10 in.)	40	1,413	369	3,968
450CA	6,401	(21 ft. 0 in.)	45	1,589	415	4,464
500CA	6,757	(22 ft. 2 in.)	50	1,766	461	4,960
600CA	7,391	(24 ft. 3 in.)	60	2,119	553	5,952
700CA	8,001	(26 ft. 3 in.)	70	2,472	645	6,944
800CA	8,534	(28 ft. 0 in.)	80	2,825	737	7,936

(Diameter of small interior pan with agitator = 43% of D .)

Relative capacities of batch and continuous pans

The capacity of a continuous pan, reckoned in $\text{m}^3/\text{t.c.h.}$, may be reckoned as about 20% higher than that of a discontinuous pan of the same useful volume.

At Tongaat and Gledhow¹⁰² it is considered that the respective capacities, reckoned in kg evaporation per hour and m^2 , were approximately $5.3\text{-}6 \text{ kg}/\text{m}^2/\text{h}$ ($1.1\text{--}1.2 \text{ lb.}/\text{sq.ft.}/\text{h}$) for continuous Fives Cail - Babcock pans, and that of batch pans about twice that figure during their time of operation. The advantage of the continuous pans is due to their superior S/V (of the order of $10 \text{ m}^2/\text{m}^3$ ($3 \text{ sq.ft.}/\text{cu.ft.}$), compared with $6\text{-}8$ ($1.8\text{--}2.4$)) and to the fact that they function in a continuous manner, whereas ordinary pans operate intermittently.

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34. Crystallisation

Object

The massecuite when discharged from the pan is at a high supersaturation. If it is allowed to stand, the sugar still contained in the mother liquor will continue to be deposited as crystals, but this massecuite is very dense and the mother liquor very viscous. Crystallisation will soon cease if the massecuite is left undisturbed, because the layer of mother liquor surrounding the crystals will be rapidly exhausted, and the viscosity of the mass will prevent the more distant molecules of sugar from circulating and coming in contact with the crystals.

If we are to take advantage of the strong tendency of the massecuite to crystallise after boiling, it must be kept in motion in order to change constantly the relative positions of the particles of mother liquor and of crystals. In factory parlance this is termed "crystallisation"; strictly speaking, of course, the whole process in the pans constitutes crystallisation of the sugar, but in the factory the term "crystallisation" is used particularly for the crystallisation in motion after dropping the massecuite from the pan.

Crystallisation, then, is a process which consists of mixing the massecuite for a certain time after dropping from the pans, and before passing to the centrifugals; and which aims at completing the formation of crystals and forcing further exhaustion of the mother liquor.

Proportions of different massecuites

We repeat in Table 34.1 figures for average proportions of the three massecuites:

TABLE 34.1

3-MASSECUITE SYSTEM

(Quantities in $\text{dm}^3/\text{t.c.}$ (cu.ft./t.c.)

	<i>Tromp</i>	<i>Our figures</i>	
		Single magma	Double magma
A massecuite	85 (3)	95 (3.4)	130 (4.5)
B massecuite	45 (1.6)	50 (1.8)	50 (1.8)
C massecuite	40 (1.4)	35 (1.3)	35 (1.3)
Total	170 (6)	180 (6.5)	215 (7.6)

Density of massecuite

For hot massecuite we have taken (Table 32.3 and Chap. 33) a specific gravity of 1.47 for massecuites (density 92 lb./cu.ft.). For cold material, between 30 and 40°C (85 and 105°F), a value of 1.50 (94 lb./cu.ft.) may be taken.

Dilution of the massecuite

When the massecuite purges badly at the centrifugals it is sometimes diluted with water, or with a suitable grade of molasses. This dilution, especially with water, impairs the exhaustion and should be avoided. If it proves necessary to dilute for fuggalling, it should be done only 2 or 3 hours before passing to the centrifugals.

Flow of the massecuite

The minimum slope of chute to be provided for discharge of the massecuite leaving the pans is given by Tromp¹ as 9° (16%). He recommends preferably 11°, or 20%. However, he indicates elsewhere² 5%, or 3°, for the cold massecuite after crystallisation. This value is certainly a minimum and it would be advisable to keep well above it.

Temperature of cooling of the massecuite

The massecuite on leaving the pans is at a temperature of 70 – 75°C (160 – 165°F). To what temperature is it advisable to cool it in the crystalliser?

Low-grade massecuites. Deerr³ considers that the best temperature to which to cool the low-grade massecuite is 41 – 43°C (105 - 110°F), and that if it is taken below that figure, the mother liquor becomes so viscous that the gain in sugar crystals is offset by the quantity of additional water needed for purging in the centrifugals.

Jenkins⁴ suggests 38°C (100°F) as about the practical limit.

If the massecuite is reheated before fuggalling, a lower temperature may be employed, 36°C (97°F), for example.

High-grade massecuites. For massecuites of higher purity the limit of temperature would be lower, since the increase in viscosity with cooling would be less than with low grades. Jenkins recommends crystallising A and B massecuites as for C strikes, but emphasises the importance of efficient pan work; and has since expressed the view that with modern pans capable of producing a massecuite of maximum crystal content, crystallisation of high grades is less attractive. Few factories have considered it in their interests to practice crystallisation of high grades, firstly on account of the danger of having the massecuite go solid in the crystalliser, and secondly on account of the space required for the extra equipment necessary. However, the operation would be interesting, particularly on B massecuite.

High-purity massecuites present the dangerous possibility of going solid, that is, of solidifying in such a manner that they can be dealt with only by attacking with a pick or by dissolving the mass. This possibility is a nightmare for overseers after stoppages due to strikes or power failures of long duration.

Such setting of a massecuite can occur if the purity is above 75 and the brix above 90, and when the stirring gear has been stopped for a few hours.

Reheating of the massecuite. If the massecuite is difficult to handle at the centrifugals, it may be reheated before centrifuging in order to reduce its viscosity. We shall discuss this again in connection with centrifugals (*cf.* p. 807).

Speed of crystallisation

Speed of crystallisation in a massecuite is a function of the supersaturation, the temperature and the crystal surface area. If a constant speed of crystallisation is to be maintained, temperature must be regulated as a function of the supersaturation, or inversely (*cf.* Table 33.2, p. 000). Experience, moreover, has shown⁵ that the best results correspond to crystallisation at constant supersaturation of the mother liquor. The object will thus be to maintain the supersaturation in the neighbourhood of 1.2.

Purity drop during crystallisation

Praeger⁶ claims that, in the 3-massecuite system, it would be possible to obtain by crystallisation an additional 6 points' purity drop for each grade of massecuite.

In Louisiana, Daubert⁷ claims that it should normally be possible to obtain, with a well-controlled C massecuite, an apparent purity drop of 27 points between massecuite and molasses, *e.g.*:

15 – 16 points in the pan

11 — 12 points in the crystalliser.

This is a maximum value, which we have rarely known to be achieved.

In Australia, Foster⁸ considers that, provided the massecuite is allowed to attain equilibrium, the true purity will be reduced by one point for each 4°C (7°F) of reduction in temperature.

Supersaturation

The same author considers that final massecuite can be discharged from the pan at a supersaturation of 1.2 without risk of the formation of false grain in a crystalliser. On the other hand, mixing a residue of cold massecuite with freshly discharged massecuite inevitably forms false grain.

Sugar recovery

Problem. What are the proportions of sugar and molasses produced by a given massecuite?

Solution. Let

P_m = purity of massecuite

P_s = purity of sugar

P_e = purity of the molasses

B_m = weight of dry substance % of massecuite

B_s = weight of dry substance % sugar

B_e = weight of dry substance % of molasses

Q_m = weight of massecuite

Q_s = weight of sugar produced

Q_e = weight of molasses obtained.

By a reasoning identical with that of the problem on p. 801, we obtain:

$$\frac{Q_s}{Q_m} = \frac{B_m}{B_s} \cdot \frac{P_m - P_e}{P_s - P_e} \quad (34.1)$$

where the first term represents the *weight of sugar % on weight of massecuite*.

Since the massecuite is generally reckoned by volume, we shall have:

$$Q_m = V_m d_m$$

V_m = volume of massecuite, in m^3 (cu.ft.) for example

d_m = density of massecuite, in kg/m^3 (lb./cu.ft.) for example.

Hence:

$$\frac{Q_s}{V_m} = \frac{B_m d_m}{B_s} \cdot \frac{P_m - P_e}{P_s - P_e} \quad (34.2)$$

We have, very closely:

$$\begin{aligned} d_m &= 1,500 \text{ kg/m}^3 \text{ (94 lb./cu.ft.)} \\ B_s &= 100 \end{aligned}$$

which gives:

$$\frac{Q}{V_m} = 15 B_m \frac{P - P_e}{P_s - P_e} \quad \left(\frac{Q}{V_m} = 0.945 B_m \frac{P - P_e}{P_s - P_e} \right) \quad (34.3)$$

where the first term represents the recovery R from 1 m^3 (cu.ft.) of massecuite, in kg (lb.) of sugar.

Example. If we assume the mean values of Table 34.2:

TABLE 34.2

YIELD OF SUGAR FROM MASSECUITES (NORMAL VALUES)

	B_m	P_m	P_s	P_e
A massecuite	95	85	99	65
B massecuite	96	72	98	50
C massecuite	98	58	95	35

we would have theoretically:

$$\begin{aligned} R_A &= 840 \text{ kg/m}^3 \text{ (53 lb./cu.ft.)} \\ R_B &= 660 \text{ kg/m}^3 \text{ (41 lb./cu.ft.)} \\ R_C &= 560 \text{ kg/m}^3 \text{ (35 lb./cu.ft.)} \end{aligned}$$

On account of the dilutions carried out, these yields in practice would barely exceed:

$$\begin{aligned} R'_A &= 800 \quad \text{kg/m}^3 \text{ (50 lb./cu.ft.)} \\ R'_B &= 600 - 650 \text{ kg/m}^3 \text{ (37 Ib./cu.ft.)} \\ R'_C &= 500-550 \text{ kg/m}^3 \text{ (31 Ib./cu.ft.)} \end{aligned}$$

In crystal per cent by weight of massecuite, Honig⁹ and Eisner¹⁰ give (Table 34.3):

TABLE 34.3

CRYSTAL YIELD % MASSECUITE (BY WEIGHT)

	<i>Purity</i>	<i>Honig</i>	<i>Eisner (3-mass.)</i>	<i>Eisner (2-mass.)</i>
A massecuite	80-90	50	51	60
B massecuite	65-80	45	47	45
C massecuite	58-65	40	38	

Exhaustion of the massecuite

It is interesting to express the efficiency of operation, not in weight of sugar per unit volume or unit weight of massecuite, but in weight of crystal recovered as percentage of sucrose in the massecuite". This ratio has the advantage of varying little with the massecuite considered, A, B or C. This recovery with A massecuite should be of the order of 60-65%, double purging giving values slightly superior to those obtained by single purging, 60% with B and 50% with C massecuite.

Exhaustion is given by a convenient expression^{11a} derived by the same reasoning as that of p. 839:

$$\text{Exhaustion} = \frac{10,000 (\text{massecuite purity} - \text{molasses purity})}{\text{massecuite purity} (100 - \text{molasses purity})} \quad (34.4)$$

Similarly:

$$\text{Crystal \% massecuite} = \frac{\text{massecuite purity} - \text{molasses purity}}{100 - \text{molasses purity}} \quad (34.5)$$

Installation of crystallisers

In most factories, the crystallisers are placed at ground level. This is a mistake which is detrimental to the recovery.

In such a case, much trouble has been taken, particularly with the third strike, to obtain a very tight massecuite. Hence it is inevitably thick, stiff, dense (sometimes wrongly called viscous) and difficult to handle, especially when it is cold, after crystallisation. The pumps refuse to handle it, and the result practically always is that the operators responsible for feeding the low-grade centrifugals add molasses, sometimes even water, to dilute the mass so that it will be more readily accepted by the pump. Hence there is a complete destruction

of all the work laboriously carried out, up to that stage, to obtain a dense massecuite; a destruction all the more complete because the operators have a tendency to abuse the dilution-water valve.

One palliative consists of having good pumps of Rota type, well located below the trough which supplies them, and fitted with short suction pipes of large diameter.

Another palliative consists of reheating the massecuite in the delivery trough, by furnishing this trough with a double bottom which is heated by exhaust steam, but this again involves some risk of re-resolution of crystal.

However, there exists only one elegant method of really and completely solving the problem; that is, to design each installation in such a way as to avoid and completely banish the handling of the massecuite by a pump. For this it is necessary to locate the pans at a higher level, above the crystallisers, which consequently will be located on the first-floor level, just below the pans; and the crystallisers in their turn should be above the distributor-mixer supplying the centrifugals, which then will be located at floor level, and just below their mixers.

This arrangement of the plant has an influence on the final recovery, the importance of which can hardly be exaggerated. The modifications to the installation which it might involve in a badly laid out factory would probably be amply repaid in the first season.

Ordinary crystalliser

The ordinary or "kneading-trough" crystalliser (Fig. 34.1) is a simple steel vessel, of U-shaped cross-section, fitted with an agitator permitting it to maintain the mass in slow and continuous motion.

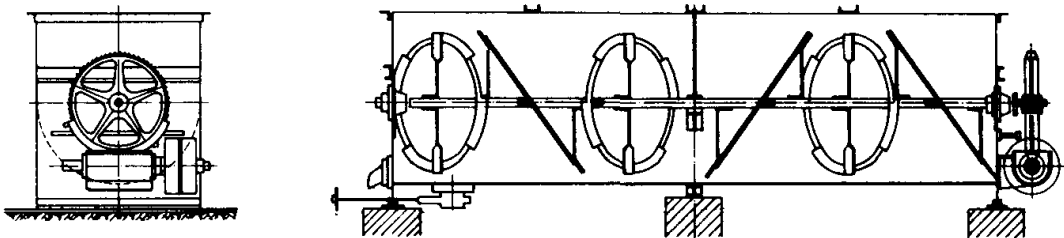


Fig. 34.1. Ordinary crystalliser.

Speed of rotation. Deerr¹² recommends a rotational speed of the screw of one turn in 1 min 45 s. Tromp² suggests $\frac{1}{2} - \frac{3}{4}$ r.p.m. This speed is not of very great importance, the best values being the lowest ($\frac{1}{2}$ r.p.m.). It has been found, by varying the speed, that it has hardly any perceptible effect as long as there is some movement.

The best values are accordingly the lowest, and the author would recommend $\frac{1}{2}$ r.p.m.

Power. The power required for the crystalliser depends on this speed of rotation. Deerr³ estimates about 25 kW/1,000 m³ (1 h.p./1,000 cu.ft.) of massecuite; Tromp² 1-2 kW (1.5-3 h.p.) per crystalliser, according to its volume (12-50 m³; 450-1,750 cu.ft.), or 6-12 m³/kW (300-600 cu.ft./h.p.).

Honig¹³ recommends that the installed power should be 210 kW/1,000 m³ (8.5 h.p./1,000 cu.ft.), in order to allow for the most unfavourable conditions. In Australia¹⁴ a figure of

375 – 400 kW/1,000 m³ (15 – 16 h.p./1,000 cu.ft.) is suggested.

For typical crystallisers, of about 20 – 50 m³ (700 – 1,800 cu.ft.) capacity, and for a speed of $\frac{1}{2}$ r.p.m., Honig's figure of 210 kW per 1,000 m³ may be used.

While crystallisers generally adopt the U shape, certain makers prefer a cylindrical cross-section, with an opening only of 600 mm (24 in.) at the top.

Time for crystallisation. Tromp¹⁵ gives the following times of crystallisation:

A massecuite: 12 h

B massecuite: 12 h

C massecuite: 72 h

General practice is to keep the A massecuite a very short time in the crystalliser, to give a short period of crystallisation for the B massecuite, and to keep as many crystallisers as possible for the C massecuite.

It is essential to provide long and careful crystallisation for the final massecuite, but there would be some advantage in doing the same for the higher massecuites. When ordinary crystallisers are provided, it is not recommended to go below the following times:

12 h crystallisation for A massecuite

24 h crystallisation for B massecuite

72 h crystallisation for C massecuite

Capacity. Deerr estimates a total capacity for the battery of crystallisers of 5 m³/t.c.h. (180 cu.ft./t.c.h.).

Tromp, from the above times, estimates the proportions of the respective massecuites as follows:

A massecuite: 50% = 85 l/t.c. (3 cu.ft./t.c.)

B massecuite: 28% = 45 l/t.c. (1.6 cu.ft./t.c.)

C massecuite: 22% = 40 l/t.c. (1.4 cu.ft./t.c.)

From these figures, and the crystallisation times just quoted, he calculates the required crystalliser capacities as:

$$85 \times \frac{12}{24} + 45 \times \frac{12}{24} + 40 \times \frac{72}{24} = 42.5 + 22.5 + 120$$

or:

$$185 \text{ l/t.c./24 h} = 44 \text{ hl/t.c.h. (160 cu.ft./t.c.h.)}$$

This is a theoretical figure. However, taking into account loss of time and allowing a necessary margin of safety, he finally quotes the following as practical figures for Cuba and

the Philippines; figures for other countries have been added (Table 34.4):

TABLE 34.4

CAPACITY OF CRYSTALLISERS, IN hl/t.c.h. (cu.ft./t.c.h.)

	<i>Minimum</i>	<i>Maximum</i>	<i>Mean</i>
Cuba (Tromp)	50 (180)	70 (250)	—
Philippines (Tromp)	—	—	75 (270)
Natal ¹⁶	40 (135)	170 (620)	—
Puerto Rico ¹⁷	17 (61)	58 (208)	33 (118)

In Cuba¹⁸, requirements are estimated as follows:

A massecuite:	6 hl/t.c.h. (21.5 cu.ft./t.c.h.)
B massecuite:	9 hl/t.c.h. (32.2 cu.ft./t.c.h.)
C massecuite:	30 hl/t.c.h. (107.5 cu.ft./t.c.h.)
or a total of:	45 hl/t.c.h. (161.2 cu.ft./t.c.h.)

Where figures are given for C massecuite only, which is the most important, we find¹⁹:

Queensland:	20 hl/t.c.h. (70 cu.ft./t.c.h.)
Hawaii:	84 hl/t.c.h. (300 cu.ft./t.c.h.)

In order to arrive at a figure among values varying so greatly, we would comment that the times of crystallisation which we have recommended above (12,24 and 72 hours) when allowing a safety margin of 25%, give figures of:

$$1.25 \left[95 \times \frac{1}{24} + 50 \times \frac{1}{24} + 35 \times \frac{72}{24} \right] \quad \begin{array}{ccc} A \text{ mass.} & B \text{ mass.} & C \text{ mass.} \\ 60 & + & 62 & + & 131 & \text{l/tc./24 h} \end{array}$$

or respectively:

$$14 + 15 + 32 \quad \text{hl/t.c.h.}$$

or a total of 60 hl/t.c.h. (222 cu.ft./t.c.h.).

Unit capacity. The unit capacity of the crystallisers, or capacity of each unit, should be fixed as a function of that of the pans. It is necessary to avoid mixing in the same crystalliser, crystals from two different pans, and also to avoid running a hot massecuite into a crystalliser partly filled with one which has already been cooled. The unit capacity of the crystallisers will thus be chosen equal to that of the pans supplying them, increased by

15–20% (crystallisers of 36 m³ for example, for pans of 30 m³). Alternatively, two crystallisers may be installed to serve one pan (two crystallisers each of 18 m³, for example, for a pan of 30 m³).

Calculation for cooling in an ordinary crystalliser

This calculation is made in a manner fully analogous to that which we have seen with relation to heaters (*cf.* p. 449). Let:

V = useful volume of crystalliser, in m³ (cu.ft.)

d = density of massecuite, in kg/m³

c = specific heat of massecuite (≈ 0.44)

P = weight of massecuite in crystalliser, in kg (lb.)

S = cooling surface of crystalliser, in m² (sq.ft.)

D = diameter of crystalliser, in m (ft.)

L = length of crystalliser, in m (ft.)

θ = ambient temperature, in °C (°F)

T_0 = initial temperature of massecuite (at $t = 0$), in °C (°F)

T = temperature of massecuite at time t , in °C (°F)

t = time of cooling, in hours

k = heat-transfer coefficient, in kcal/m²/°C/h (B.T.U./sq.ft./°F/h).

We assume the crystalliser to be of standard U section. Its cooling surface is equal to the area of the walls (ends and sides) of the vessel plus that of the surface of the massecuite.

Generally, the value of 7 (1.4) is taken for k^{20} . This is an overall approximate value, which allows for the massecuite surface, where the heat transfer is assisted by the absence of a metal surface, but adversely affected by convection.

Now:

$$S = 2 \times \frac{\pi D^2}{8} + 2 \frac{D^2}{2} + \frac{\pi D}{2} L + 2 \frac{D}{2} L + DL$$

or:

$$S = D \left[D \left(1 + \frac{\pi}{4} \right) + 2L \left(1 + \frac{\pi}{4} \right) \right] = D(D + 2L) \left(1 + \frac{\pi}{4} \right)$$

Let:

$$\frac{L}{D} = m \quad L = mD$$

We have:

$$S = D^2 \left(1 + \frac{\pi}{4} \right) (1 + 2m) \quad (34.6)$$

Similarly,

$$V = \frac{\pi D^2}{8} L + \frac{D^2}{2} L = \frac{mD^3}{2} \left(1 + \frac{\pi}{4} \right) \quad (34.7)$$

Hence:

$$\frac{S}{V} = \frac{2(1 + 2m)}{mD} \quad (34.8)$$

The formula for cooling, similar to that for heating, is:

$$\ln \frac{T_0 - \theta}{T - \theta} = \frac{kS}{PC} t \quad (34.9)$$

or:

$$T = \theta + (T_0 - \theta) e^{t(-kS/PC)} \quad (34.10)$$

But:

$$\frac{kS}{PC} = \frac{kS}{Vdc} = \frac{2k(1 + 2m)}{mDdc}$$

Hence:

$$T = \theta + (T_0 - \theta) e^{-(2k/dc) \cdot t(1 + 2m)/mD} \quad (34.11)$$

Replacing the quasiconstant expressions by their values, we thus have:

$$T = \theta + (T_0 - \theta) e^{-0.02 \cdot t(1 + 2m)/mD} \quad (T = \theta + (T_0 - \theta) e^{-0.07 \cdot t(1 + 2m)/mD}) \quad (34.12)$$

Now taking the mean or commonest values for the other terms:

$$\begin{aligned} T &= 70^\circ\text{C} \quad (160^\circ\text{F}) \\ &= 30^\circ\text{C} \quad (86^\circ\text{F}) \\ D &= 1.8 \text{ m} \quad (6 \text{ ft.}) \\ m &= 3.33 \end{aligned} \quad T = 30 + 40 e^{-0.027t} \quad (34.3,$$

we obtain the cooling graph as given in Fig. 34.2 ($^\circ\text{C}$).

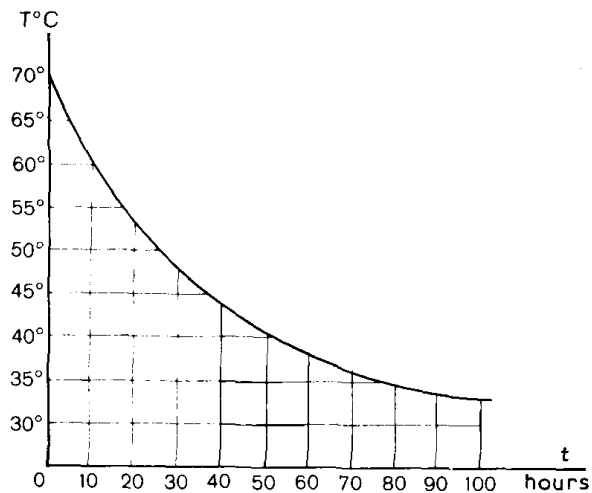


Fig. 34.2. Cooling curve for massecuite in ordinary crystalliser.

TYPES OF CRYSTALLISER

Crystalliser with double helix

These are *crystallisers* which are widely used in certain British countries. They are analogous to ordinary crystallisers, but carry two shafts and two helical stirrers rotating in opposite directions, being driven by two worm wheels from the same worm (Fig. 34.3). The upper helix is provided with a triple stirrer strip and one-third of its height is above the massecuite level.

Since the crystalliser is narrower in relation to its volume than the ordinary type, it offers a relatively greater cooling surface; and the upper helix brings thin layers of massecuite into contact with the air and so gives a more rapid cooling.

Contrary to an idea which was widespread until recently, it has been found that this method of stirring the massecuite does not present any disadvantage, and that the crystalliser with a double helix allows of a cooling time shorter by half than that required for ordinary crystallisers.

Reports from the Philippines²¹ indicate a reduction even of two-thirds in the time of crystallisation. There would, however, be a slight emulsification with air, which may

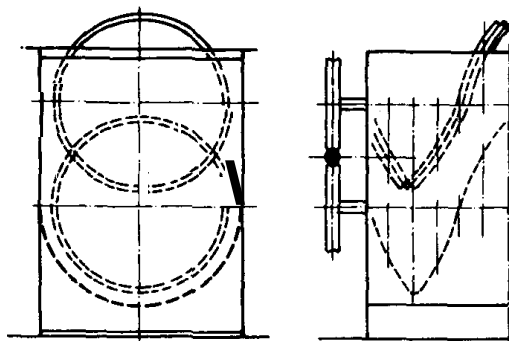


Fig. 34.3. Double-helix crystalliser.

sometimes be detrimental to the fugging. Any introduction of air bubbles into a massecuite, syrup or molasses leads to an increase in viscosity.

Tromp²² indicates that a capacity of 3.75 m³/t.c.h. (133 cu.ft./t.c.h.) would be sufficient for final massecuite with this type of crystalliser. It is presumed that this capacity is given for a duty equivalent to that which would be obtained with the 7.5 mVt.c.h. (266 cu.ft./t.c.h.) indicated above for all massecuites with ordinary crystallisers (*cf.* p. 731, Table 34.4).

Ragot crystalliser

The Ragot crystalliser is an ordinary crystalliser in which the stirring strip of the helix has been replaced by a coil carrying water.

This will obviously allow of rapid cooling, and also presents the advantage that reheating of the massecuite can also be carried out before fugging by replacing the cold water in this coil with hot water.

Water-cooled crystallisers

Modern water-cooled crystallisers are practically always designed with the cold-water inlet and hot-water outlet arranged in the shaft, which is then in the form of a central tube carrying the cold water, surrounded by an outer tube so that the hot water returns through the annular space between the inlet tube and this outer sleeve. The inlet and the outlet for the water are generally located on the one end of the crystalliser.

The heat-exchange surface is attached to the outer tube.

The form of this surface varies from one designer to another. In one system widespread in the American hemisphere, it consists of two concentric radial tubes. In the Fletcher - Blanchard (Fig. 34.4), the heat-exchange surface consists of tubes with closed ends, one series straight and one series curved, into which and from which the water flows during the rotation. In the Fives Lille crystalliser (Fig. 34.5), it is in the form of a hollow helical strip, which on the outside contributes to a lateral movement of the massecuite, and on the inside, like the preceding model, is traversed by the cooling water as the shaft rotates. Another model has spiral plates (Fig. 34.6).

All these crystallisers permit the same degree of cooling to be obtained as for the ordinary type of crystalliser, without any disadvantage and in a much shorter time, generally in 12–20 h, averaging say 16 h, as compared with 2–3 days. They thus afford a great economy in space required in the factory.

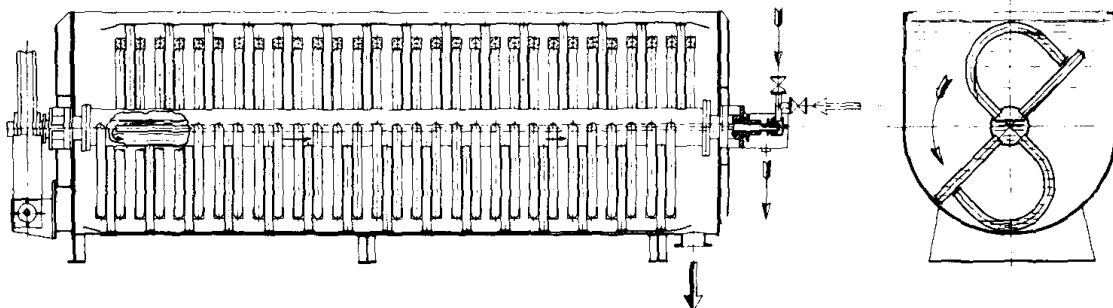


Fig. 34.4. Blanchard crystalliser, Fletcher patent.

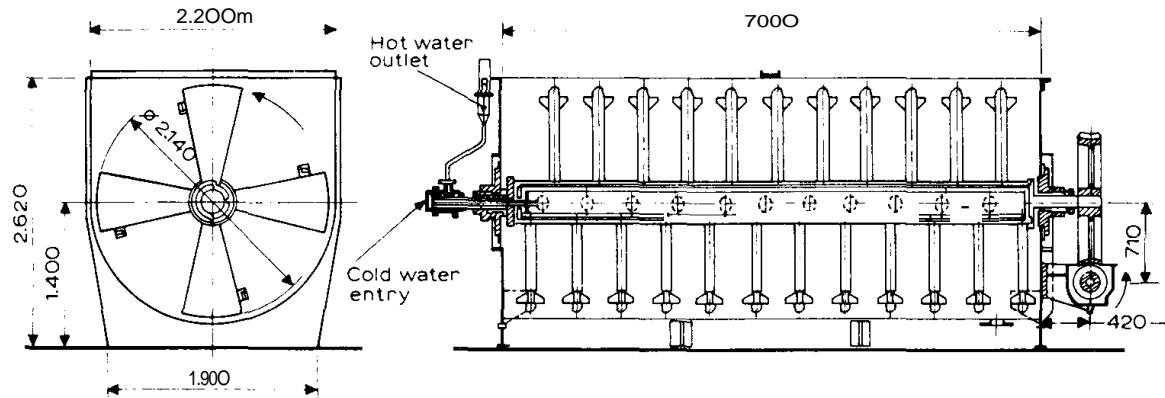


Fig. 34.5. Crystalliser with "flag" stirrers (Fives Cail – Babcock).

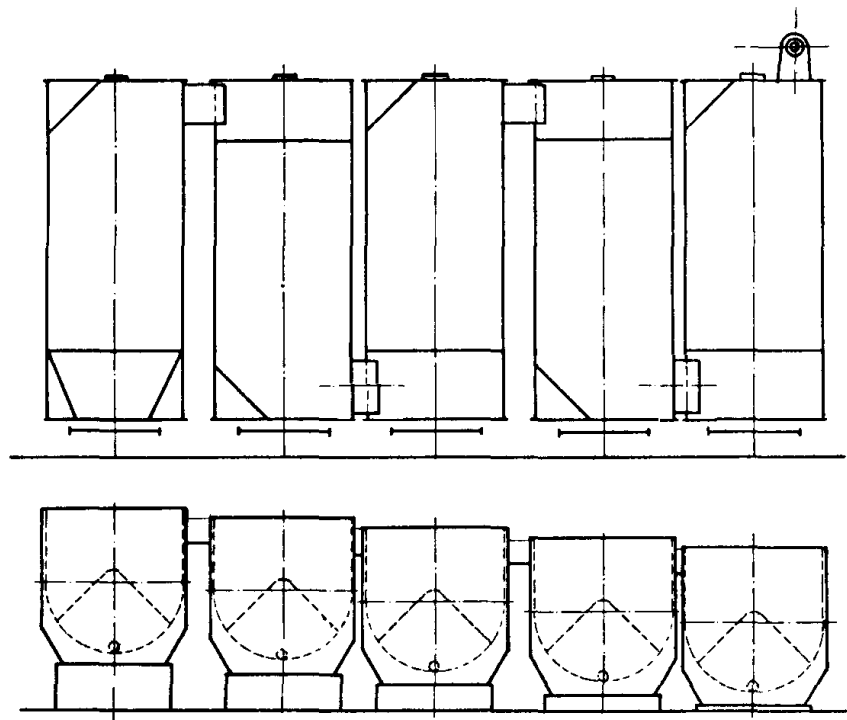


Fig. 34.6. Arrangement of crystallisers in series.

The cooling could be as rapid as desired, as the crystallisation reduces the saturation temperature by $2-5^{\circ}\text{C}$ ($4-8^{\circ}\text{F}$) per hour, while it is generally not possible to cool more rapidly than $1-1.5^{\circ}\text{C}$ ($2-3^{\circ}\text{F}$) per hour²³.

The rate of crystallisation decreases with decrease in purity or temperature. Crystallisation is thus more rapid at the beginning of cooling, and particularly higher with A and B than with C masseuite.

However, it is recommended that the masseuite should not be cooled immediately and abruptly as soon as it is discharged from the pan into the crystalliser. It will be left to mix for one to two hours before circulating cooling water, to avoid risk of formation of false grain.

It is agreed²⁴ that the heat-exchange surface for the cooling elements should be of the order

of 1–2.5 m²/m³ (0.3–0.8 sq.ft./cu.ft.) of capacity. Honig recommends that it should not exceed 2 m²/m³ (0.6 sq.ft./cu.ft.).

Capacity. In Hawaii, a capacity of 4.25 m³/t.c.h. (150 cu.ft./t.c.h.) is recommended for water-cooled crystallisers, compared with 8.5 (300) for ordinary crystallisers, for C massecuite; in South Africa^{24a} a total of 5.4 mVt.c.h. (190 cu.ft./t.c.h.) for all massecuites. Baikow²⁵ recommends for C massecuite 2.25 mVt.c.h. (80 cu.ft./t.c.h.), plus 2 crystallisers, one filling and one emptying.

Series, The series of Fives Cail-Babcock water-cooled crystallisers ranges from 23 to 60 m³ (800–2,100 cu.ft.) in twelve different sizes. The *S/V* ratio varies from 1.05 to 1.7 m²/m³ (0.3–0.5 sq.ft./cu.ft.); the speed is 1 r.p.m., with power of 0.075–0.1 kW/m³ (3 h.p./1,000 cu.ft.).

Vacuum crystallisers. In the operation of the boiling house, it may be necessary to hold in reserve storage some massecuites or magmas to be reprocessed later. For this purpose cylindrical crystallisers are used, completely closed and maintained under vacuum, with inlet pipes for syrup or molasses and sometimes a jacket for admission of steam, condensates being withdrawn from below. Speed of rotation of the agitators is generally 1 r.p.m. Apart from these features and the thickness of the walls, these crystallisers present no particular features.

Conditions of operation. B.M.A. recommend that the difference in temperature between the massecuite and cooling water should not exceed 12°C (22°F), otherwise crystals of sugar collect on the cooling elements and so reduce the efficiency of heat transfer.

They insist also that the cooling elements and any baffles should be designed so as to avoid any introduction of air into the massecuite, for the reason indicated above. They specify also that horizontal crystallisers do not operate satisfactorily unless the ratio of non-sucrose to water is lower than 3:

$$\frac{B - S}{100 - B} < 3 \quad (34.14)$$

Above that value, only vertical water-cooled crystallisers treat massecuite efficiently. They do not give rise to the short-circuiting which occurs in horizontal crystallisers, allowing pockets of massecuite to stagnate in certain corners, at the expense of exhaustion and regularity of crystals.

Crystallisers in series. When a battery of horizontal crystallisers is available, air- or water-cooled, it is advisable to arrange them to operate in series rather than individually. Assuming that they are of identical design and dimensions, they should then be located so that each member of the series is 140 mm (5½ in.) higher than the following (Fig. 34.6). The massecuite thus enters at one end of the first unit, leaves at the far end and overflows by a gutter, slightly inclined, to enter the second unit; continuing in the same way from the second into the third,

the massecuite circulates alternately to and fro along the separate crystallisers. The capacity of the units is thus fully utilised.

This arrangement unfortunately does not prevent the establishment of local currents and consequent stagnation of massecuite in the corners of the tank. This fault may be partially remedied by closing off and covering the dead corners by welded plates. For the same reason, a baffle is interposed in the upper part of each tank towards the middle point, to avoid local currents and oblige the massecuite to circulate to the bottom of the tank. The elliptical or spiral agitators contribute effectively to homogenising the massecuite. Care will of course be taken to see that these plates across the dead corners do not interfere with the rotation of the agitators.

Werkspoor crystalliser

This crystalliser (Fig. 34.7) has an exterior form similar to that of the ordinary crystalliser. However, the shaft, instead of carrying a helix, is fitted with discs (Fig. 34.8) with a gap in the form of a 45° or 60° sector, the cut-out parts of alternate discs being spaced 180° apart. Both shaft and discs are hollow and designed to permit circulation of water. The massecuite is introduced continuously at the end from which the shaft is driven, and flows along the crystalliser by gravity, passing from one space between discs to the next by means of the open sector of the disc, and overflows at the opposite end of the crystalliser; the cooling water

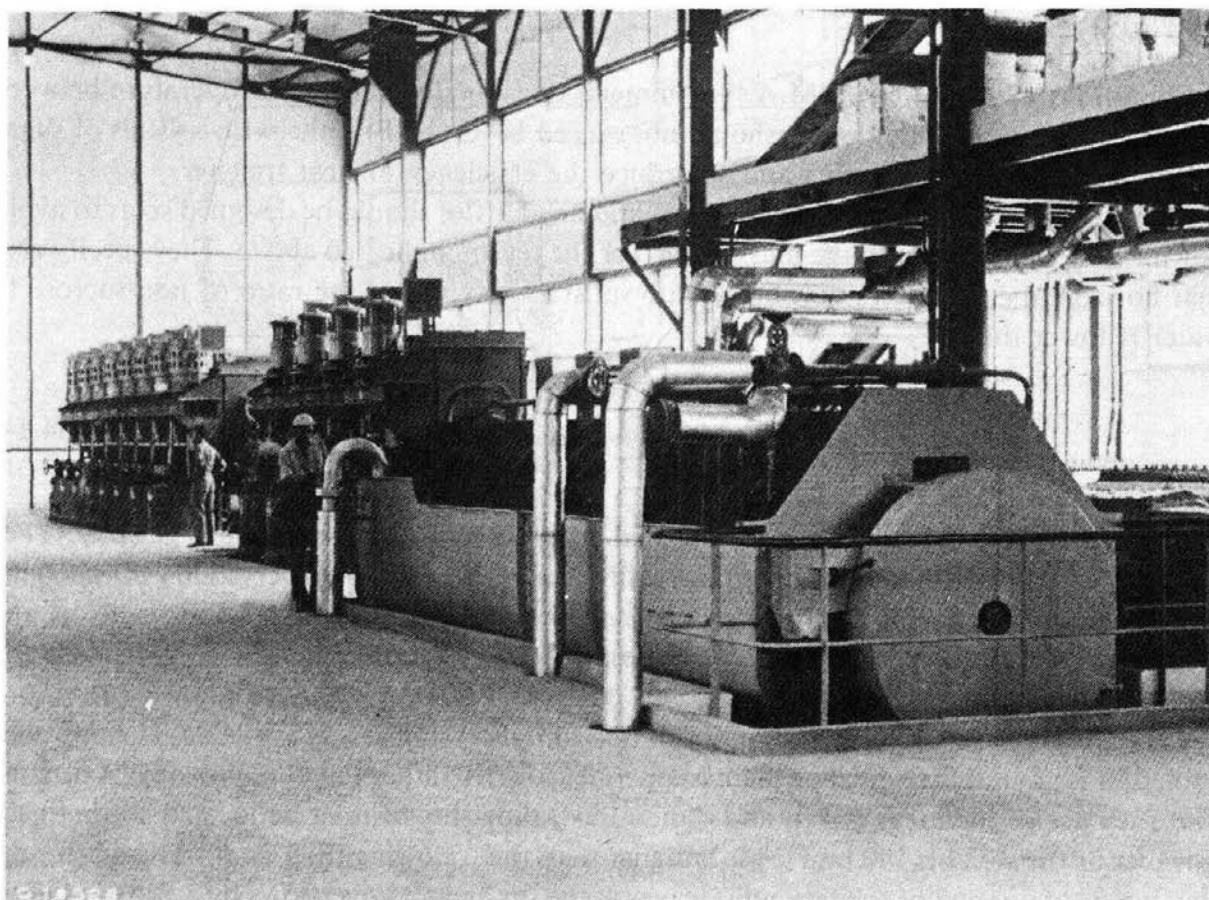


Fig. 34.7. Werkspoor crystalliser.

enters at the latter end, which is thus the outlet end for the massecuite, passes in succession through all the discs, and returns through a pipe or the hollow shaft of the crystalliser to leave at the end at which it entered.

This is therefore a *countercurrent* circulation. It presents the advantage that the hot massecuite entering comes into contact only with water which has already been heated, and that at any point the temperature of the cooling water is progressively lower as the massecuite becomes cooled (Fig. 34.9). These conditions practically eliminate risk of false-grain formation.

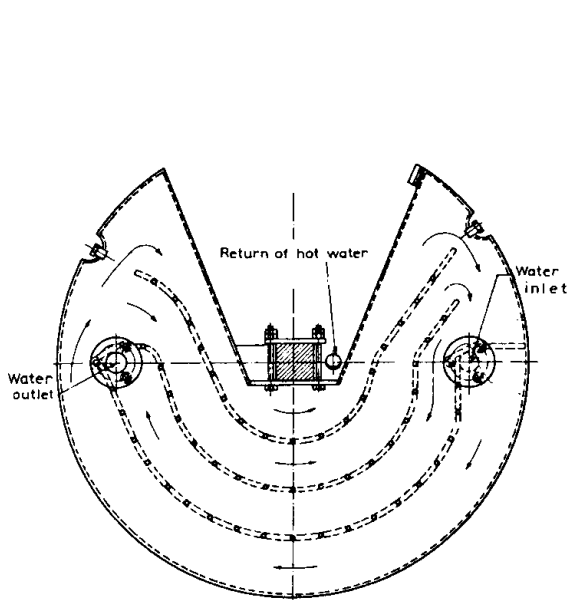


Fig. 34.8. Disc of Werkspoor crystalliser.

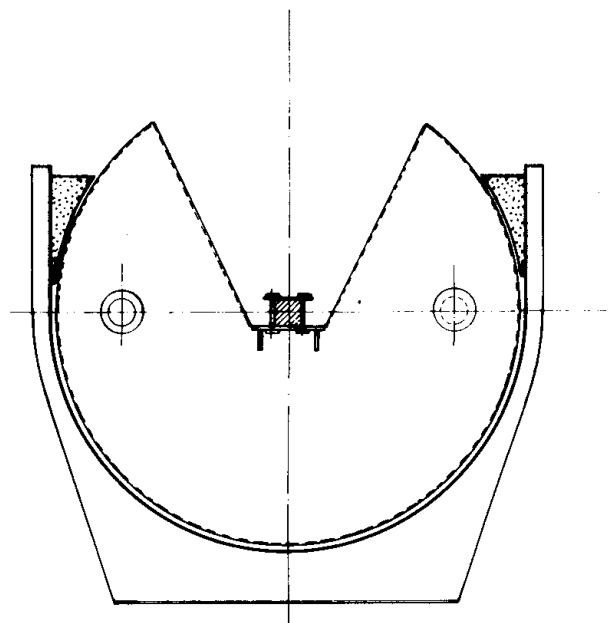


Fig. 34.8a. Massecuite baffles in Werkspoor.

Power required. This is about half of that indicated for the ordinary crystalliser (*cf.* p. 729). Honig²⁶ has observed that the power for Werkspoor crystallisers is also substantially lower than that required for crystallisers with the cooling water passing through tubes, and is of the order of half the requirements of the latter, whereas the heat transmission is of the same order. This is an important advantage when very heavy massecuites are being treated.

Quantity of water required. Let:

T_0 = temperature of the massecuite entering the crystalliser

T = temperature of the massecuite leaving

t_0 = temperature of water entering

t = temperature of water leaving.

The quantity of water required would be theoretically:

$$w = \frac{(T_0 - T)c_1}{t - t_0} \text{ kg of water per kg of massecuite} \quad (34.15)$$

c = specific heat of the massecuite = 0.44.

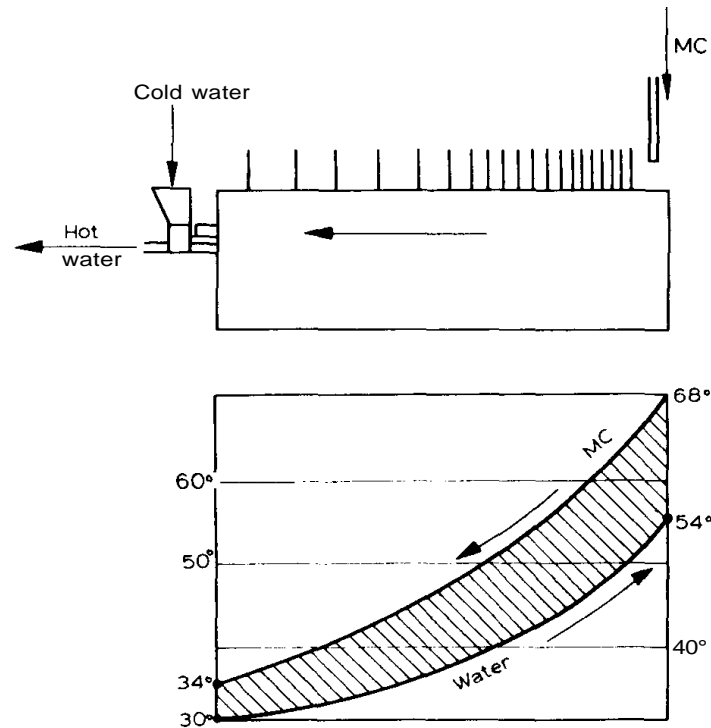


Fig. 34.9. Graph of temperatures in Werkspeer crystalliser.

In practice, we shall have:

$$W = \alpha Vdc \frac{T_n - T}{t - t_0} \quad (34.16)$$

W = total quantity of water required, in kg/h (lb./h)

a = coefficient taking into account the cooling of the molasses from the centrifugal which is added to the massecuite: a value of 1.15 – 1.20 is often taken

V = volume of massecuite to be treated, in m³/h (cu.ft./h)

d = density of the massecuite = 1,500 kg/m³ (94 lb./cu.ft.).

It is not necessary to consider losses of heat during the cooling operation, because:

(1) The water circuit is completely submerged in the massecuite, hence the efficiency is practically equal to unity.

(2) There is an approximate compensation between two associated thermal phenomena, which are secondary and which we shall neglect for this reason, namely:

(a) The massecuite is also cooled through the outer walls of the crystalliser, and through its surface exposed to the air.

(b) On the other hand, the cooling water must absorb, in addition to the sensible heat of the massecuite, the heat of crystallisation of the sucrose which deposits on the crystals during the process. This heat of crystallisation generally represents from 8 to 10% of the sensible heat involved.

However, the favourable effect (a) is generally somewhat greater than the unfavourable effect (b); consequently the apparent efficiency is slightly greater than 1.

In practice, it is found that the quantity of water used is of the order of:

$$w = 0.75 - 0.80 \text{ kg/kg of massecuite}$$

or approximately:

$$w' = 1.2 \text{ m}^3/\text{m}^3 \text{ of massecuite}$$

Tromp²⁷ gives $w = 0.8$ kg/kg, and the figures which he quotes correspond to a coefficient $a = 1.4$. He adds that the cooling should never be regulated by altering the ratio of water to massecuite, since this would disturb smooth operation. It is necessary on the contrary to maintain this ratio at 0.8 weight for weight.

Dilution. To facilitate circulation and crystallisation, the massecuite may be diluted in the crystalliser. For this purpose a molasses or syrup as close as possible to the mother liquor of the massecuite should be used, for example, the final molasses in a C massecuite. This is fed to the massecuite by a pipe above the crystalliser, with a nozzle above each disc. Since these nozzles tend to become blocked after stoppages or the weekly shutdown, it is recommended that the pipe be replaced by a notched horizontal trough, with a notch above each disc (or better, 3 notches, the central one in the axial plane of the disc). There is no disadvantage in continuing this dilution to the outlet end of the crystalliser; this does not affect the purity of the molasses from the centrifugals if the molasses used for dilution is close to the molasses purity.

Crystal deposits. The walls and the cooling elements tend to collect deposits of sugar which, on the elements, interfere with heat transmission and, on the walls, may end up by falling, obstructing the outlets and causing out-of-balance loads in the centrifugals. This disadvantage may be avoided by raising the temperature of the cooling water, by coating the walls with epoxy resin and by installing scrapers against the discs.

Sometimes a layer of denser massecuite adheres to the discs: this may be due to too rapid cooling. It can be detected by taking a sample of massecuite from the disc by hand: it is more viscous than the body of the material. Warm water may also be circulated through the discs for 3 minutes every 4 hours, in place of cold water; this prevents formation of this thick layer, as well as deposition of sugar.

Precautions. It is necessary to feed the massecuite into the side where the disc descends into the mass, not at the side where it emerges. If the crystalliser is filled up to the shaft, small partitions should be arranged in the space between the wall of the tank and the discs, in the plane of the discs.

Cooling surface. The cooling surface should be proportional to the capacity of the crystalliser, or more precisely, to the quantity of massecuite to be cooled per hour. It depends also on the massecuite temperature entering the crystalliser, the temperature of cooling water available, the degree of cooling required, etc. Tromp indicates as optimum the temperatures which we quote in Table 34.5 and which we have used in the graph of Fig. 34.9.

TABLE 34.5

WATER-COOLED COUNTERCURRENT CRYSTALLISERS: OPTIMUM TEMPERATURES (TROMP)

	(°C)	(°F)
Massecuite entering	68	154
Massecuite leaving	34	93
Water entering	30	86
Water leaving	54	129

Generally, the values for vacuum which we have recommended will give masseccuities leaving the pans at 75 - 80°C (167 - 176°F), and which, even after a period in the storage mixer, will still be at 70- 75°C (158 - 167°F). On the other hand, cooling the massecuite below 40°C (104°F) is often not attempted; and the temperature of cooling water is not a matter of choice.

The true coefficient of heat transfer in a Werkspoor has been established by the experiment station of Java and reported by Honig²⁸, who had participated in the determinations.

It was of the order of:

$$k = 35-50 \text{ kcal/m}^2/\text{°C/h} \text{ (7 - 10 B.T.U./sq.ft./°F/h)}$$

on masseccuities of high purity (> 70) with a new and clean crystalliser. A similar determination, made in Australia²⁹ on C masseccuities, with rather older crystallisers with some scaling, has given:

$$k = 12 \text{ kcal/m}^2/\text{°C/h} \text{ (2.5 B.T.U./sq.ft./°F/h)}$$

However, taking into account the favourable factors indicated above, designers generally apply, for the simplified method of calculation corresponding to eqn. (34.17), more optimistic apparent coefficients as given in Table 34.6. We add values which we consider it is prudent to use when considering performance over a normal period of years.

It is important, moreover, to point out that while with ordinary crystallisers the rotational speed has little influence on crystallisation, with water-cooled crystallisers, on the other hand, it does affect the heat-transfer coefficient³⁰.

TABLE 34.6

APPARENT HEAT-TRANSFER COEFFICIENT FOR WERKSPoor CRYSTALLISERS, IN kcal/m²/°C/h (B.T.U./sq.ft./°F/h)

	<i>Used by manufacturers</i>	<i>Recommended</i>
For A masseccuities	75 (15)	40-60 (8 - 12)
For B masseccuities	65 (13)	35-50 (7 - 10)
For C masseccuities	25 (5)	25 (5)

Calculation of cooling surface. We have:

$$S = a \frac{Vdc}{k} \cdot \frac{T_0 - T}{(T_0 - t) - (T - t_0)} \ln \frac{T_0 - t}{T - t_0} \quad (34.17)$$

- S = cooling surface of the Werkspoor, in m²(sq.ft.)
- a = factor taking into account the diluting molasses added
- V = volume of massecuite (before dilution), in l/h (cu.ft./h)
- d = specific gravity of the massecuite = 1.5
- c = specific heat of the massecuite = 0.40-0.44
- k = heat-transfer coefficient, given by Table 34.6
- T_0 = temperature of the massecuite entering the apparatus, in °C (°F)
- T = temperature required for massecuite leaving, in °C (°F)
- t_0 = inlet temperature of cooling water, in °C (°F)
- t = outlet temperature of water, in °C (°F)

We assume generally:

$$\begin{aligned} T_0 &= 75^\circ\text{C} (167^\circ\text{F}) \\ T &= 40^\circ\text{C} (104^\circ\text{F}) \quad \text{and} \quad a = 1.15 \\ t &= 50^\circ\text{C} (122^\circ\text{F}) \end{aligned}$$

We would recall that:

$$\ln x = 2.3 \log x$$

Cooling time. In the cane sugar factory, the cooling times allowed are generally as given in Table 34.7.

TABLE 34.7

COOLING TIMES GENERALLY ALLOWED IN WERKSPOORS

A massecuite (if boiling 3 massecuites)	1½ h
1st massecuite (if boiling 2 massecuites)	2 h
B massecuite (if boiling 3 massecuites)	2 h
C or final massecuite	12 – 15 h

Standard dimensions. The Werkspoor licence for France has been acquired by the Compagnie de Fives Lille. This firm manufactures crystallisers for A massecuite of which the diameter D is either 2.8 or 4 m. The discs have an outside diameter D' about 20-30 mm (1 in.) less than the interior diameter D of the vessels ($D' = 2.78$ m, for example, for a crystalliser of $D = 2.80$ m). The open sector of the discs is 45° or 60°, and their unit cooling surface s may be calculated approximately by:

(a) 45° sector, square arms:

$$S = (1 + 0.1D) D'^2 (1 + 0.03D) D'^2 \quad (34.18)$$

S = cooling surface of one disc, in m^2 (sq.ft.)

D' = exterior diameter of a disc, in m (ft.)

D = inside diameter of vessel, in m (ft.).

(b) 60° sector, round shaft:

$$s = 1.225 D'^2$$

The height of massecuite level above the axis is about $1/20 - 1/10$ of the diameter. However, the capacity of the crystalliser is generally estimated as if the tank contained no discs, shaft, or pipes and assuming that the massecuite level was limited to the axis (which amounts to assuming that the volume of massecuite above the axis compensates for the volume occupied by the shaft, the discs, etc.). For the working volume, so defined per unit length of the tank, we have therefore:

$$u = \frac{\pi D^2}{8} = 0.3927 D^2 \quad (34.19)$$

u = theoretical working volume per unit length of the crystalliser, in m^3/m (cu.ft./ft.)

D = diameter or interior width of the tank, in m (ft.).

In the latest models, where the level of massecuite is kept well above the shaft, generally about $0.3D$ above the shaft, the useful volume of massecuite above the axis may be reckoned as $0.3D^2L$.

The tank of the crystalliser is generally proportioned so that the length is about $3 - 3.5$ times its diameter, without allowing this secondary consideration to override the eventual requirements of space required and difficulties of installation.

The space between discs increases from inlet to outlet of massecuite, but the mean value of this spacing should not fall below 20 cm (8 in.). It is determined by the formula:

$$e = \frac{L - 0.5(1 + p)}{n - 1 - p} \quad \left(\frac{L - 1.6(1 + p)}{n - 1 - p} \right) \quad (34.20)$$

e = average spacing between discs, in m (ft.)

L = total length of the crystalliser tank, in m (ft.)

p = number of intermediate bearings (the length between bearings should be at least 3 m (10 ft.) and at most 6 m (20 ft.))

n = number of discs (which should always be an odd number).

For equal performance, it is more expensive than a vertical crystalliser, due to the cost of fabricating the discs.

Design of a Werkspoor crystalliser. *Data.* We shall assume we are dealing with a factory working at 200 t.c.h., employing a 3-massecuite double-magma system, and obtaining 125 1 (4.5 cu.ft.) of A massecuite per t.c. We require to calculate the dimensions of a Werkspoor crystalliser intended to treat the A massecuite.

Volume and dimensions of the Werkspoor. The quantity of massecuite to be treated is 125 l/t.c., or a total volume of:

$$V = 125 \times 200 = 25,000 \text{ l/h} \quad (900 \text{ cu.ft./h})$$

which will be increased by the addition of molasses for diluting to:

$$V = 1.15 \times 25,000 = 28,750 \text{ l/h} \quad (1,035 \text{ cu.ft./h})$$

The time for cooling will be 1 hour 30 min, according to Table 34.7. Hence the capacity of the crystalliser:

$$C = 28,750 \times 1.5 = 43,125 \text{ l} \quad (1,552 \text{ cu.ft.})$$

We assume the choice of a Fives Cail - Babcock Werkspoor. These are supplied in only two diameters: 2.8 and 4 m (9.2 and 13 ft.). The 4-m size being much too large for a factory of 200 t.c.h., we choose the smaller size. The discs will thus have a diameter of 2.78 m, and the useful volume of the crystalliser will be:

(a) The lower half-cylinder: $u' = 0.3927 D^2 L$

(b) The volume above the shaft: $u'' = 0.30 D^2 L$

Hence:

$$V = (0.3927 + 0.30) D^2 L = 0.6927 D^2 L$$

and we require:

$$V = 0.6927 \times 2.8^2 \times L = 43,125 \text{ l} \quad (1,552 \text{ cu.ft.})$$

To find the useful volume, we deduct the volume lost, due to:

(a) The shaft; we assume it cylindrical, of 0.30 m diameter:

$$v' = 0.7854 \times 0.3^2 \times L = 0.0707 \text{ m}^3/\text{m}$$

(b) The discs; we assume that they are spaced at intervals of 200 mm, the minimum value generally employed, and have a thickness of 50 mm. Thus there are 5 per m; if the open gap is 45°, the volume occupied by the discs will be:

$$v'' = 0.7854 \times 2.78^2 \times \frac{315}{360} \times 0.05 \times 5 = 1.3278 \text{ m}^3/\text{m}$$

The total volume lost is thus:

$$v' + v'' = 0.0707 + 1.3278 = 1.3985 \text{ m}^3/\text{m}$$

The useful volume is thus:

$$(0.6927 \times 2.8^2 - 1.3985) L = 43.125 \text{ m}^3 \quad L = \frac{43.125}{4.0523} = 10.70 \text{ m (35.78 ft.)}$$

Cooling surface. If we assume that the massecuite has cooled to 70°C (158°F) in the storage mixer, and that cooling water is available at 25°C (77°F), and if we accept 50° (112°F) as the exit temperature of the warm water, and assume a heat-transfer coefficient of 60 kcal/m²/°C/h (12 B.T.U./sq.ft./°F/h), the cooling surface required (eqn. (34.17), taking 0.4 as the specific heat of the massecuite, will be:

$$s = 1.15 \times \frac{25,000 \times 1.5 \times 0.4}{60} \times \frac{70 - 40}{(70 - 50) - (40 - 25)} \times \ln \frac{70 - 50}{40 - 25} = 495 \text{ m}^2 \text{ (5,430 sq.ft.)}$$

Surface area of a disc. (360° - 60° = 300°).

$$s = 1.225 \times \frac{5}{300} \times 2.78^2 = 9.94 \text{ m}^2 \quad (107 \text{ sq.ft.})$$

Number of discs.

$$n = \frac{495}{9.94} = 49.8$$

(Calculation in British units gives 50.7.) We assume 51 discs. Checking the mean spacing (eqn. (34.20)), we have:

$$e = \frac{L}{n - 2} \times 2 = 0.198 \text{ m} \quad (0.66 \text{ ft.})$$

To avoid a spacing closer than 200 mm, we make the length of the crystalliser 10.80 m (35.4 ft.).

Water consumption. We have (eqn. (34.16)):

$$W = 1.15 \times 25,000 \times 0.4 \times \frac{70 - 40}{50 - 25} \times 1.5 = 20,700 \text{ l/h} \quad (46,500 \text{ lb./h})$$

It is seen that with the temperatures adopted, the quantity of cooling water necessary is approximately 1 l per h for each l/h of massecuite to be cooled. In other words, the volume of water required is roughly equal to the volume of massecuite to be cooled.

Checking an existing Werkspoor. When a Werkspoor is installed and it is desired to know what temperatures of massecuite and water it would give for the quantities of massecuite to be cooled and with the available cooling water, we have:

$$\frac{T - t_0}{T_0 - t} = e^{-(kS/\alpha Vdc) + (kS/W)} = m$$

$$\frac{t - t_0}{T_0 - T} = \frac{\alpha Vdc}{W} = r$$

(the symbols having the same values as in eqns. (34.16) and (34.17); hence we have:

$$T = t_0 + m (T_0 - t)$$

$$t = t_0 + r (T_0 - T)$$

Solving for T and t :

$$T = \frac{t_0 (1 - m) + mT_0 (1 - r)}{1 - mr} \tag{34.21}$$

$$t = t_0 + r (T_0 - T)$$

We may recall that:

- (1) If $y = e^{-x}$, then $-x = \ln y = 2.3 \log y$
- (2) $\text{Log } e^{-x} = -0.4343x$
- (3) If a negative logarithm is found, for example, $\log a = -0.372$, we shall write: $\log a = T.628$.

Results. Table 34.8 gives some results which we have obtained, operating with 3 massecuites, with a Werkspoor handling the first 2 massecuites, A and B:

TABLE 34.8

EXAMPLE OF RESULTS WITH A WERKSPOOR CRYSTALLISER		
	A massecuite	B massecuite
<i>When dropped from pans</i>		
Brix of massecuite	94.4	95.6
Apparent purity of massecuite	80.2	65.5
Apparent purity of mother liquor	61.5	50.1
<i>Leaving the Werkspoor</i>		
Purity of molasses (no washing)	56.7	46.2
Total purity drop	23.5	19.3
Purity drop of molasses	4.8	3.9
Temp. of mass. entering crystalliser, in °C (°F)	72.5 (163)	72 (162)
Temp. of mass. leaving crystalliser, in °C (°F)	46 (115)	48 (118)
Temperature of cooling water, in °C (°F)	24 (75)	27 (81)
Cooling time	1 h 40 min	1 h 45 min

In Java³¹ figures of the following order were reported:

Cooling: 25°C (45°F) in a period of 2 h 30 min
Purity drop of the mother liquor: 5°

Storage vessel. Since the Werkspoor operates continuously, and the pans batchwise, it is obviously necessary to provide an intermediate vessel to receive the massecuite between the pans and the Werkspoor.

For this duty an ordinary mixer is used (or sometimes two). It is necessary that its capacity should be 1.5 times that of the largest pan discharging into it.

Spacing between discs. We know that:

- (1) The speed of crystallisation is greater at higher temperatures.
- (2) This speed increases with the supersaturation, but in practice attains a maximum at a relatively low supersaturation
- (3) Viscosity increases with supersaturation

It is therefore of advantage to work with a relatively low supersaturation and to cool as rapidly as possible the hot massecuite at the inlet, since it permits of a maximum rate of crystallisation.

It is for this reason that the Werkspoor progressively increases the spacing between cooling discs from the massecuite entry to the massecuite outlet.

Reheating. The massecuite is reheated before fugalling, with the object of reducing its viscosity. The Werkspoor is well adapted to this operation; it is sufficient to provide a little extra length, so as to enable three or four reheating discs to be placed after the cooling discs. The calculation for this reheating portion is made in a similar manner to that for the cooling portion. However, the reheating should be rapid and of short duration; hence the reheating discs are placed close together.

In order to avoid re-resolution of sugar in the neighbourhood of the heating surface, it will be advisable to use, both in the Werkspoor and in any other reheating equipment used before fugalling, a heating medium of relatively low temperature, for example, bled vapour at about 75°C (167°F) (or 46-48 cm (18- 19 in.) vacuum), or preferably water at a temperature very slightly above the saturation temperature of the massecuite. Payne³² estimates that this water should not be at a temperature higher than 2°C (4°F) above the saturation temperature, to avoid risk of local overheating.

Use of the Werkspoor. Compared with ordinary crystallisers, the Werkspoor offers the advantage of taking up much less space and of improving the exhaustion. The latter advantage is perhaps most marked with A and B massecuites; with final massecuite, its use is somewhat more critical. However, notably in Queensland, it is normally used with low-grade massecuites, and with excellent results when used with efficient reheating arrangements.

Vertical cooler-crystallisers

Massecuites leave the vacuum pans at 75 -80°C (165 - 175°F). If these are ordinary batch

pans, the massecuite goes to holding crystallisers, where it cools to about $70-75^{\circ}\text{C}$ ($160-165^{\circ}\text{F}$) and from which it goes to the crystallisers proper. If the pans are of continuous type, the massecuite can go directly to the crystalliser. It is generally cooled to about 40°C (105°F) but is then too viscous for centrifuging. It is then reheated to $50-52^{\circ}\text{C}$ ($122-125^{\circ}\text{F}$) for feeding to the centrifugals.

The solubility of sucrose decreases progressively as the temperature is reduced, and as mixing favours contact between crystals and mother liquor, its passage through the crystallisers permits completion of formation of crystals and exhaustion of the molasses. The most effective equipment for performing this function is the vertical crystalliser-cooler. Werkspoor crystallisers would perform this function equally well but, for equal capacity, they are appreciably more expensive.

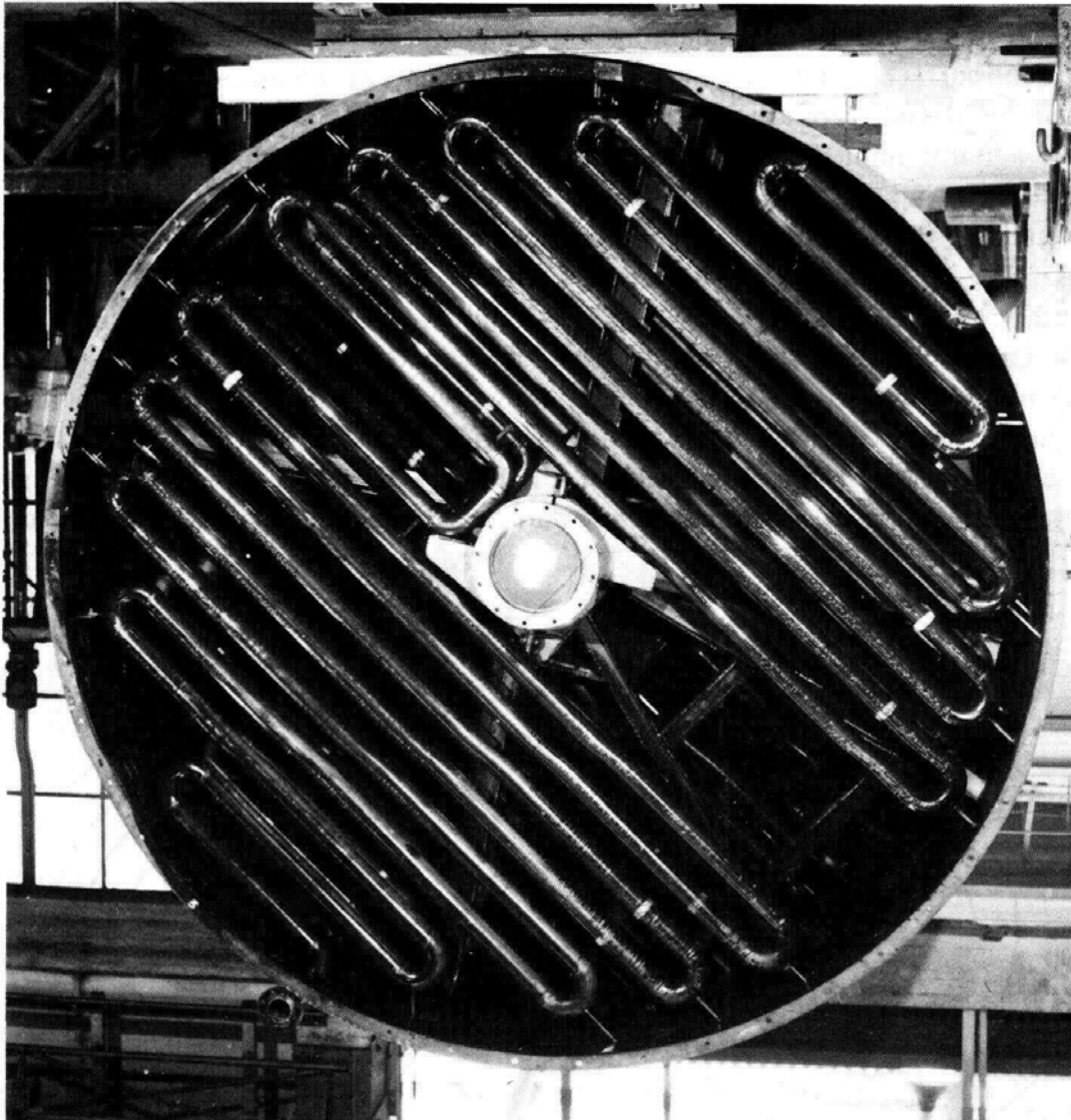


Fig. 34.10. View of the upper bank of cooling elements of a vertical crystalliser.

Description. Vertical crystallisers are manufactured by SEUM and Fives Cail—Babcock in France, by B.M.A. in Germany, and in several other industrial sugar countries.

They consist of a cylindrical tank with a vertical axis, in steel plate 8–12 mm in thickness, 3.60–5.20 m (12–17 ft.) in diameter and 7–16 m (23–52 ft.) in height. The roof of the tank supports the drive gear for the stirrers and also provides a man door and massecuite entry. A vertical shaft, consisting of a large-diameter pipe, is suspended from a substantial bearing and supported at the bottom by a thrust bearing; it carries at regular intervals several groups of horizontal stirrers covering practically the full interior diameter of the tank, alternate stirrers being set at 90° spacings. In the French designs, the cooling elements consist of horizontal layers of parallel pipes, 6–19 in number, connected by U-bends and filling the interior circle of the crystalliser (Fig. 34.10). These layers of tubes are connected one to the other by pipes outside the crystalliser. In neighbouring layers, the pipes are alternately at 90° to those above and below.

In the B.M.A. equipment, the cooling elements consist of discs with a gap of 30 or 60° , consisting of two sheets of metal separated by vertical ribs in the form of concentric circles covering the major part of the remaining 300 or 330° , and forcing the water to cover practically the full surface of the disc (Fig. 34.11). The elements are connected in series by suitable pipes, and the 30 – 60° gaps are displaced in successive discs by 90° in the anticlockwise direction. The agitator immediately below rotating in the clockwise direction thus assists the massecuite flow to pass to the lower stage by the gap in the disc. In the case of an upward-flow crystalliser, the displacement of the gap in the disc is in the opposite direction, that is, clockwise. The pipes are of 76.2 mm (3") diameter, and thickness 2.9 mm.

In the Fives Cail - Babcock and SEUM crystallisers, the drive of the stirring gear is effected by means of 3 pins fixed on a disc at 120° , actuated by 2 jacks which act alternately on these pins producing a 60° movement, thus giving a complete revolution in 6 such movements. The jacks are actuated by a hydraulic system powered by electrically driven pumps serving the group of crystallisers. In the SEUM drive, the driving disc is controlled by a clutch, the setting of which may be regulated according to the resistance offered by the

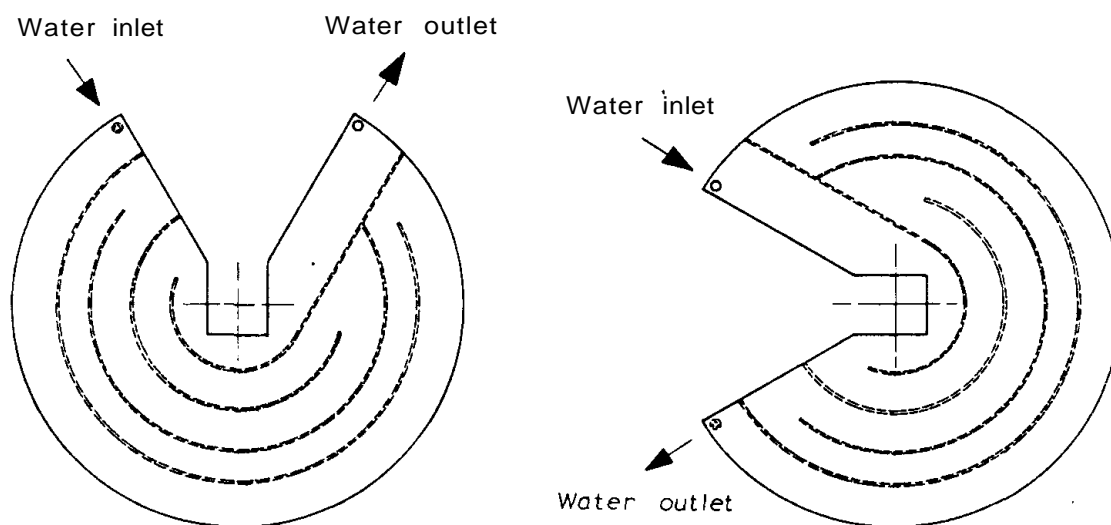


Fig. 34.11. Horizontal section of a cooling disc of a B.M.A. vertical crystalliser with discs.

massecuite; thus the degree of tightness of the massecuite may be controlled. The stirrers do not stop, as a valve slows the mechanism down when it approaches the maximum resistance. The **B.M.A.** drive is effected by an electric motor with reducing gear. A “diluter” is mounted on top of the crystalliser; this communicates with the different stages by exterior pipes, and permits dilution with molasses or water.

Particulars of the Fives **Cail - Babcock**, **SEUM** and **B.M.A.** are given in Tables 34.9, 34.10 and 34.11, respectively. The Fives **Cail - Babcock** series offers adjustable speed up to 0.6 r.p.m., while the **B.M.A.** series offers a choice of 2 heat-exchange surfaces for each size of crystalliser.

TABLE 34.9

FIVES CAIL - BABCOCK VERTICAL CRYSTALLISER-COOLER

Capacity (m ³)	80	100	120	150	175	200	300
Height (m)	7	8.2	9.7	11.7	13.6	11.3	16
Diameter (m)	4.45	4.45	4.45	4.45	4.45	5.20	5.20
Rows of tubes	6	7	8	13	15	12	19
Heat-exchange surface (m ²)	86	100	115	187	218	250	375
Power (kW)	11	15	15	19	22	26	38

TABLE 34.10

SEUM VERTICAL CRYSTALLISER-COOLER HEATERS

Capacity (m ³)	150	179	200	227	275	310
Height of massecuite (m)	10.	11.95	13.50	10	12.10	13.70
Diameter (m)	4.50	4.50	4.50	5.50	5.50	5.50
Rows of tubes	12	15	17	12	15	17
Heat-exchange surface (m ²)	190	237	269	293	366	415
Torque (m kg)	17,000	25,000	30,000	30,000	40,000	40,000
Power (kW)	15	19	23	23	38	38

TABLE 34.11

B.M.A. VERTICAL CRYSTALLISER-COOLER

Capacity (m ³)	100	150	185
Height (m)	10	12	12
Diameter (m)	3.6	4.5	4.5
Heat-exchange surface (m ²)	93 or 186	150 or 300	175 or 350
Power (kW): <i>n</i> = 0.8 r.p.m.	11	15	15
<i>n</i> = 0.55 r.p.m.	7.5	11	11

Application. Vertical crystallisers may be grouped in series; they are often installed in threes, the massecuite descending in the first, rising in the second, and descending again in the third; this avoids pumping between the different units. However, since the density of massecuite increases as it cools, it is more logical to pump it to the top and let it descend in all units. The three cooling units may be followed by a fourth used for reheating before fuggalling, in which the massecuite naturally rises while being heated.

The temperature sought in cooling massecuites is of the order of 40°C (105°F). To obtain such a result with low-grade material, it is essential that cooling should be slow. Generally, it is recommended that the cooling rate should not be greater than 0.8–1°C (1.5–1.8°F) per hour, which requires 36–50 hours, and limits the mean temperature difference between massecuite and water to 15°C (27°F). The volume of vertical crystallisers facilitates maintaining these conditions. If a temperature difference of 25°C (45°F) could be tolerated, a cooling rate of 3°C/h (5°F/h) could be maintained. Some authors state that a rate of 2°C/h (3.5°F/h) could be maintained without producing false grain, while maintaining a supersaturation of 1.2. Care should be observed in attempting this; according to B.M.A., massecuites with a ratio of 4.5 of non-sugars to water may be treated without difficulty, which indicates the high efficiency of their equipment.

Advantages. Vertical crystallisers appear destined to replace other types, even the *Werkspoor*, on account of their advantages:

- (a) Large volume for small floor space occupied
- (b) Suitability for installation out of doors
- (c) High efficiency, due to better contact between massecuite and heat-exchange surface (high S/V)
- (d) Facility of automatic regulation
- (e) Little or no short-circuit flow
- (f) Very good thermal efficiency, leading to improved purity of molasses
- (g) The efficacy of these crystallisers is such that a viscosity of 12,000–13,000 poises may be obtained in the outgoing massecuite; normally a value of 9,000—10,000 is readily obtained
- (h) Cost is reduced.

When two or three crystallisers are available, one-third, half or two thirds of the last one may be used for reheating the massecuite before centrifuging.

Design for vertical crystallisers for C massecuites

We consider here the requirements for a vertical-crystalliser station for cooling and reheating of C massecuite; conditions assumed are a factory of 250 t.c.h. producing 40 l of C massecuite/t.c. (1.43 cu.ft./t.c.), or 10,000 l/h, amounting to 11,500 l/h (415 cu.ft./h) by addition of 15% of molasses for dilution. The temperature of massecuite from the pan is assumed as 75°C (165°F), to be cooled to 40°C (105°F) and then reheated to 50°C (110°F) before fuggalling. At 1°C per hour, we require 35 hours, hence a crystalliser capacity of 400 m³ (14,200 cu.ft.).

If we choose crystallisers of the SEUM or Five Cail–Babcock type of 150 m³ (5,300 cu.ft.) each, we require 3 units of 150 m³ ($D = 4.45$ m, $H = 10.5$ m), of which the first two would be used for cooling and, of the third unit, one-third would be used for cooling and the re-

maintaining two-thirds for reheating. Following the recommendations above, these would be arranged for downward flow in the first and second units, with a pump between, and upper flow in the third unit, which is connected to the bottom of the second by a very short pipe.

These crystallisers have 12 or 13 layers of heat-exchange tubes each of $14.4 - 15.8 \text{ m}^2$ ($155 - 170 \text{ sq. ft.}$) of heat-exchange surface, or a total of 190 m^2 ($2,044 \text{ sq. ft.}$) per crystalliser. In these designs approximately 1 m^2 of tube surface is installed per m^2 of cross-section of the equipment. The interval between layers of tubes is approximately 800 mm (32 in.). The heat-transfer coefficient may be taken as $25 - 30 \text{ kcal/m}^2/\text{°C/h}$ ($5 - 6 \text{ B.T.U./sq.ft./°F/h}$) for the tubular units, and 10 (2) for the walls of the tank.

Cooling. Since the areas of walls and tubes increase linearly along the travel of the masscuite, we can add the effects of the two coefficients, while remembering that they apply



Fig. 34.12 Series of three vertical crystallisers (B.M.A.).

to different areas. For the reheating, on the other hand, only the former, for tubes, is relevant.

The upper limit permissible for the temperature difference between massecuite and water is 15°C (27°F) otherwise crystals may deposit on the tubes and cause a marked drop in efficiency of heat transfer. Accepting this figure, we design for a temperature of 60°C (140°F) for the water leaving the crystalliser, with water entering at 25°C (77°F).

The area of tubes for cooling is $190 \times 2.3 = 437 \text{ m}^2$ (4,700 sq.ft.) and the area of the walls $\pi D \times 10.5 \times 2.3 = 338 \text{ m}^2$ (3,635 sq.ft.).

The heat quantity (total heat transfer per hour) coming into play is:

$$11,500 \times 1.5 \times 0.44 (75 - 40) = 265,450 \text{ kcal/h (1,071,566 B.T.U./h)}$$

This corresponds to:

$$(a) \text{ Tube layers: } 437 \times 15^\circ \times 25 = 163,875 \text{ kcal/h (649,728 B.T.U./h)}$$

$$(b) \text{ Walls: } 338 \times 30^\circ \times 10 = 101,400 \text{ kcal/h (402,284 B.T.U./h)}$$

$$265,275 \text{ kcal/h (1,052,012 B.T.U./h)}$$

We assume a temperature difference of 30°C (54°F) for the walls, taking the mean massecuite temperature as $0.5(75 + 40) = 57.5^\circ\text{C (135.5}^\circ\text{F)}$, and ambient temperature as $27.5^\circ\text{C (81.5}^\circ\text{F)}$.

Quantity of water required. The quantity of water to be used per kg of massecuite is:

$$w = \frac{(75 - 40) 0.44}{60 - 25} = 0.44 \text{ kg/kg massecuite}$$

and the total quantity:

$$W = 11,500 \times 1.5 \times 0.44 \frac{75}{60 - 25} \sim \frac{60}{25} = 7,590 \text{ kg/h (17,000 lb./h)}$$

0.44 = specific heat of massecuite

1.5 = density of massecuite (93.6 lb./cu.ft.).

Reheating. If we were to use for reheating the water leaving crystalliser No. 1, the quantity would be insufficient to obtain the required temperature rise of 10°C (18°F). It will be necessary to pass this water through a small heater to raise its temperature to something above 60°C (140°F) and compensate the quantity by the higher temperature. The water will in any case lose 10°C, since:

$$7,590 \text{ kg water} \times 10^\circ\text{C} = 11,500 \times 1.5 \times 0.44 (50 - 40) = 75,900 \text{ kcal}$$

It is a question of finding to what temperature t' this water should be heated to obtain the 10°C rise required, by means of the 133 m² of crystalliser surface reserved for reheating. As the coefficient of heat transfer is 30 kcal/m²/°C/h (6 B.T.U./sq.ft./°F/h), and the mean temperature of the massecuite between 40 and 50°C is 45°C, we require:

$$133 \times 30 (t' - 45) = 75,900 \quad t' = 64^\circ\text{C} (147^\circ\text{F})$$

It will be sufficient to heat the water from 60 to 64°C (140 to 147°F) in order to heat the massecuite to 50°C. Care will be taken to insulate the upper two-thirds of the reheating crystalliser.

Coefficients. We recall the values of the heat-transfer coefficients involved in these calculations, with the possible range of variations in practice:

Specific heat of massecuite:

Used in calculations:	0.44
Quoted by Honig ³³	0.525, 0.45 – 0.50
Quoted by other authors:	0.35-0.4

Precise values may be calculated by formulae (33.48) and (36.3).

Heat-transfer coefficient:

	(kcal/m ² /°C/h)	(B.T. U./sq.ft./°F/h)
Tubes (heating or cooling):	25-30	5-6
Walls:	10	2

Precise figures. When the temperatures differ by more than those assumed, the simple calculation above is not satisfactory. It is then necessary to use the precise formulae:

Heat transferred to the massecuite:

$$M = \alpha Vdc (T_0 - T)$$

Heat-exchange surface required:

$$S = \frac{\alpha Vdc}{k} \frac{T_0 - T}{(T_0 - t) - (T - t_0)} \ln \frac{T_0 - t}{T - t_0}$$

The calculation is often made also in the following manner: the logarithmic mean temperature Δt_m between hot and cold massecuite is calculated:

$$\Delta t_m = \frac{(T_0 - t) - (T - t_0)}{\ln \frac{T_0 - t}{T - t_0}} \quad S = \frac{M}{k \Delta t_m}$$

- S = heat-exchange surface, in m^2 (sq.ft.)
 a = coefficient taking into account the diluting molasses added
 TO = massecuite temperature at entry, in $^{\circ}C$ ($^{\circ}F$)
 T = required exit temperature of massecuite, in $^{\circ}C$ ($^{\circ}F$)
 t_0 = entry temperature of cold water, in $^{\circ}C$ ($^{\circ}F$)
 t = exit temperature of water, in $^{\circ}C$ ($^{\circ}F$).

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55. *Centrifugal Separation*

Object

Once the mother liquor has been exhausted to the practical limit for the strike concerned, it remains only to separate the crystals in order to obtain the sugar in the commercial form.

This operation is carried out in centrifugal dryers, which are called variously: "centrifuges", "centrifugals", or commonly abbreviated to "fugals". The operation is known as "purging", "centrifuging", "centrifugalling" or "fugalling".

Until recently, batch machines were the only ones in use to any extent; in recent years, however, continuous machines have been used to an increasing extent, and are now extensively used, at least for low-grade massecuite.

BATCH CENTRIFUGALS

The centrifugal machine

Practically all the batch centrifugals in existence at present are based on the Weston machine, named after the British engineer who originated it in its present form.

It consists of a cylindrical basket designed to receive the massecuite to be treated, and carried on a vertical shaft, which is driven from its upper end by a motor or pulley (Fig. 35.1). The basket is pierced with numerous holes to allow the molasses to escape, and is provided with circumferential hoops to withstand the centrifugal force; the basket is lined with metal gauze, which serves to retain the sugar while allowing the molasses to pass through.

The basket is open at the top to allow massecuite to be fed into it, and a bottom opening allows the sugar to be discharged when the machine is stopped. While the machine is running, that is, during charging and drying, this discharge opening is generally closed by a cone of thin sheet metal.

The basket is attached to the shaft by a spider occupying the bottom opening, while allowing ample space for discharge of the sugar. It is surrounded by a casing designed to catch the molasses separated and to protect the operator from the moving parts. This "monitor casing" has an opening at the top corresponding to that of the basket, which may be closed by means of two half-covers on hinges, provided with a hole through which the shaft passes.

There are generally several centrifugals, forming a battery, and arranged in a line. The massecuite to be fugalled is fed into a distributor, which resembles a small crystalliser or mixer placed behind the heads (or driving motors) of the machines. At each centrifugal, a hinged chute enables massecuite to be run from the mixer to charge the basket.

For the high-grade centrifugals, water pipes and steam pipes run along the length of the battery, to allow of steaming or washing the sugar.

At the end of the drying period the centrifugal is stopped by means of a brake, which

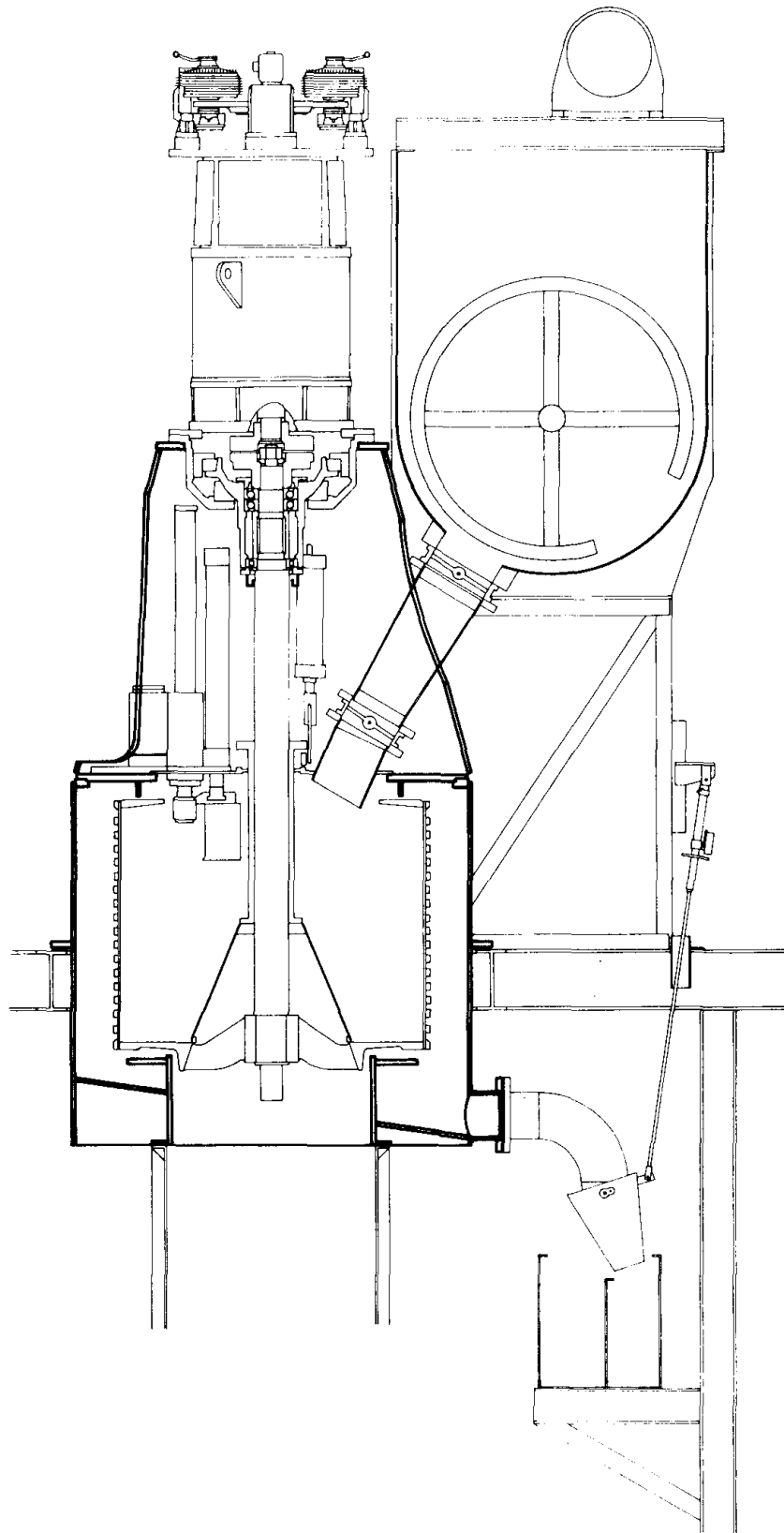


Fig. 35.1. Section of a C411 centrifugal (Fives Cail - Babcock).

generally consists of brake-shoes provided with a suitable friction lining and surrounding a drum on which they tighten when released.

All centrifugals are now automatic. The many operations necessary for their functioning, *i.e.* starting, charging, control of charging speed, closing the massecuite gate, acceleration, washing with water and steam, drying at high speed, change to low speed and control of discharging speed, opening the discharge cone, drying out the sugar, starting the next charge, all are carried out by means of a complex assembly of controls, programmed to operate in the correct sequence.

Basket. The baskets are constructed of greater or lesser thickness according to the centrifugal force to which they are to be subjected. Those which are subjected to the highest stresses work close to the limit of tensile strength of the metal; they are made of chrome-copper steel and their hoops sometimes of nickel steel.

The baskets of centrifugals 60 cm (24 in.) in height have:

For 1,000 r.p.m.: 5-6 mm ($\frac{3}{16}$ - $\frac{1}{4}$ in.) thickness of basket, with 6 hoops

For 1,500 r.p.m.: 5-6 mm thickness of basket, and 12 - 15 hoops

The baskets of centrifugals of 1,370 mm diameter and 1,067 mm height (54 x 42 in.) are 5 mm ($\frac{3}{16}$ in.) in thickness and have 16 hoops of 35 mm ($1\frac{3}{8}$ in.) width; these are for 1,200 r.p.m. (Fives Cail-Babcock, Model C411). The B.M.A. machines of similar dimensions, 1,320 x 1,200 mm (52 x 48 in.), have baskets 13 mm ($\frac{1}{2}$ in.) in thickness, without hoops, but with 3 reinforcing bands of 28 mm (1 in.) one each at top and bottom, and one at mid-height.

The holes in these baskets are of 7 mm ($\frac{9}{32}$ in.) diameter, and are spaced at about 22 mm ($\frac{7}{8}$ in.) between centres.

Gauzes. On account of this wide spacing, a simple perforated sheet or a single woven gauze would be unsatisfactory as lining: the greater proportion of the perforations would lie against the smooth side-wall of the basket, and would not allow the molasses to escape. The basket is then generally furnished with two different linings:

(7) A backing gauze, an ordinary woven gauze in brass or copper wire of 0.75 - 1.5 mm ($\frac{1}{32}$ - $\frac{1}{16}$ in.) diameter with openings of 5-10 mm ($\frac{3}{16}$ - $\frac{3}{8}$ in.), which serves to separate the gauze proper from the wall of the basket: or preferably, a "stabbed" lining consisting of brass or special steel sheet, punched and perforated to give suitable projections; this is more uniform and more effective

(2) A *gauze* proper, designed to retain the crystals; previously, this was sometimes a woven copper gauze, of Liebermann type (Fig. 35.2), single or double, but now is nearly always a perforated sheet.

In certain factories, two backing gauzes are used. Tromp¹, for example, recommends the interposition of a 7-mesh woven gauze between the 4-mesh backing gauze and the perforated lining. Krieg similarly advises the use of a "sandwich" gauze between the lining, of perforated sheet, and the stabbed backing sheet; or use of the following linings²:

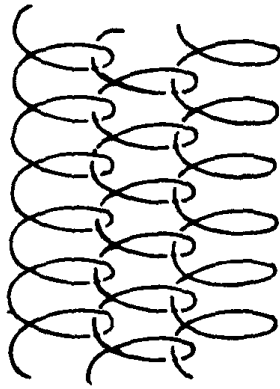


Fig. 35.2. Diagram of Liebermann lining.



Fig. 35.3. Slotted gauze. Perforations in rows.



Fig. 35.4. Slotted gauze. Staggered arrangement.

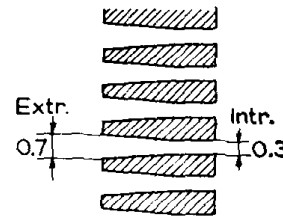


Fig. 35.5. Section of perforations (mm).

(a) A backing gauze, in annealed brass 16 meshes/dm (4 meshes/in.), of wires 1.2 mm ($\frac{3}{64}$ in.) in diameter, placed against the basket.

(b) An intermediate gauze, also in annealed brass, of 28 meshes/dm (7-mesh), 0.75 mm ($\frac{1}{32}$ in.) diameter, between this backing gauze and the perforated lining.

This additional gauze facilitates the flow of molasses, but is not indispensable.

Perforated gauze. Gauzes formed of perforated sheet are of pure copper, stainless steel, or special alloy. They are provided with horizontal slots 4 mm in length and 0.3–0.4 mm in width, generally 0.35 mm, arranged either in columns (Fig. 35.3) or in staggered formation (Fig. 35.4). The pitch of these perforations in the vertical direction is 1 mm (100 slots per 100 mm). The area of openings is of the order of 24–26% of the total area.

The width of 0.3–0.4 mm refers to the interior face of the gauze: the perforations are in fact pyramidal (Fig. 35.5), this arrangement having the object of avoiding the possibility of crystals of the exact dimension of 0.4 mm becoming jammed in the perforations of the gauze.

This width of 0.4 mm is obviously important. On its precise value depends the size of the fine grain which the gauze will allow to pass, or even the size of the grain of the *massecuite*, since in certain countries the final *massecuites* are often boiled with a grain size of 0.2–0.3 mm. In this case it is obviously necessary to modify the width of the perforations, or better, to use close-woven Liebermann linings. Gauzes with round holes of 0.5 mm are also used.

The thickness of the perforated sheet varies from 0.45 to 0.80 mm, often 0.5 mm (stainless steel) and 0.7 mm (brass). The thinner gauzes have a shorter life but give better operation.

The linings should be cut to a length somewhat greater than the interior circumference of the basket, in order to ensure that there is no gap in the lining once it is in place; but the lap arranged should be at a minimum, since it forms a vertical band of greater resistance to flow of molasses. The gauzes should be cut to the lengths given in Table 35.1.

TABLE 35.1

LENGTH OF CENTRIFUGAL SCREENS

<i>Diam. of machine</i>		<i>Inside circumference of basket</i>		<i>Length of perforated sheet</i>	
<i>(in.)</i>	<i>(mm)</i>	<i>(mm)</i>	<i>(ft./in.)</i>	<i>(mm)</i>	<i>(ft./in.)</i>
40	1,015	3,192	10 ft. 5 in.	3,230	10 ft. 7 in.
42	1,065	3,352	11 ft.	3,380	11 ft. 1 in.
48	1,220	3,830	12 ft. 7 in.	3,870	12 ft. 8 in.
54	1,370	4,309	14 ft. 2 in.	4,350	14 ft. 3 in.

The margin of overlap allowed is about 25 – 38 mm (1 - 1 ½ in.), the actual overlap being about 60 mm (2 | in.), due to the thickness of the backing gauze.

The lap of the gauze should be arranged in such a way that, while the basket is rotating, the resistance of the air tends to hold the free end against the other, and not so that the air tends to enter between the two ends.

The principal European suppliers are Krieg & Zivy in France, and Fontaine in Germany.

Summary of some concepts of gyrotory mechanics

The study of centrifugals involves several concepts in mechanics, which we shall recall here, in order to simplify the discussion which follows.

Notation. We shall use the following nomenclature:

- r = radius of gyration, in m (ft.)
- V = tangential velocity, in m/s (ft./s)
- n = speed of rotation, in r.p.m.
- ω = angular velocity, in radians/s
- t = time, in s
- m = mass = p/g , in kg wt./m/s/s (f.p.s. units)
- p = weight, in kg (lb.)
- g = acceleration of gravity = 9.81 m/s/s (32.16 ft./s/s)
- F_r = centrifugal force, in kg (lb.)
- F_t = tangential force, in kg (lb.)
- j_r = radial acceleration, in m/s/s (ft./s/s)
- j_t = tangential acceleration, in m/s/s (ft./s/s)
- C = torque, in kg m (lb.ft.)
- T = work, in kg m (lb.ft.)
- p = power, in kg m/s or W (ft.lb./s or h.p.)

I = moment of inertia, in kg wt. m^2 (lb. wt. sq.ft.) = p/gr^2
 E_k = kinetic energy, in kg m (f.p.s. units).

Equations of definition. The principal ones are:

(1) Weight:

$$P = mg$$

If we take as unit of force the kilogram (pound) weight, then for:

$$p = 1 \text{ kg wt. } \quad m = \frac{1}{9.8} = 0.102 \quad \left(p = 1 \text{ lb. } \quad m = \frac{1}{32.10} = 0.0311 \right) \quad (35.1)$$

0.102 is then the mass of a body weighing 1 kg wt., in the industrial system of units (0.0311 for 1 lb.).

In the system where the basic figure is mass instead of force, one kilogram mass is the mass of a body of weight one kilogram weight.

(2) Angular velocity:

$$\omega = \frac{2\pi n}{60} = 0.105 n \quad (35.2)$$

(3) Radial acceleration:

$$j_r = \frac{v^2}{r} = \omega^2 r \quad (35.3)$$

(4) Centrifugal force:

$$F = mj_r = \frac{mv^2}{r} = m\omega^2 r \quad (35.4)$$

(5) Tangential force:

$$F_t = mj_t = m \frac{dv}{dt} = mr \frac{d\omega}{dt} \quad (35.5)$$

(6) Torque:

$$C = F_t r = mr^2 \frac{d\omega}{dt} \quad (35.6)$$

(7) Power:

$$P = C\omega \tag{35.7}$$

(8) Work:

$$T = \int P dt = \int C\omega At \tag{35.8}$$

(9) Moment of inertia:

$$I = \int mr^2 \tag{35.9}$$

(10) Kinetic energy:

$$F_v = \int mv^2 = \int mr^2\omega^2 = \omega^2 \int mr^2 = I\omega^2 \tag{35.10}$$

Value of the torque. We have

$$\Sigma F_t = \int mr \frac{d\omega}{dt} \tag{35.11}$$

whence:

$$C = \Sigma rF_t = \int mr^2 \frac{d\omega}{dt} = I \frac{d\omega}{dt} \tag{35.12}$$

The torque is often represented by the motive torque C_m decreased by the torque C_f of friction or other resistance:

$$C = C_m - C_f \tag{35.13}$$

Kinetic-energy theorem. We have sometimes to utilise the theorem: half the change in kinetic energy of a system in a given time is equal to the sum of the work done by external forces acting on the system during that time.

Dimensions of centrifugals

Size of centrifugals is characterised by the two principal dimensions:

- (1) The interior diameter of the basket: D
- (2) The interior height of the basket: H .

These two dimensions are standardised; formerly, there were:

5 diameters: 30 36 40 42 48 in.
 and
 4 heights: 18 20 24 30 in.

but the first two diameters and the first height have disappeared, due to labour costs and *the requirements of automation, since automatic control equipment is delicate to install with small machines*; hence only the following dimensions are now used or fabricated:

4 diameters:	1,015	1,065	1,220	1,370	mm
	40	42	48	54	in.
and					
5 heights:	610	760	915	1,065	1,220 mm
	24	30	36	42	48 in.

The (1,015-mm) 40-in. height is now seldom used. These dimensions give the following 10 combinations (Table 35.2):

TABLE 35.2

STANDARD DIMENSIONS OF CENTRIFUGALS: <i>D</i> x <i>H</i>		
<i>British dimensions</i> (in.)	<i>Metric dimensions (mm)</i> (continental makers)	
	<i>Nominal</i>	<i>Actual</i>
40 x 24	1,015 x 610	1,016 x 609.6
40 x 30	1,015 x 760	1,016 x 762
42 x 24	1,065 x 610	1,067 x 609.6
42 x 30	1,065 x 760	1,067 x 762
48 x 24	1,220 x 610	1,219 x 609.6
48 x 30	1,220 x 760	1,219 x 762
48 x 36	1,220 x 915	1,219 x 914.4
48 x 42	1,220 x 1,065	1,219 x 1,067
54 x 42	1,370 x 1,065	1,370 x 1,067
54 x 48	1,370 x 1,220	1,370 x 1,219

The diameter of 1,015 mm (40 in.) is practically a duplication of that of 1,065 mm (42 in.). It corresponds to American standards, whereas 1,065 mm is the European size.

Krieg³ has proposed the following tolerances, which are fairly generally accepted, and should be rigorously observed:

TABLE 35.3

FABRICATION TOLERANCES		
<i>Fabrication</i>		
	<i>"Precision"</i>	<i>"Regular"</i>
Diameter	+0	+0
	-2 mm (0.080 in.)	-4 mm (0.160 in.)
Height	+0	+2 mm (0.080 in.)
	-0.3 mm (0.012 in.)	-0.3 mm (0.012 in.)

The true dimensions, together with the indication "precision" or "regular", should appear on the inner edge of the top opening of the basket. Dimensions marked in this way would avoid confusion when ordering accessories.

Speed of centrifugals

A factor quite as important as the dimensions, from the point of view of centrifugal capacity, is the speed.

Speed and diameter are the factors determining the centrifugal force, *i.e.* the force available for expelling the molasses during fugging.

Now the centrifugal force is proportional (*cf.* eqn. (35.4)) to $\omega^2 r$, *i.e.* proportional to $\omega^2 D$. If then we take as standard speeds for a 1,065-mm (42-in.) centrifugal 1,000 and 1,500 r.p.m., it will be necessary, to realise the same centrifugal force, to run machines of other diameters at the speeds given in Table 35.4.

TABLE 35.4

SPEEDS GIVING EQUAL CENTRIFUGAL FORCES IN CENTRIFUGALS OF DIFFERENT DIAMETERS
(Centrifugal force = *G* times weight)

Diameter		Speed of rotation (r.p.m.)	
(in.)	(mm)	<i>G</i> = 516	<i>G</i> = 1,161
40	1,015	1,025	1,537
42	1,065	1,000	1,500
48	1,220	935	1,403
54	1,370	882	1,323

If we consider a centrifugal of a given speed, and if its speed is varied, we shall obtain a more rapid and more complete drying as the centrifugal force and therefore the speed of rotation is increased.

In other words, if one centrifugal is running at a higher speed than another, both being identical machines and fugging the same massecuite, the machine running at higher speed will complete its drying before the other.

It is found by experience that in such a case the same degree of drying is obtained by running the two machines for times inversely proportional to their centrifugal force, *i.e.* to the square of the speed:

$$\theta n^2 = \text{constant} \tag{35.14}$$

θ = time of drying.

Numerous experiments have confirmed this. In Hawaii, for example⁴, it has been ascertained that 1,015-mm (40-in.) machines working on low-grade massecuite gave the same results, with regard to purity of sugar and of molasses, by running for:

- 68.2 min at 1,150 r.p.m.
- 44.3 min at 1,400 r.p.m.
- 33 min at 1,600 r.p.m.

In all cases, charging and discharging occupy an additional time of about 5 minutes in all, part of which it would be logical to include as operating time. Thus the relationship (35.14) is verified to a close approximation.

We see from this the great gain which is offered by high speeds with regard to centrifugal capacity.

Comment. We have just been comparing the work of the same machine at different speeds. If we had compared centrifugals of different dimensions, it is obvious that the centrifugal force, the only factor influencing drying, will vary also with diameter. We should have then:

$$B Dn^2 = \text{constant} \quad (35.15)$$

whereas the factor D does not appear in comparisons of one machine at different speeds, or of machines of the same diameter when compared with each other.

It follows that the centrifugal effect has the value:

$$\frac{kDn^2}{\theta}$$

The highest speeds attained are of the order of 2,200 r.p.m., in centrifugals of 1,015 mm (40 in.) diameter⁵.

Mean equivalent radius

In a rotating centrifugal, the centrifugal force on the sugar crystals against the gauze is obviously greater than that on the crystals on the interior surface of the sugar layer. It is of interest to ascertain what is the radius for which the centrifugal force is equal to the mean force exerted on the whole mass of sugar. This radius is termed the "mean equivalent radius" and will be designated by R_m (Fig. 35.6).

Let us consider a fugal of axis yy' , of height h , and interior radius R at the gauze. Let r be the corresponding radius at the inner wall of the sugar layer. The thickness of the sugar wall is thus $(R - r)$. The centrifugal basket will be assumed to have a flat bottom, and the small quantity of sugar contained within the top conical plate above the height h will be

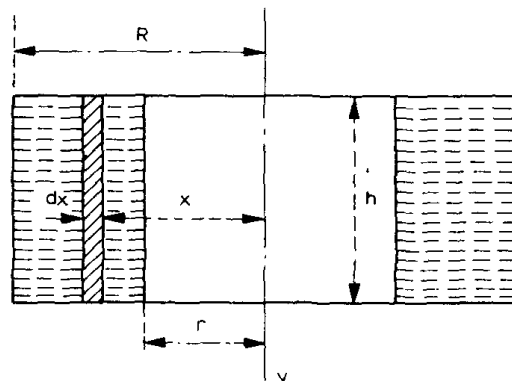


Fig. 35.6. Calculation of mean equivalent radius.

neglected. Considering the cylinder of infinitesimal thickness dx , situated at radius x , the centrifugal force to which this cylinder is subjected will have the value:

$$dF = m \omega^2 x = 2 \pi x dx h \rho \omega^2 x = 2 \pi h \rho \omega^2 x^2 dx$$

ρ being the specific gravity of the massecuite.

The total centrifugal force will be:

$$F = \int_r^R 2 \pi h \rho \omega^2 x^2 dx = 2 \pi h \rho \omega^2 \left[\frac{x^3}{3} \right]_r^R = \frac{2 \pi h \rho \omega^2}{3} (R^3 - r^3)$$

The total mass of sugar has the value:

$$M = (\pi R^2 h - \pi r^2 h) Q = \pi h Q (R^2 - r^2)$$

If this were concentrated at the radius R_m , it would be necessary for the centrifugal force to be equal to F :

$$F = \frac{2 \pi h \rho \omega^2}{3} (R^3 - r^3) = \pi h Q (R^2 - r^2) \omega^2 R_m$$

hence:

$$R_m = \frac{2}{3} \frac{R^3 - r^3}{R^2 - r^2} \tag{35.16}$$

We thus have the figures of Table 35.5.

TABLE 35.5

EQUIVALENT MEAN RADIUS R_m

Diameter of basket		Value of R_m for sugar layer of thickness (mm):				
(mm)	(in.)	152 (6 in.)	178 (7 in.)	203 (8 in.)	229 (9 in.)	254 (10 in.)
1,015	40	436 (17.2 in.)	425 (16.7 in.)			
1,065	42	461 (18.2 in.)	450 (17.7 in.)	440 (17.3 in.)		
1,220	48	537 (21.1 in.)	526 (20.7 in.)	515 (20.3 in.)	504 (19.8 in.)	
1,370	54	613 (24.1 in.)	601 (23.7 in.)	590 (23.2 in.)	579 (22.8 in.)	568 (22.4 in.)

Table 35.4 has been calculated according to the equivalent mean radius, the basis being $D = 1,065$ mm (42 in.), thickness = 152 mm (6 in.), with thicknesses for other sizes proportional to the respective diameters. In Table 35.5 the radii corresponding to the commoner thicknesses are given in italics.

Ratio between centrifugal force and gravity

The effect of various centrifugals is compared by the ratio of the centrifugal force exerted on the material to that of gravity.

Gravity exerts a force equal to the weight: $p = mg$. The centrifugal force itself has the value:

$$F = m\omega^2 R_m$$

The ratio sought, G , thus has the value:

$$G = \frac{F}{p} = \frac{m\omega^2 R_m}{mg} = \frac{4\pi^2 n^2}{60^2 \times 9.8} \cdot R_m = 1,119 \left(\frac{n}{1,000} \right)^2 R_m \quad (35.17)$$

$$\left(28.4 \left(\frac{n}{1,000} \right)^2 R_m \right)$$

R_m = equivalent mean radius, in m (in.).

Example. For a 1,220-mm (48-in.) centrifugal with a 203-mm (8-in.) layer rotating at 1,500 r.p.m., the ratio G will be:

$$G = 1,119 \times 2.25 \times 0.515 = 1,297 \quad (G = 28.4 \times 2.25 \times 20.3 = 1,297)$$

Cycle

It must not be supposed that running at operating speed is the only important operation in the course of the fugalling. The other phases of the process occupy a substantial fraction of the cycle of operations, which is greater in high-grade masseccutes than with low-grade products, and is naturally greater with more rapid cycles.

The following, for example, is the succession of operations for a complete cycle, in the treatment of high-grade masseccuite:

Charging and acceleration:	1 min	
Drying and washing at speed:	0	40 s
Electrical braking:	0	15
Mechanical braking:	0	10
Discharging:	0	20
Complete cycle:	2 min	25 s

Operation at normal speed, of course, occupies a much greater proportion of the cycle for low-grade products, fugalling of which is much slower and more difficult.

We may estimate, according to the speed of the machines, the times given in Table 35.6.

TABLE 35.6

TIMES OF FUGALLING

A massecuites	2 – 6 min
B massecuites	4 – 10 min
C massecuites	10 – 45 min

The factors which influence the time of fugalling are:

- (a) The viscosity of the molasses, which depends on its temperature and purity (whether A, B or C massecuite)
- (b) The size and regularity of the crystals
- (c) The rapidity of acceleration of the machine, that is, the time required to reach its operating speed
- (d) The centrifugal force developed by the centrifugal at its operating speed
- (e) The time of braking and of discharging.

Moreover, washing with water and steam may contribute to lengthening the cycle.

The inactive time which is most unfavourable to capacity is phase (e), *i.e.* the period of deceleration. While the acceleration takes about the same time, it still constitutes a useful phase of the operation, since, from the beginning of filling and accelerating, the amount of molasses removed is quite substantial. The deceleration period, on the contrary, is dead time without useful effect. The low capacity of certain centrifugals is thus due simply to their ineffective braking.

Capacity

The working capacity or the sugar production of a centrifugal machine depends on two main factors:

- (a) Contents of the basket in volume of massecuite
- (b) Time of the cycle.

(a) **Capacity of the basket.** The capacity of the basket may be expressed either in volume of massecuite or in weight of sugar. The quantity of sugar to be obtained from a massecuite varies considerably (*cf.* p. 728), and it is preferable to express the contents in volume of massecuite, when it is desired to compare one machine with another, while using weight of sugar when considering the sugar production from the machine.

The shape of the basket is now very simple, the bottom and the upper rim of the basket being flat. The volume of massecuite thus depends on:

- (1) The area of the lining of the centrifugal
- (2) The thickness of the layer of massecuite.

Centrifugals are designed to receive a layer of massecuite or of sugar proportional to their diameter. The manufacturers generally provide for a massecuite layer of maximum thickness equal to 14 or 15% of the diameter. Since a margin of safety in filling is always allowed, it is necessary to estimate for calculation purposes a maximum of 14% for calculations of power and load on machine, and a mean of 10–12% for calculation of capacity:

$$e_M = 0.28 R = 0.14 D \quad (35.18)$$

$$e_m = 0.20 R = 0.10 D \quad (35.19)$$

e_M = maximum thickness of massecuite

e_m = mean thickness of massecuite

R = inside radius of basket

D = inside diameter of basket.

However, this expression is applicable without qualification only to high-grade massecuites which fugal readily. With low grades, it is generally the properties of the massecuite which determine the thickness of layer which can be handled. The latter may sometimes drop to half the thickness corresponding to the flange of the top of the basket, or even lower.

The volume of massecuite per charge will have the value:

$$V = \pi e H (D - e) \quad (\text{flat top and bottom}) \quad (35.20)$$

This is readily obtained by means of Guldin's theorem.

V = volume of massecuite corresponding to the thickness e of massecuite, and expressed in dm^3

e = thickness of the massecuite, in dm

H = interior height of the centrifugal basket, in dm

D = inside diameter of the basket, in dm

The same equation applies in any consistent system of units; for example, with all linear measurements in feet.

If we give e values of $0.12 D$ and $0.14 D$, we obtain the figures of Table 35.7.

TABLE 35.7

VOLUME OF MASSECUIE CONTENT FOR VARIOUS CENTRIFUGALS

$D \times H$			Maximum volume of massecuite			
			<i>Theoretical</i> ($e = 0.14 D$)		<i>Practical</i> ($e = 0.12 D$)	
(mm)	(in.)	(in.)	(l)	(cu.ft.)	(l)	(cu.ft.)
1,015 x 610	40 x 24	• 40 x 24	246	8.68	215	7.59
1,015 x 760	40 x 30	40 x 30	305	10.77	267	9.43
1,065 x 610	42 x 24	42 x 24	272	9.60	237	8.38
1,065 x 760	42 x 30	42 x 30	337	11.90	295	10.41
1,220 x 610	48 x 24	48 x 24	356	12.57	311	10.98
1,220 x 760	48 x 30	48 x 30	442	15.60	386	13.63
1,220 x 915	48 x 36	48 x 36	528	18.64	491	16.27
1,220 x 1,065	48 x 42	48 x 42	618	21.82	539	18.33
1,370 x 1,065	54 x 42	54 x 42	778	27.46	681	24.04

It may readily be verified that the two columns of this table correspond very closely to the equations:

$$V_t = 390 D^2H \quad (V_t = 0.000225 D^2H) \quad (35.21)$$

$$V = 340 D^2H \quad (V_p = 0.000196 D^2H) \quad (35.22)$$

V_t = maximum theoretical contents of basket, in dm³ (cu.ft.) of massecuite
 V = maximum practical contents of the basket, in dm³ (cu.ft.) of massecuite
 D = diameter of basket, in m (*in.*)
 H = height of basket, in m (*in.*).

(b) Length of cycle. We have already considered (p. 768) the factors influencing the time of cycle. Those which depend on the characteristics of the machine are:

- (1) The centrifugal force developed when running at speed
- (2) The rate of acceleration
- (3) The rate of slowing down and discharging.

The first factor, centrifugal force (and consequently the running speed), is the most important, particularly for low-grade materials. Equation (35.15) shows that the higher this force, the shorter the cycle. The centrifugal force is the product of two factors: D and n , the latter to the second power; both of these work in opposition to the two other factors, (2) and (3). When D and n are increased, the acceleration time cannot at the same time be reduced except at the expense of power consumption. Only the discharging time is practically independent of dimensions and speed: it is substantially constant and takes about half a minute.

Considering flat-bottomed machines, eqn. (35.22) gives us:

$$Y = N V_p = 0.340 D^2HN \quad (0.000196 D^2HN) \quad (35.23)$$

Y = volume of massecuite handled per hour, in 1 (cu.ft.)
 D = diameter of basket, in m (*in.*)
 H = height of basket, in m (*in.*)
 N = number of cycles per hour.

Capacities for different sizes are given in Table 35.8.

To convert these capacities in volume of massecuite to capacity in weight of sugar, it will be necessary to multiply them by the figures for yield (R_A , R_B or R_C) on p. 728 or preferably by the actual yields obtained in the factory under consideration.

Screen area

In some countries, centrifugal capacity of a factory is often expressed in terms of screen area of the machines. Obviously this gives only a rough measure of capacity, since it takes into account neither the actual contents of the basket nor the speed, nor the number of cycles per hour. It amounts to substituting for eqn. (35.23) an equation of the form $C = k DH$.

With older machines, a figure of 0.2 m²/t.c.h. (2.2 sq.ft./t.c.h.) was considered necessary for A and B massecuites and 0.3 (3.3) for C massecuite. With modern machines, it should be ample to provide:

TABLE 35.8

CAPACITY OF CENTRIFUGALS IN LITRES OF MASSECUIE PER HOUR

<i>D</i> x <i>H</i> (in.)	<i>D</i> x <i>H</i> (mm)	<i>D</i> ² <i>H</i> (l)	0.340 <i>D</i> ² <i>H</i> (l)	Number of cycles per hour											
				2	4	6	8	10	12	15	18	20	22	25	30
40 x 20	1,015 x 510	524	178	357	713	1,070	1,426	1,783	2,140	2,675	3,209	3,566	3,923	4,457	5,349
40 x 24	1,015 x 610	629	214	428	856	1,284	1,712	2,140	2,567	3,209	3,851	4,279	4,707	5,349	6,418
40 x 30	1,015 x 760	787	267	535	1,070	1,605	2,140	2,674	3,209	4,012	4,814	5,349	5,884	6,686	8,023
42 x 20	1,065 x 510	578	197	393	786	1,179	1,573	1,966	2,359	2,948	3,538	3,931	4,324	4,914	5,897
42 x 24	1,065 x 610	694	236	472	944	1,415	1,887	2,359	2,831	3,538	4,246	4,718	5,189	5,897	7,076
42 x 30	1,065 x 760	867	295	590	1,179	1,769	2,359	2,948	3,538	4,423	5,307	5,897	6,487	7,371	8,845
48 x 30	1,220 x 760	1,133	385	770	1,540	2,311	3,081	3,851	4,621	5,777	6,932	7,702	8,472	9,628	11,553
48 x 36	1,220 x 915	1,359	462	924	1,849	2,773	3,697	4,621	5,546	6,932	8,318	9,243	10,167	11,553	13,864
48 x 42	1,220 x 1,065	1,585	539	1,078	2,156	3,234	4,312	5,389	6,467	8,084	9,701	10,779	11,857	13,474	16,168
54 x 42	1,370 x 1,065	2,007	682	1,365	2,729	4,094	5,459	6,824	8,188	10,235	12,283	13,647	15,012	17,059	20,471

For British units: 1 l = 0.0353 cu.ft.

0.10 m²/t.c.h. (1.1 sq.ft./t.c.h.) for A and B sugars
 0.10 mVt.c.h. (1.1 sq.ft./t.c.h.) for C sugar

or a total of 0.20 mVt.c.h. (2.2 sq.ft./t.c.h.).

But, to provide a safe margin, the areas are increased to:

0.12 mVt.c.h. (1.3 sq.ft./t.c.h.) for A and B sugars
 0.13 mVt.c.h. (1.4 sq.ft./t.c.h.) for C sugars

or a total of 0.25 mVt.c.h. (2.7 sq.ft./t.c.h.).

We would emphasise that these figures have always only an indicative and approximate value.

Screen areas for various sizes of machine are given in Table 35.9.

TABLE 35.9

SCREEN AREA FOR VARIOUS SIZES OF CENTRIFUGAL

<i>Size</i>		<i>Area</i>	
		<i>(sq.ft.)</i>	<i>(m²)</i>
<i>(mm)</i>	<i>(in.)</i>		
1,015 x 610	40 x 24	20.9	1.95
1,015 x 760	40 x 30	26.2	2.43
1,065 x 610	42 X 24	22	2.04
1,065 x 760	42 x 30	27.5	2.55
1,220 x 760	48 X 30	31.4	2.92
1,220 x 915	48 x 36	37.7	3.50
1,220 x 1,065	48 x 42	44.5	4.10
1,370 x 1,065	54 x 42	49.5	4.60

Moment of inertia

The moment of inertia, the definition of which we have given in eqn. (35.9), plays an important role in the calculation of power consumed by centrifugals.

The moment of inertia depends on the mass of the rotating bodies and their distance from the axis of rotation. It is different therefore according as the centrifugal is empty or full of massecuite, and again changes progressively as the molasses is eliminated during the fugalling. We may thus distinguish three principal values of the moment of inertia:

- (1) Moment of inertia of the empty centrifugal
- (2) Moment of inertia of the centrifugal filled with massecuite immediately after charging
- (3) Moment of inertia of the centrifugal filled with sugar, at the end of the fugalling.

Moreover, the mass of the rotating material is not the same for all machines of the same dimensions. For example, a centrifugal designed to resist a centrifugal force of 500 x g would not be designed in the same way as a basket intended to withstand a force of 1,200 x g: special steels of higher strength would be employed in the latter case, the thickness of metal would be increased, the number of hoops increased, etc.

It follows that the moment of inertia depends, not only on the dimensions of the centrifugal, but also on the speed for which it has been designed, and the extent to which it is filled.

Table 35.10 gives moments of inertia (kg wt.m² and lb.wt.sq.ft.) for various centrifugals.

TABLE 35.10

MOMENT OF INERTIA OF CENTRIFUGALS

Dimensions				Speed (r.p.m.)	Moment of inertia					
					kg wt.m ²			lb.wt. sq.ft.		
(mm)		(in.)			Empty	Charged	Final	Empty	Charged	Final
1,015	x	610	40 x 24	1,500	9	17	15.5	65	123	112
1,015	x	760	40 x 30	1,500	10	20	18	72	145	130
1,065	x	610	42 x 24	1,500	11	20	16	80	145	116
1,065	x	760	42 x 30	1,500	12.5	23	20	90	166	145
1,220	x	760	48 x 30	1,450	24	38.8	34.7	173	280	251
1,220	x	915	48 x 36	1,300	25	44	37	180	318	268
1,220	x	1,065	48 x 42	1,200	26.3	50.5	41.5	190	365	300
1,370	x	1,065	54 x 42	1,200	46	78.6	71.4	332	568	517

(For values in kg m², multiply the above figures by 9.8; for values in lb.sq.ft., multiply the values in the table by 32.16.)

Variation of moment of inertia. The moment of inertia of a centrifugal varies according as it is empty, charged with massecuite or full of sugar. It is of interest to know how these moments of inertia vary with the characteristics of the centrifugal. For this purpose we shall discuss what are the limitations which its dimensions and its speed impose, and which are reflected in the design and the inertia of the machine.

Stress in the basket. The most important element concerned in the moment of inertia of the empty machine is the basket. Now, the thickness of the shell of the basket, or rather the strength of the combination of shell and hoops, must be sufficient to withstand the centrifugal stresses to which it is subjected.

Let M be the total mass of the basket and the massecuite at the full charge for which it should be designed. The force to which the wall of the basket is subjected has a value of:

$$F_r = M\omega^2 r \quad (35.24)$$

This force is distributed over the whole surface of the wall. Hence the stress per unit area:

$$P = \frac{M\omega^2 r}{\pi DH} \quad (35.25)$$

From the point of view which concerns us (the effect of variation in the weight of the

basket), it reduces to assuming, for simplification, that the basket consists of a shell of uniform thickness, without perforations and without hoops. In this case, if f is the maximum stress permissible for the metal employed, we shall have (Fig. 35.7):

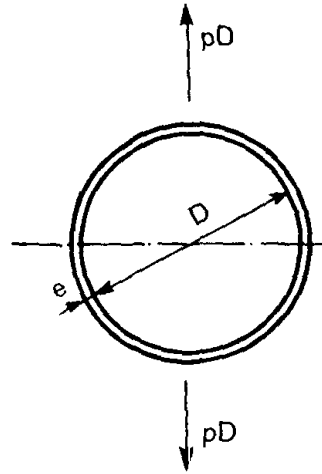


Fig. 35.7. Stresses in basket.

$$e = \frac{pD}{2f} \tag{35.26}$$

e = thickness of metal.

Or:

$$e = \frac{M\omega^2 r D}{2\pi D H f} = \frac{M\omega^2 r}{2\pi H f} \tag{35.27}$$

Now M is proportional to $D^2 H$ (cf. eqns. 35.21, 35.22), as far as the principal load, *i.e.* the *massecuite*, is concerned. Using k and k_1 for the numerical coefficients, we shall have then:

$$e = k \frac{D^2 H n^2 D}{H f} = k_1 D^3 n^2 \tag{35.28}$$

It is seen that the thickness of the basket should be proportional to $D^3 n^2$.

The mass m of the basket will therefore be proportional to the volume of the metal comprising it:

$$m = k_2 \pi D e H = k_3 \pi D D^3 n^2 H = k_4 D^4 H n^2$$

and its moment of inertia I_p will be:

$$I_p = m D^2 = k_4 D^6 H n^2 \tag{35.29}$$

Actually, the moment of inertia of the empty centrifugal varies much less rapidly as a function of each of the three factors D , H and n . This is due to the fact that the motor, the shaft and the parts other than the basket do not have to withstand the centrifugal force exerted by the massecuite, also to the fact that, as the centrifugal stresses increase, the designers employ steel of higher strength; these are more expensive but make the basket lighter. Finally, the moment of inertia of the empty centrifugal increases only according to the formula:

$$I_p = 7nD^5 \sqrt{H} \quad (I_p = 0.074 nD^5 \sqrt{H}) \quad (35.30)$$

Moment of inertia of the massecuite or sugar. The weight of massecuite fugalged is proportional to D^2H (cf. eqns. 35.21, 35.22). Its moment of inertia I_s will thus be proportional to:

$$I_s = k D^2 H D^2 = k D^4 H$$

and will have the value:

$$I_s = 9D^4 H \quad (I_s = 0.17 D^4 H) \quad (35.31)$$

Total moment of inertia while drying. The total moment of inertia during the drying period will thus have the form:

$$I = I_p + I_s = 7 D^4 H \left(\frac{nD}{\sqrt{H}} + 1.3 \right) \quad (35.32)$$

These values assume D and H expressed in m (ft.), / in m^2 kg wt. (square feet-pounds weight), and n in thousands of r.p.m.

Power requirements

There are two types of power to be considered with centrifugal machines:

- (a) The power for starting, or power required during the period of acceleration: P_a
- (b) Power while running: P_r .

The latter is obviously much smaller than the former, since it corresponds only to maintaining the speed, whereas the power for starting corresponds to the expenditure of energy necessary to bring the centrifugal from rest to its operating speed, thus giving it a considerable kinetic energy.

- (a) **Acceleration period.** The theorem of kinetic energy gives us:

$$T_a = \frac{I\omega^2}{2} = Q \int_0^t P_a dt \quad (35.33)$$

T_a = work corresponding to the acceleration period

Q = efficiency

t = duration of the acceleration period.

Or, assuming the power is constant and equal to the mean power during the acceleration period:

$$\frac{I\omega^2}{2} = \rho P_a t \quad (35.34)$$

Or:

$$P_a = k \frac{In^2}{t} \quad (35.35)$$

k = numerical coefficient.

Replacing I by its value taken from eqn. (35.32), corresponding to its mean value for the acceleration period, we have:

$$P_a = \frac{k}{t} D^4 H n^2 \left(\frac{nD}{\sqrt{H}} + 1.3 \right) \quad (35.36)$$

In practice, we may take:

$$P_a = 0.3 \frac{D^4 H n^2}{t} \left(\frac{nD}{\sqrt{H}} + 1.3 \right) \quad (35.37)$$

P_a = power consumed by the centrifugal during the acceleration period, in kW

D = inside diameter of basket, in m.

H = interior depth of basket, in m.

n = speed of rotation at full speed, in thousands of r.p.m.

t = duration of acceleration period, in min.

(b) Period at speed. When running at speed, we may assume that the torque due to frictional resistances has the value:

$$C = D^4 H n (1 + 4n) \quad (35.38)$$

C = torque due to frictional resistance, in m kg.

D = diameter of basket, in m.

H = depth of basket, in m.

n = speed of rotation, in thousands of r.p.m.

Hence the power consumption while at speed:

$$P_s = \frac{C\omega}{75} = D^4 H n (1 + 4n) \frac{105 n}{75}$$

or:

$$P = \frac{D^4 H n^2}{370} (1 + 4n) \quad (35.39)$$

P_r = power consumed by the centrifugal at speed, in kW.

Power and capacity

From eqn. (35.15), we may deduce:

$$N = K n^2 D \quad (35.40)$$

N = number of cycles per hour.

This equation, however, is valid only for the running period. The accelerating time, on the other hand, would be detrimentally affected by n and by D ; in more precise terms, eqn. (35.37) shows that, merely to retain the same accelerating time t with a centrifugal of larger diameter, it is necessary to expend a power consumption P more than proportional to D^4 ; we thus see the enormous influence of diameter. Braking time is similarly affected. Discharging time remains approximately the same in all cases. Since with low-grade products the running period predominates, we may consider eqn. (35.40) as a close approximation for this case. If we relate it to eqn. (35.23), the latter becomes:

$$Y = k D^3 H n^2 \quad (35.41)$$

The power exerted per unit weight of massecuite treated or per unit rate of sugar produced is thus:

(a) Accelerating:

$$\frac{P_a}{Y} = k' \frac{D^3 H n^2}{\sqrt{H}} (1 + 4n) = k' \frac{D}{t} \left(\frac{nD}{t} + 1.3 \right) \quad (35.42)$$

(b) Running:

$$\frac{P_r}{Y} = k'' \frac{D^4 H n^2}{D^3 H n^2} (1 + 4n) = k'' D (1 + 4n) \quad (35.43)$$

It is seen that the power consumption per unit weight of sugar increases with diameter of machine, and, to a smaller extent, with the speed.

This expression for the relationship of specific power as increasing with D and to a lesser extent with n , is, however, more marked with A and B sugars.

On the other hand, the height H scarcely affects the situation: capacity and power are both approximately proportional to H . Hence the advantage of choosing a centrifugal giving the greatest depth of basket for a given diameter.

The diameter, however, offers an advantage, in addition to its influence on capacity. Since alternating current at 50 cycles generally fixes the speed at 1,500 r.p.m., only the diameter can in effect be varied to increase the centrifugal force. Hence the advantage of large diameters, at least for low-grade material, provided of course that the speed of 1,500 r.p.m. is used. For A and B sugars, the centrifugal force developed at 1,200 or 1,300 r.p.m. is ample, even at 1,065 mm (42 in.) diameter.

Typical values of power consumption at speed. Application of eqn. (35.39) gives the figures shown in Table 35.11:

TABLE 35.11

POWER CONSUMPTION OF CENTRIFUGALS AT SPEED

Dimensions				Power, in kW (h.p.), required at:		
(mm)		(in.)		1,000 r.p.m.	1,500 r.p.m.	2,000 r.p.m.
1,015 x	610	40 x	24	3.4 (4.5)	10.7 (14.2)	24.5 (32.6)
1,015 x	760	40 x	30	4.3 (5.7)	13.4 (17.9)	30.7 (40.9)
1,065 x	610	42 X	24	4.1 (5.5)	13.0 (17.3)	—
1,065 x	760	42 x	30	5.2 (6.9)	16.4 (21.8)	—
1,220 x	760	48 X	30	8.9 (11.8)	27.8 (37.1)	—
1,220 X	915	48 X	36	10.6 (14.1)	33.4 (44.5)	—
1,220 x	1,065	48 X	42	12.4 (16.5)	39.0 (52)	—
1,370 x	1,065	54 X	42	19.8 (26.4)	62.5 (83.3)	—

ELECTRIC DRIVE

There were previously three classical methods of drive for centrifugals: hydraulic, belt and electric motor. Only electric drive has survived. Thus we consider only electrically driven centrifugals.

Coupling

These machines are driven by a vertical electrical motor, the shaft of which is in line with that of the centrifugal, and which drives it either by a direct coupling, by means of a friction clutch, or a hydraulic coupling. The friction clutch, the brake lining of which wore rapidly, is no longer employed. The hydraulic coupling consisted of a driving wheel fixed to the motor shaft, and driving, through the medium of oil, a turbine wheel fixed to the centrifugal shaft. This assured a smooth drive. This method of drive was very successful throughout 1955 under the name of "fluid drive", but it involved a high power consumption, of the order of 13 kWh per tonne of sugar, or more than double the normal consumption; for this reason it has fallen from favour. Direct drive is now practically universal.

Motors for centrifugal drive

There are two principal methods of electric drive for centrifugals:

(1) By 3-phase induction motor: this may have a wound rotor, but more often is of squirrel-cage type. The latter type is distinguished by its simplicity and robustness.

(2) By direct-current motors supplied through thyristors from the a.c. network of the factory. This solution offers several advantages:

(a) Choice of speed of the centrifugal independent of the frequency of the a.c. supply. Speed is no longer tied to 1,500 r.p.m.

(b) Precise control of speed, hence of the drying time.

(c) Specific power consumption one-third to two-thirds of that of an induction motor: 3–4 kWh/tonne of sugar in the masecuite, as against 8–10 kWh/tonne.

(d) Elimination of the peak current demands for each change of polarity of an induction motor. The power demand of the d.c. motor is constant and controllable during acceleration and braking.

(e) Less heating of the motor. The higher power consumption of the induction motor causes heating of the rotor before changing of polarity and braking. The efficiency of the d.c. motor is much higher, leaving about one-third of the heat to be dissipated, thus rendering the motor much smaller.

(f) Operation without sudden peak loads, due to elimination of the abrupt acceleration with an induction motor. Mechanical parts are thus less critical.

(g) Flexibility of operation, due to control of the rates of acceleration and braking, thus permitting adaptation of the fugalling to the quality of masecuite handled. This regulation is not possible with induction motors.

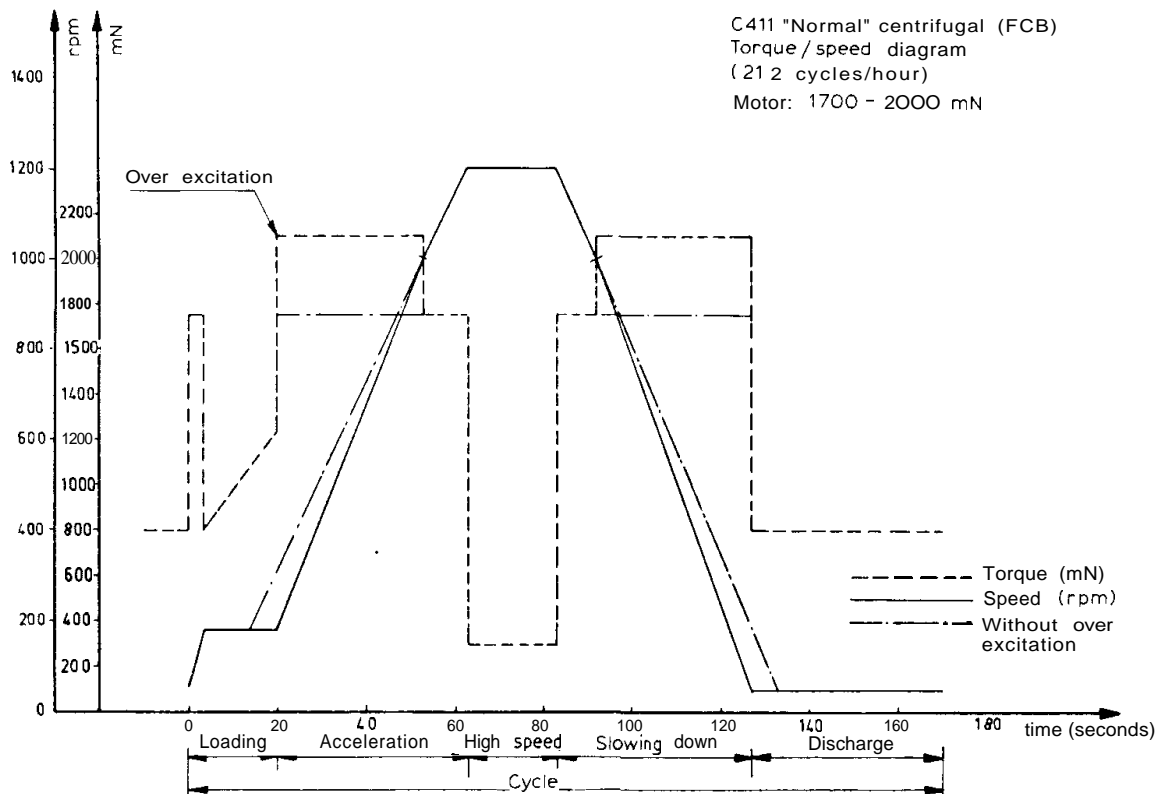


Fig. 35.8. Diagram couple/speed of a C411 centrifugal.

Regenerative braking

One of the interesting characteristics of electric centrifugals is that, thanks to regenerative braking, they permit of a partial recovery of the energy expended. With alternating current this is obtained by means of a motor of double polarity, which can work with half the normal number of poles. Such a motor will run for example at 1,500 r.p.m. (4 poles) or 750 r.p.m. (8 poles), and if it is switched abruptly from 4 to 8 poles, it will return energy to the line until its speed has fallen to 750 r.p.m.

In practice, this regenerative braking is effective only down to about 60% of the normal speed. However, this corresponds to more than half the stored energy, since:

$$1^2 - 0.6^2 = 0.64$$

It will be seen that the energy thus recovered represents only a small fraction of the total energy used. It is no less interesting on that account.

With d.c. motors, a much greater proportion of the stored energy may be recovered. Indeed, with present-day motors, supplied by thyristors, as the supply is unidirectional, regenerative braking is obtained by reversing the direction of the excitation current. The motor thus works as a generator.

Operating conditions of centrifugal motors

The efficiency of electric motors driving centrifugals is low, since they operate alternately during acceleration and during running at speed, and there are no motors in existence which give a good efficiency in conditions varying so greatly.

The same disadvantages are encountered again with regard to power factor. The power factor increases to 0.80 or 0.90 for a brief period during the acceleration, but will fall to 0.40 or 0.50 while running at speed. At that part of the cycle, the power of the motor is considerably greater than that required to maintain the speed (on the subject of the power factor, *cf.* p. 1074).

It is of advantage, therefore, not to seek too rapid an acceleration. A compromise should be sought between a motor too powerful, which will accelerate rapidly and which will have a low efficiency and a low power factor while running, and a motor of too low a power, which will lose much time in acceleration.

For high-grade massecuite, it is of advantage to have somewhat more powerful motors, since they are nearly all the time engaged in accelerating. For low-grade products, on the other hand, it will be of advantage to take care of the power factor on account of the long period of running at speed.

On the other hand, what limits the power of an electric motor is the heat produced. If it is capable of supplying 50 kW when running continuously, it would, for example, be able to give an output of 60 kW if it were stopped for $\frac{1}{2}$ minute in every 3 or 4. Conversely, a motor which delivers a peak power at short intervals will be obliged to run for the rest of the time at a lower power output than the mean continuous power of which it is capable, to avoid exceeding its maximum permissible temperature.

Now centrifugal motors necessarily work in a very intermittent manner: they furnish a high peak power during acceleration, run for some time at speed, then have an idle period cor-

responding to the braking and discharging. The motor to be adopted will depend then on the duration of the cycle, and on the conditions of acceleration which are desired. We shall see below "Choice of electric motor" (at foot of this page) how the power should be determined as a function of these diverse requirements.

Power-factor correction

In an electrified factory, it is the centrifugal station which generally plays the greatest part in depressing the power factor, on account of the abnormal conditions of operation of its motors.

At Puunene, in Hawaii⁶, measurements were made of the power factor of a battery of 15 centrifugals of 1,065 x 610 mm (42 x 24 in.), using 3-phase current at 60 cycles, 440 V, with 2-speed motors (1,200/600 r.p.m.) taking 22.5 kW (30 h.p.) during acceleration and 3.25 kW (5 h.p.) during running. For the running period a value of $\cos \Phi = 0.442$ was found. After connecting static 3-phase condensers of 10 kVA to the motors, the power factor was increased to 0.876 for the same period. The total current taken by the motor, previously 23.15 A, was thus reduced to 11.7 A, or a saving of 50%.

Speeds for charging and ploughing out

The centrifugal is fed with masecuite at a speed between 100 and 350 r.p.m., preferably in the neighbourhood of 240 r.p.m.; discharging by plough at a speed between 30 and 60 r.p.m., preferably about 35 r.p.m.

The firm of Broadbent supplies electric motors capable of operating at five different speeds, three of these being fixed and two adjustable, for feeding and ploughing out⁷.

Special characteristics of electric drive

Although they are almost universal at the present day, electrically driven centrifugals do possess some disadvantages. They demand of their motors contradictory operating conditions, since these conditions cannot be optimal at the same time for accelerating and for operation at speed. The motor is thus the result of a compromise, the most obvious effect of which is to reduce the power factor.

Electrical centrifugals are rather expensive, since they demand as many motors as there are machines. Each machine, however, is independent, forming a separate unit by itself; hence ease of locating each machine. If trouble is encountered with one motor, only one machine is shut down; hence minimum interruption to production.

The individual motor has enabled the realisation of machines of highest performance, highest capacities, and best quality of sugar produced. Upkeep expense is less: breakdown, stoppages and replacements of motors are rare. Complete automation has made them economical in labour.

These advantages have led to their quasi-monopoly. Their real competitor is still an electric machine: the continuous centrifugal, which is destined to be further improved and to gradually replace the batch centrifugal.

Choice of electric motor

Three factors are involved in choice of motor.

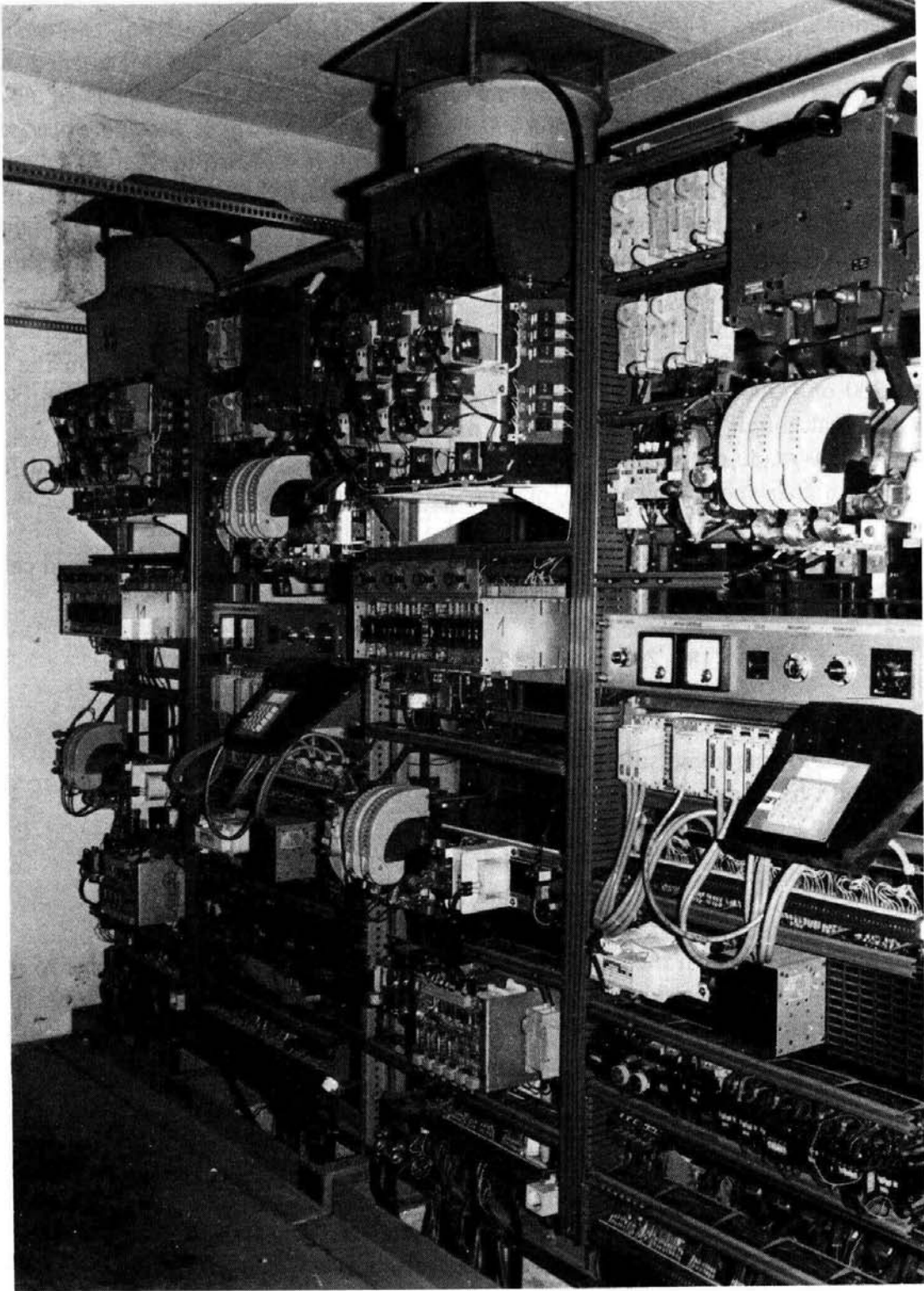


Fig. 35.9. Control panel of a group of automatic centrifugals

(1) Speed. The supply available being generally a.c. limits the number of possible speeds for induction motors to two. For normal European supply, corresponding to 3-phase 50 cycles, these are 1,000 and 1,500 r.p.m. The speed of 1,000 r.p.m. is now exceeded, which no longer leaves any choice for induction motors. For American standards of 60-cycle 3-phase supply, the possible speeds are 1,200 and 1,800 r.p.m., both of which are acceptable, the former for larger diameters, and the latter for machines of 1,015 mm (40 in.) diameter; in the latter case an exceptional centrifugal force and separating power is available.

If intermediate speeds are required, it is necessary to resort to a d.c. motor, which allows European manufacturers to depart from the "magic number" of 1,500 r.p.m. and to adopt in all cases the speed considered optimal.

With European standards, we may take:

(a) For high-grade sugars, 1,500 r.p.m. for 1,065-mm (42-in.) machines. On the other hand, 1,200 or 1,300 r.p.m. may be sufficient for 1,220-mm (48-in.) or 1,370-mm (54-in.) centrifugals, although 1,500 r.p.m. may in many cases be preferable.

(b) With low-grade **massecuite**, a speed of 1,500 r.p.m. is strongly recommended for all sizes, and a higher speed if possible for 1,065-mm (42-in.) machines.

(2) Cycle. The operating cycle is more or less fixed, on the one hand by the speed chosen, on the other hand by the **massecuite** to be treated. Generally, we may reckon:

18 – 25 cycles per hour for A and B sugars
2 – 4 cycles per hour with low grades

these figures allowing some safety margin.

Motors should be designed for the maximum number of operations, say:

25 - 30 per hour for A and B
6 per hour for low grade

(3) **Acceleration time.** In the case of high-grade **massecuites**, it is of advantage to reduce the acceleration time to a minimum, in order to increase the useful operating time, and so improve the capacity of the machine. There is, however, obviously a limit below which the acceleration time should not be reduced, imposed either by heating of the motor or by the maximum peak load of which the motor is capable.

The maximum current which a motor can stand without damage is generally of the order of three times the mean current corresponding to the nominal continuous load for the motor. With low-grade **massecuite** the position is much easier, since it is preferable to accelerate more slowly: too abrupt an acceleration provokes a settling of the crystals which can sometimes obstruct the interstitial passages for the molasses, when the crystals are not uniform.

Recent installations and present trends

Until the end of the Second World War, all operations of batch centrifugals were still manually performed. Since that time, enormous progress has been made in automation. A

first stage consisted of relieving the operator of the laborious operations, particularly discharging, which was realised by means of "ploughs" (Fig. 35.10). The equipment necessary for the various elementary operations of the cycle was furnished with servomotors, and the operator then had only to give the signal to initiate these operations at the appropriate moment, generally by simple push-buttons. Next, all these movements have been put under the control of an installation of contactors, time switches or time delay relays, which control and carry out all the necessary operations of the cycle. The fugal operator has become redundant and a simple supervisor is sufficient; present-day fugals are now entirely automatic.

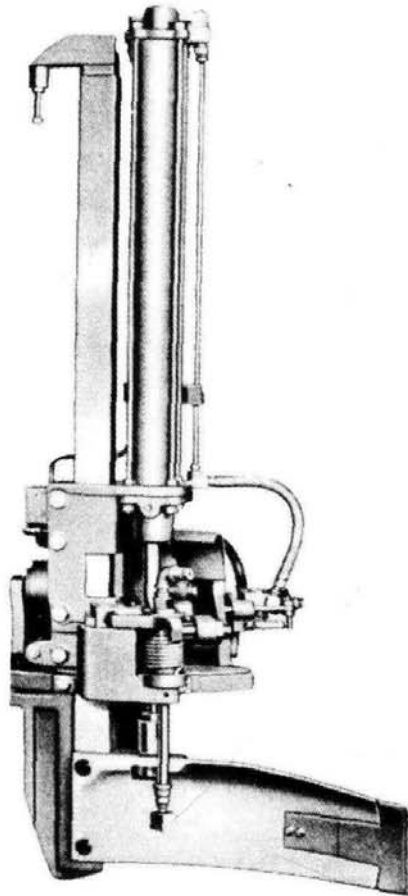


Fig. 35.10. Automatic plough (Western States).

The actuating medium is often compressed air, actuated by electrically operated valves. Pneumatic operation can thus be applied to the massecuite feed valve, to wash water, to separation of run-off molasses, to mechanical braking, to the closing device at the bottom of the basket, to the cover of the monitor casing. There are also discharge ploughs actuated by compressed air or electric motor.

With the aid of knurled knobs and graduated dials, the various periods of the cycle can be set and regulated: starting, acceleration, running at speed, braking, discharging. Charging is regulated and the massecuite valve closed with the aid of a feeler which is set according to the thickness of the massecuite layer required.

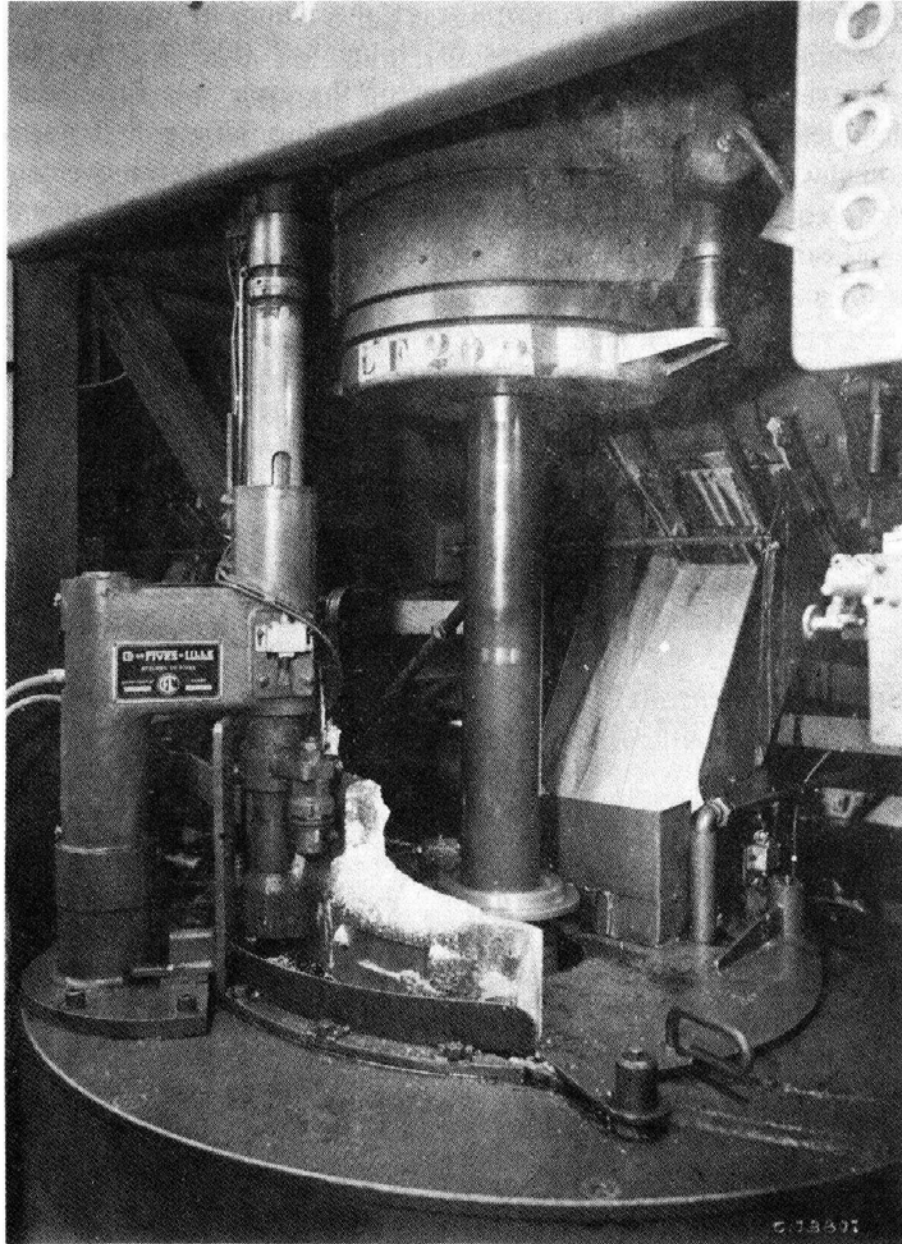


Fig. 35.11. Battery of 3 "Compact 220" centrifugals (48 x 30 in.) Induction-motor drive (Fives Cail – Babcock).

In the field of French machines, the most notable at present is the "Compact" of Fives Cail – Babcock (Fig. 35.11). These machines require no overhead steelwork, as the motor is supported by a cone mounted on the monitor casing of the centrifugal; hence economy in installation costs and great stability. The cone is easily and rapidly dismantled.

Fives Cail – Babcock offer the "Compact" machine in three models, the standard C411 and two optional models for factories treating less than 220 t.c.h. Figures for these are given in Table 35.12, which also includes some figures for B.M.A. machines, also widely employed.

Fives Cail – Babcock reckon the capacity of a C411, with massecuite purity of 88, as 24 t/h of massecuite (16 m³/h), with sugar output of 410 t/24 h.

TABLE 35.12

"COMPACT" CENTRIFUGALS (FIVES CAIL - BABCOCK)

Model	D x H (mm)	D x H (in.)	Massecuite thickness		Massecuite capacity (max.)		Speed (r.p.m.)	Cycles (per h)	Sugar (t/24h)
			(mm)	(in.)	(kg)	(lb.)			
C411	1,370 x 1,065	54 x 42	195	8	1,300	2,865	1,200	24	410
C221	1,220 x 760	48 x 30	175	7	700	1,540	1,450	25	
C331	1,220 x 1,065	58 x 42	175	7	1,000	2,200	1,300	24	

B.M.A. CENTRIFUGALS

Model	D x H (mm)	D x H (in.)	Massecuite thickness		Massecuite capacity (max.)		Speed (r.p.m.)
			(mm)	(in.)	(kg)	(lb.)	
Standard	1,320 x 800	52 X 31½	180	7	800	1,750	
	1,320 x 1,000	52 x 39	180	7	1,000	2,200	
Variant	1,320 x 1,000	52 x 39			1,000	2,200	1,500
	1,320 x 1,000	52 x 39			1,250	2,750	1,500
	1,320 x 1,220	52 x 48			1,550	3,400	1,500

The basket is of stainless steel, the hoops of alloy steel of very high tensile strength. The fugals are entirely automatic, the control equipment, with time switches and indicating instruments, being housed in cabinets, with a small panel at the machine for manual control for test purposes, and another panel where in case of trouble a signal light serves to indicate the location of the fault.

The machines of one battery are coordinated by a device assuring a regular sequence of commencing the cycles of different machines, in order to avoid simultaneous starting or stopping and to achieve a steady power consumption.

The plough is actuated by compressed air, a piston moving it vertically and swinging it into position. Its blade remains outside the casing during charging and spinning. Discharge takes place at a speed of 40 r.p.m.

Figure 35.8 (p. 780) shows a graph of speed and torque for a "Compact C411" on A massecuite.

As an example of recent installations from other countries, we take one of the best known, that of Roberts (Western States) (Fig. 35.12). This organisation manufactures the machines shown in Table 35.13.

The 1,220 X 760 mm (48 X 30 in.) machines running at 1,450 r.p.m. and the 1,370 x 1,015 mm (54 x 40 in.) machines (1,160 r.p.m.) are generally designed for low-grade massecuite.

All the centrifugals listed in Table 35.13 can be supplied as fully automatic machines. The motors are two-speed, of squirrel-cage type. Water washing is applied under a pressure of 3.5 kg/cm² (50 p.s.i.), by a pipe with 6, 7 or 9 nozzles, each delivering 9 l/min (2 gal./min). The wash period may be adjusted between 3 and 12 seconds. Discharge is effected by rotating the basket in the opposite direction by an independent 2.3-kW (3-h.p.) motor; this has permitted the designer to leave the plough inside the basket, its blade facing forward during the

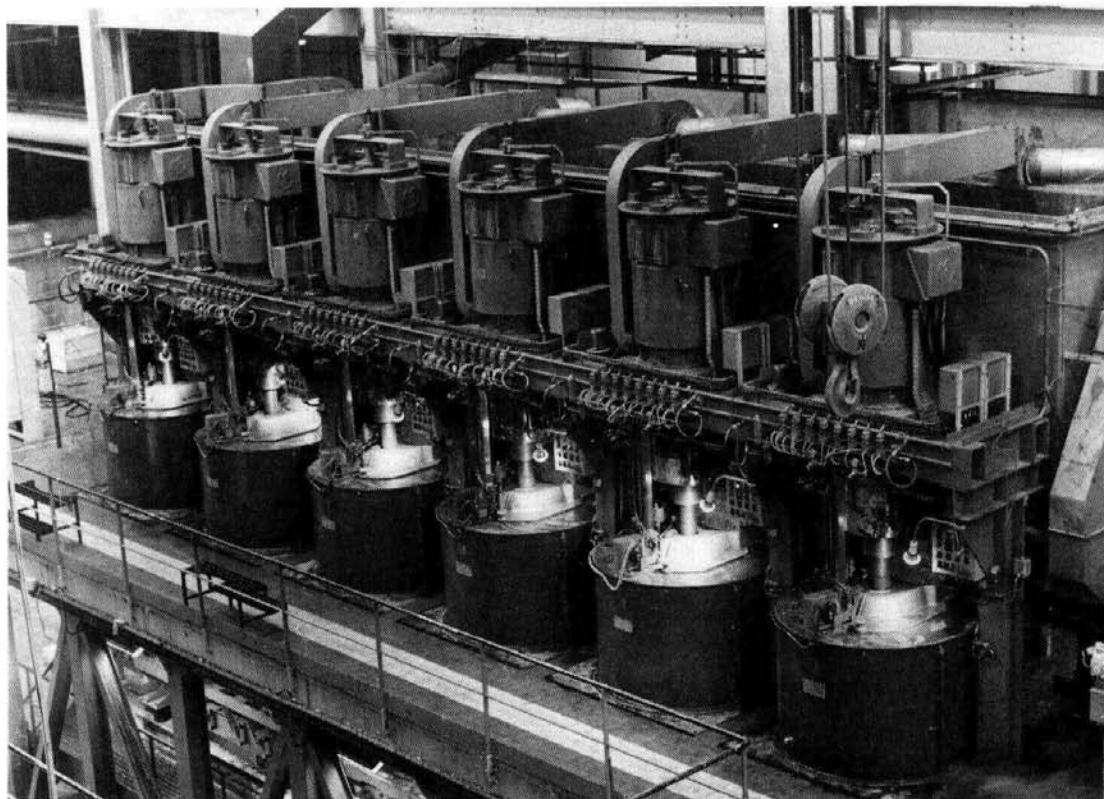


Fig. 35.12 Battery of 6 Roberts G & D centrifugals. 54 x 40 in.: d.c. and thyristor static converter (Western States).

TABLE 35.13

WESTERN STATES G8 CENTRIFUGALS

$D \times H$ (mm)	$D \times H$ (in.)	Massecuite thickness		Massecuite capacity		Speed (r.p.m.)	Factor G	Cycles		Massecuite capacity	
		(mm)	(in.)	(dm^3)	(cu.ft.)			(s)	(per h)	(l/h)	(cu.ft./h)
1,015 x 760	40 x 30	150	6	315	11.2	1,150	645	202	18	5,650	200
				315	11.2	1,450	1,026	170	21	6,700	237
1,220 X 760	48 X 30	175	7	445	15.7	960	542	210	17	7,620	270
				445	15.7	1,160	792	180	20	8,890	314
				445	15.7	1,450	1,238	165	22	9,700	342
1,220 x 915	48 x 36	175	7	530	18.8	960	542	215	17	8,890	314
				530	18.8	1,160	792	185	19	10,400	368
1,370 x 1,015	54 x 40	175	7	670	23.9	860	497	230	16	10,400	365
				670	23.9	960	620	210	17	11,300	400
				670	23.9	1,050	741	190	19	12,600	443
				670	23.9	1,160	905	180	20	13,200	466

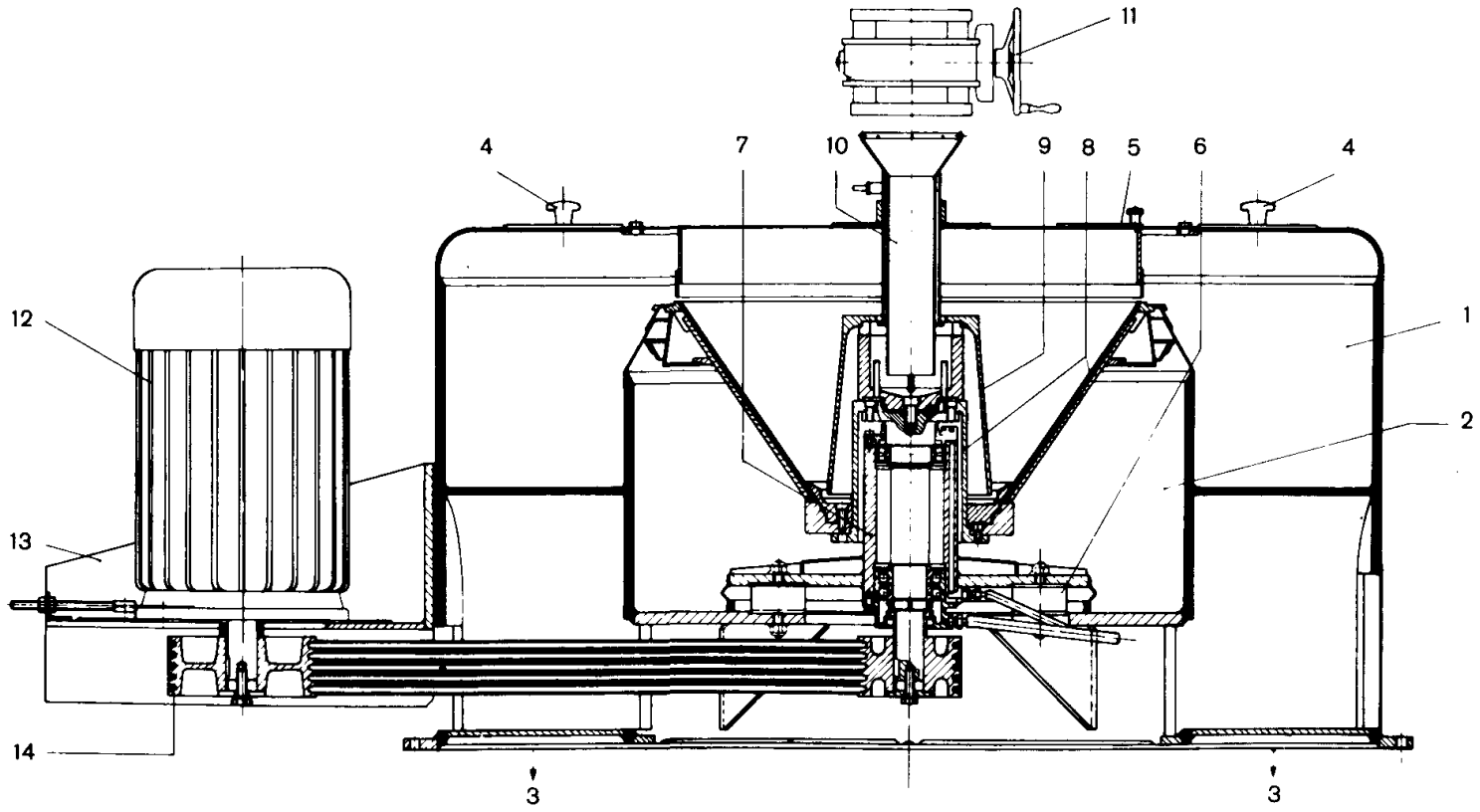


Fig. 35.13. Continuous centrifugal K 850 (M.A.).

ploughing, but backward relative to the operating rotation. The end portion of the blade is flexible. The discharging speed of the basket is 35 r.p.m.

The electrical installation provides numerous safety devices, *e.g.* the motors are equipped with thermal overload relays; they cannot be switched to high speed without having passed through low speed; the small reverse motor for ploughing cannot be switched on unless the main motor has been switched off; ploughing cannot be carried out unless washing has taken place; the plough and the mechanism for lifting the discharge cone are interconnected.

OPERATION OF CENTRIFUGALS

Washing

From the centrifuging point of view, three fractions of the molasses must be distinguished:

(a) The excess molasses, which is the proportion in excess of what is necessary to just fill the voids between the crystals. This excess renders the massecuite fluid and is removed very readily.

(b) The proportion just necessary to fill the voids between crystals. This has to be removed by flowing between the crystals which are then touching each other. It is more difficult to remove than the excess fluid.

(c) The molasses film surrounding the crystals. This is much the most difficult to remove, on account of its adhesion.

(A) Water washing. Due to this film, it is seldom possible to produce a suitable commercial sugar without some washing. This consists of applying water in a fine spray on the surface of the sugar, when the first two fractions of molasses have been removed. The water washes the molasses film and removes it; at the same time it unavoidably dissolves a small amount of sugar, and the washing period thus gives a molasses of higher purity and lower brix, called "rich syrup" or "light molasses". This is sometimes separated from the first portions of molasses separated, which are known as "poor" or "heavy" molasses, if the quantity of it is worth the extra trouble. However, to avoid detrimental effects on the purity drop, the amount of washing must be kept to a minimum; with automatic centrifugals, washing for example will occupy from 2 to 8 s; 2 - 3 s often being sufficient to give the required improvement in colour and quality.

It has been found in Mauritius⁸ that the polarisation of the sugar was better when the washing was effected later in the course of the drying period (30 - 40 s after high speed has been reached).

Chapman⁹ estimates that the best moment for applying wash is when 75% of the molasses has been eliminated. With high-speed centrifugals operating on a short cycle, the washing, which lasts from 3 to 10 s, is effected about 50 s after charging or about 20 s before the end of the period of running at high speed.

Beetz¹⁰, in Germany, has given a formula relating the quantity of water used for washing to the corresponding rise in purity of the sugar:

$$P' = 100 - \frac{100 - P}{1 + W} \quad (35.44)$$

Tromp¹² quotes a maximum of 8% of sugar for wash water.

The values used for modern centrifugals are generally much lower than these figures, often less than half. They often vary between 2 and 5% on weight of sugar.

Double purging. Double purging is at present employed in the manufacture of white sugar. With raw sugar, it is rarely used for high-grade massecuites, but is often used for C sugars.

It consists of first purging the massecuite in a battery of centrifugals without any washing. The molasses obtained is thus a "poor" or "heavy" molasses. The sugar is discharged into a mixer located below these first centrifugals, where it is remixed to a massecuite by mingling with higher-purity molasses or with syrup, clarified juice, or water. From there it is sent to the mixer-distributor of a second battery of centrifugals, called "finishing" centrifugals, where it is purged with water and steam washing. The molasses so obtained is a "rich" or "light" molasses.

This procedure gives a beautiful sugar and an excellent separation of molasses, but requires two batteries of machines instead of one.

Double purging of C sugar. With C massecuite, double purging improves the exhaustion, gives considerable safeguard against irregularities of work in the final massecuite, and assists a better elimination of starch¹³, hence a better filterability of A and B sugars.

The C sugar obtained in the first set of fugals is remixed as a magma in the mixer below the machines, generally by mingling with B molasses. Fresh B molasses is used, taken from the storage tanks, diluted to 70° brix and reheated to 70°C (158°F).

After the second purging, the resulting C sugar is again mixed to a magma with syrup. About 30% of syrup will be used for 70% of C sugar; or in terms of dry substance, 20% in the syrup and 80% in the sugar. The rich syrup obtained by the second purging in the finishing fugals will be returned to the C massecuite and not to the B strikes; it will be pumped to the B storage tank or preferably to a special tank alongside, without recirculation¹⁴.

Relative capacities of first and finishing fugals. The battery of machines for first purging is practically the same as for single purging.

The finishing machines, having a much lighter and more rapid duty, can be substantially smaller in number. Davies¹⁵ gives them a capacity $2\frac{1}{2}$ times higher than that of the first purging machines, or, for conditions in Jamaica, 1.7 and 4.2 m³/h (60 and 150 cu.ft./h), respectively, or 5 machines for first purging for every 2 finishing machines, if they are similar machines. In the same tests, he has obtained with double purging a sugar of 95.2 polarisation, compared with 88.8 for single purging. When the fugalling of C sugar is slow and laborious, the relative capacity of the finishing machines is still more marked and their number may be as low as one-third of that of the first purging machines.

Purity of low-grade sugar

It is estimated in Hawaii¹⁶ that the purity of the sugar from the low-grade massecuite should be maintained in the region of 70 — 75 apparent. If the purity falls below this figure, the quality of the high-grade sugars would feel the effects of it, in darker colour and in filterability

in the refinery; and there would be an increase in the volume of low-grade massecuite to be handled at pans, crystallisers and centrifugals (it is necessary to allow for 30% more low-grade massecuite for sugar at a purity of 65 relative to sugar at a purity of 75).

These Hawaiian values are very low; when raw sugar of a high polarisation is manufactured, by the single- or double-magma system, it is necessary to maintain a purity of 80–85 for the C sugar, 85 if possible. With double purging, this may go as high as 88–92.

Conditions for good performance. Good performance at the centrifugals depends on two essential properties of the massecuite to be treated:

- (a) Uniformity of grain size
- (b) Viscosity of the mother liquor.

When the grain is not uniform, fugalling can become very difficult. A definite symptom is given by the appearance of a film of molasses on the inner face of the wall of sugar, which persists sometimes until the end of the fugalling.

In this case, and if the type of machine permits, a slower and more gradual acceleration of the machine should be tried: a centrifugal with very rapid acceleration provokes the formation against the gauze of a tightly-packed layer of sugar which seriously retards the passage of the molasses, since the fine grain lodges in the interstices between the larger grains. The best period of acceleration, in this case, is such as to accelerate from zero to 1,500 r.p.m. in approximately $2\frac{1}{2}$ –3 minutes. A slow machine, or rather one with a long acceleration period, will give in this case a better-quality sugar and a better elimination of molasses; at the same time it offers the advantage of a much lower peak power consumption.

Such slow acceleration is recommended for C massecuites; but the proper solution consists above all of arranging to produce massecuites with a grain size as uniform as possible.

Ventilation and desiccation

Chapman¹⁷ recalls that various tests have demonstrated the unfavourable effect of evaporation and cooling on the fugalling of low-grade sugars: the proportion of residual molasses may be found to increase by more than half (*e.g.* 23% on crystal, compared with 15%). He recommends accordingly that covers should be provided always for low-grade fugals and that they should be kept closed while running to minimise air circulation and cooling of the massecuite, and that a very small quantity of steam should be introduced inside the basket, to maintain a high humidity with the same object.

Cleaning

It is advisable to clean the centrifugal gauzes with steam, at least once per day, in order to remove crystals which lodge in the openings. The manufacturers supply a small pipe especially for this purpose.

Damaged gauzes

Sometimes a centrifugal gauze becomes torn or split. This accident is recognised by the presence of crystals in the molasses. Frequent checks should be made for such trouble: the crystals are readily detected by feeling between the fingers. The torn gauze must be repaired or replaced immediately.

Such escape of crystals may equally be due to an excessive clearance (greater than 0.8 mm) between the gauze and the top of the basket, or to defects in the basket or defects in the gauze.

The firm of Krieg and Zivy supplies "retaining rings", or "anti-leak", or an "anti-leak band" designed to remedy this fault and to render the centrifugal tight against leakage of crystals. One is sometimes surprised at the decrease in purity of molasses which the use of such anti-leak devices realises.

CONTINUOUS CENTRIFUGALS

With batch machines there are many disadvantages: the complexity of the operations of each cycle, as expressed in a most impressive array of electrical equipment for automatic machines; the low efficiency in the use of energy in these machines with their accelerating and braking phases, and the impossibility of devising an electric motor well adapted to all phases of the cycle; the dead times; and particularly the disadvantages inherent in discontinuous operation of any process. All these motives have long led technologists to seek a method of continuous operation. However, while such a method appeared basically so much simpler than one involving the imposing trappings of automatic centrifugals, a long and laborious task has been involved to bring it into practical use; and it still leaves something to be desired in certain aspects. Since 1960 it has, however, entered a phase of rapid expansion and has been adopted by most newly built factories.

Continuous operation is obviously ideal from the mechanical point of view: the machine runs at a constant speed without stopping, is fed by a continuous stream of material, furnishes a constant output of sugar, and requires no operator and only intermittent supervision. It is the one method for the future.

Equipment

There are many different continuous machines. They may be divided into two principal types:

- (1) Pusher type, which were soon abandoned
- (2) Conical machines, which in turn may be divided into:
 - Those with axis horizontal (similarly abandoned)
 - Those with axis vertical.

Centrifugals with axis vertical. This is the type which has predominated and is now the only one made. It is made by **Hein-Lehmann, B.M.A.**, Western States, Silver, Fives **Cail-Babcock** and **Smith-Mirrlees** (Figs. 35.13-35.17).

The perforated basket has a cone angle of 25 - 34° and rotates at 1,200-2,600 r.p.m. Its diameter is 80 - 125 cm (32 - 50 in.). It uses a perforated screen with openings varying from 0.09 mm for B masseuite to 0.06 mm for C masseuite; perforations may be as low as 0.04 mm. The output of masseuite is of the order of:

- 4 – 8 t/h for C massecuite «
- 7 – 13 t/h for double purging of C
- 7 – 15 t/h for B massecuite
- 20 – 30 t/h for A massecuite

The power required is **30 – 90 kW**.

The massecuite, heated to about **50°C (122°F)**, falls as a thread into the middle of the cone, the apex of which is replaced by a small plain bowl (Hein - **Lehmann**, Silver) or a perforated bowl (**B.M.A.**). Washing is effected by a pipe running along a generatrix of the cone. The heavy and light molasses are recovered separately. The sugar escapes over the rim of the cone (Figs. 35.13 and 35.14).

The Western States Company indicates that, for C massecuite, a common optimal quantity of wash water is 15 - 25 l/m³ (**0.10 - 0.15 gal./cu.ft.**) of massecuite. In Mauritius, an optimal figure for **B.M.A.** machines is given as 35 l/m³ (**0.22 gal./cu.ft.**), under 2 kg/cm² (30 p.s.i.) pressure. Conrad specifies that the wash should be applied at the lowest point of the basket and should extend upwards for **150 – 200 mm (6 – 8 in.)** at most, this length depending on the viscosity of the massecuite, and that the spray should be normal to the basket, the nozzles being about 50 mm (2 in.) from the screen. One nozzle may well be sufficient, with an orifice of 1.2 mm; an orifice of 2 mm is too large. If the point of application of the wash is brought further up the cone, the purity of molasses will be increased. As for purity of the sugar, a figure of **85 – 88** should be **obtained¹⁸**.

The thickness of the layer of sugar is very small: **2 – 5 mm**. At the discharge, it is hardly more than the dimension of the biggest crystals.

Feed of massecuite is controlled by an automatic regulator according to the power absorbed by the motor. With **B.M.A.** machines, this regulator adjusts an iris diaphragm placed in the massecuite feed pipe.

The Western States machines are provided with a motor of 30 kW (40 h.p.) and a large multiple V-belt of rubber.

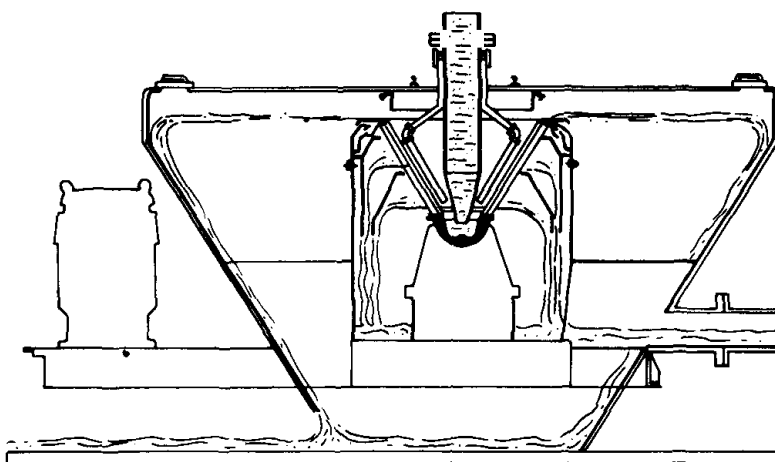


Fig. 35.14. Silver continuous centrifugal for commercial sugar.

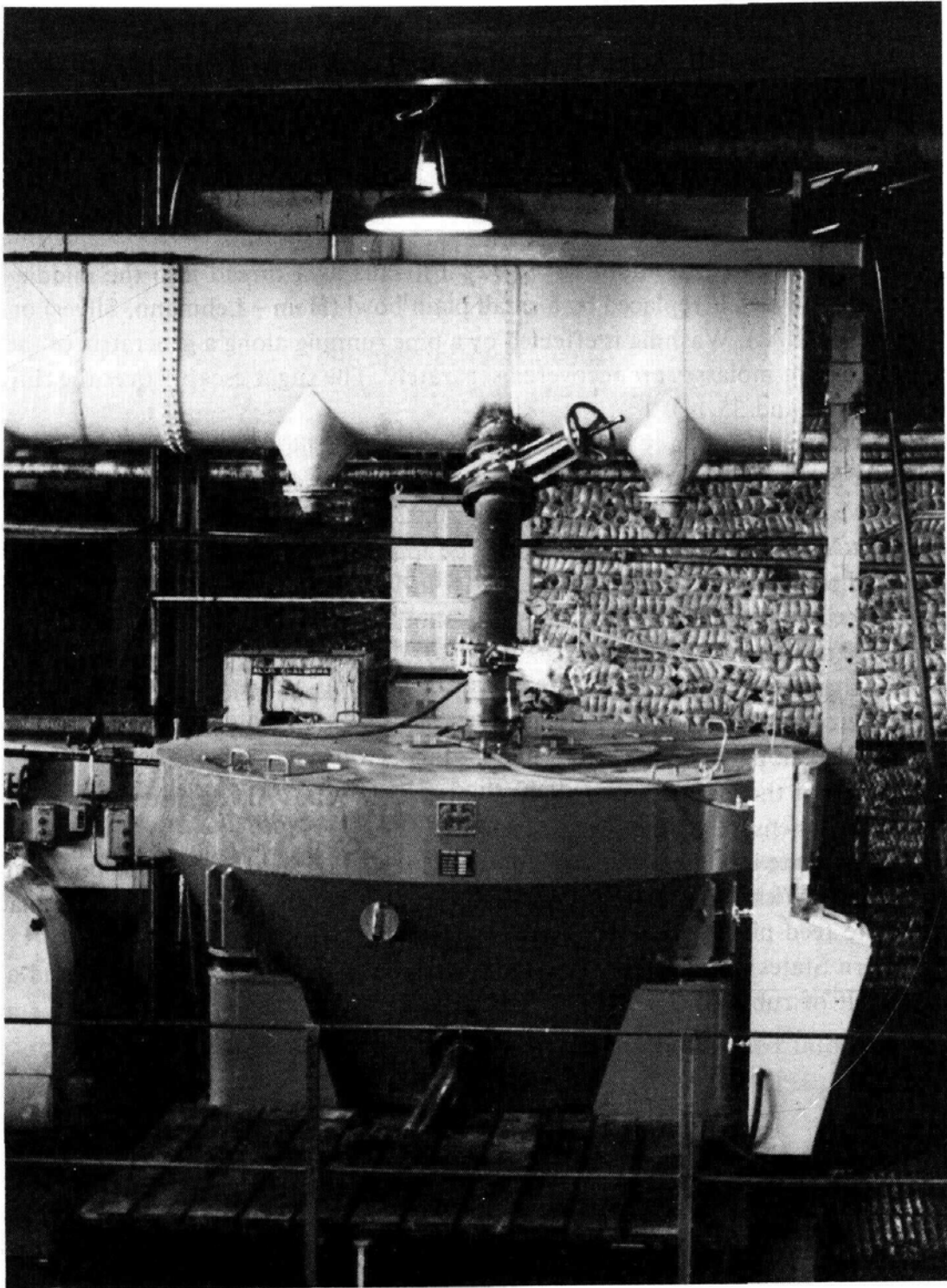


Fig. 35.15. Continuous centrifugal with large diameter casing FC1000 GC3.

Grain breakage

One of the main grounds of complaint against continuous centrifugals is that they break the crystals. This damage is due to two causes:

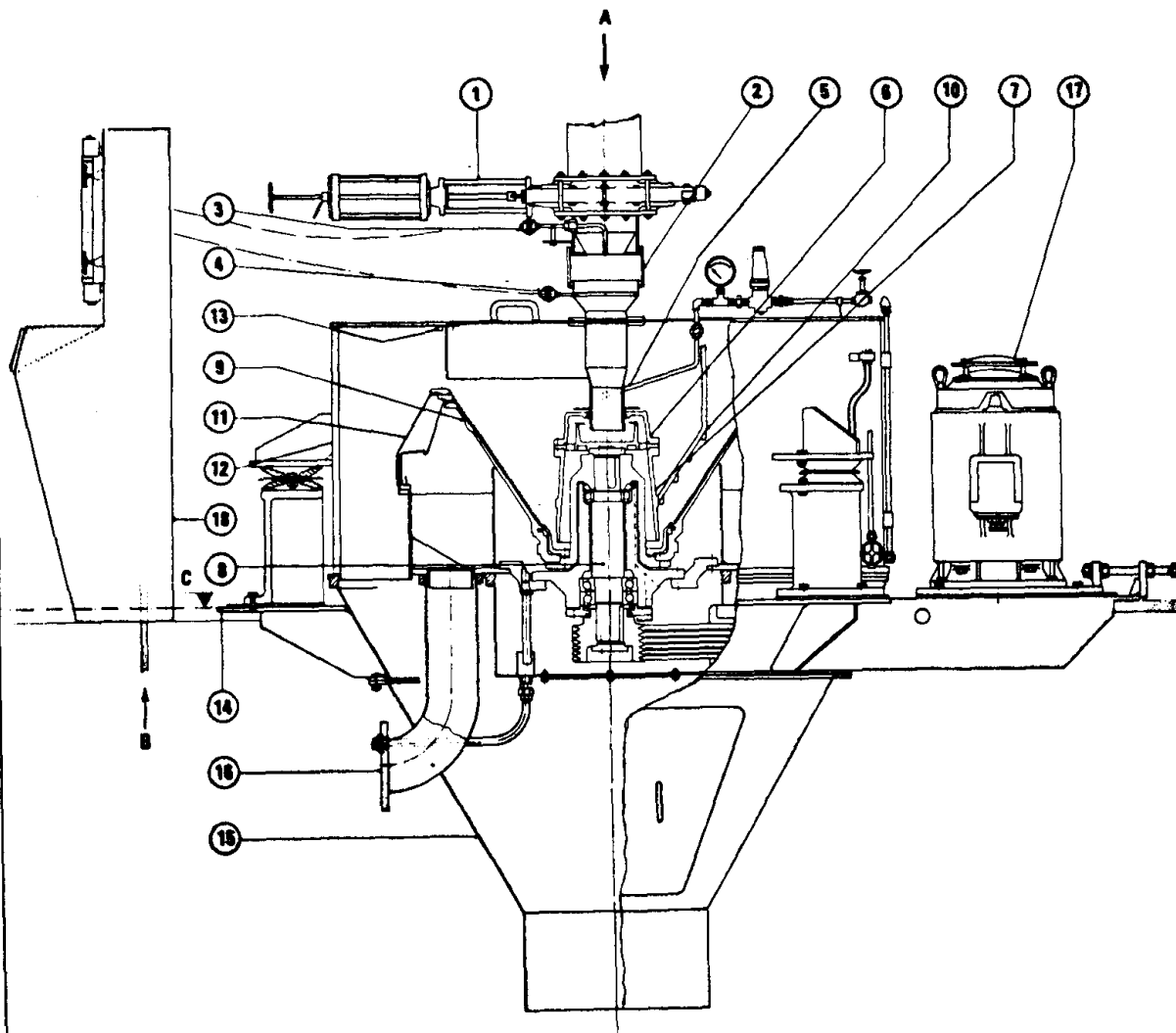
- (a) The crystal is eroded during its travel up the cone.
- (b) It is broken by the violent shock when it is thrown out of the cone. To avoid this latter

- A Arrivée de masse-cuite
- B Arrivée d'eau
- C Plancher de service
- 1/ Vanne pneumatique de réglage de débit
- 2/ Alimentation centrale avec regard
- 3/ Clairçage coaxial central
- V Clairçage coaxial annulaire
- 5/ Réchauffage vapeur de masse-cuite
- S/ Boidistributeur
- 7/ Pot d acceleration
- 8/ Pivoterie
- 9/ Panier équipé
- 10/ Clairçage radial
- 11/ Chicane
- 12/ Cuve suspendue
- 13/ Tube de nettoyage de cuve
- 14/ Plaque de base
- 15/ Trémie d'évacuation du sucre
- 16/ Sortie des égouts
- 17/ Moteur
- 18/ Pupitre de commande

- A Masecuite intake
- B Water inlet pipe
- C Service platform
- 1/ Pneumatic flow regulating valve
- 2/ Central feed with inspection opening
- 3/ Coaxial central washing system
- 4/ Coaxial ring type washing system
- 5/ Masecuite steam heating
- 6/ Feed bowl
- 7/ Acceleration pot
- 8/ Pivot system
- 9/ Equipped basket
- 10/ Radial washing system
- 11/ Baffle
- 12/ Suspended chamber
- 13/ Casing cleaning tube
- 14/ Base-plate
- 15/ Sugar discharge hopper
- 16/ Runoff discharge pipe
- 17/ Motor
- 18/ Control deak

- A Alimentación de masa cocida
- B Entrada de agua
- C Plataforma de servicio
- 1/ Válvula neumática de regulación de flujo
- 2/ Alimentación central con mirilla
- 3/ Lavado coaxial central
- 4/ Lavado coaxial anular
- 5/ Recalentamiento por vapor de la masa cocida
- 6/ Cubeta de distribución
- 7/ Cubeta de aceleración
- 8/ Pivoteria
- 9/ Canasta equipada
- 10/ Lavado radial
- 11/ Tabique
- 12/ Cámara suspendida
- 13/ Tubo de limpieza de la cuba
- 14/ Placa de base
- 15/ Tolva de descarga del azúcar
- 16/ Salida de la miel
- 17/ Motor
- 18/ Pupitre de maniobra y control

$\alpha = 34^\circ$



3

Fig. 35.16. Continuous centrifugal FC1000 GC3.

disadvantage, certain manufacturers (B.M.A., Western States, Silver, FCB) provide machines with an outer casing of very large diameter, so that the crystals have a long travel before striking the casing, and are slowed down by air resistance. These large-diameter machines are particularly offered for commercial A and B sugars (Fig. 35.14).

Fives Cail – Babcock supply, mainly for B massecuite, the FC1000 GC3 (Figs. 35.15 and 35.16) the casing of which is 3 m (10 ft.) in diameter, and the basket angle 30°. They also supply the FC1000 GCV4, with a diameter of 4 or even 5 m (13 – 16.5 ft.), and a basket angle

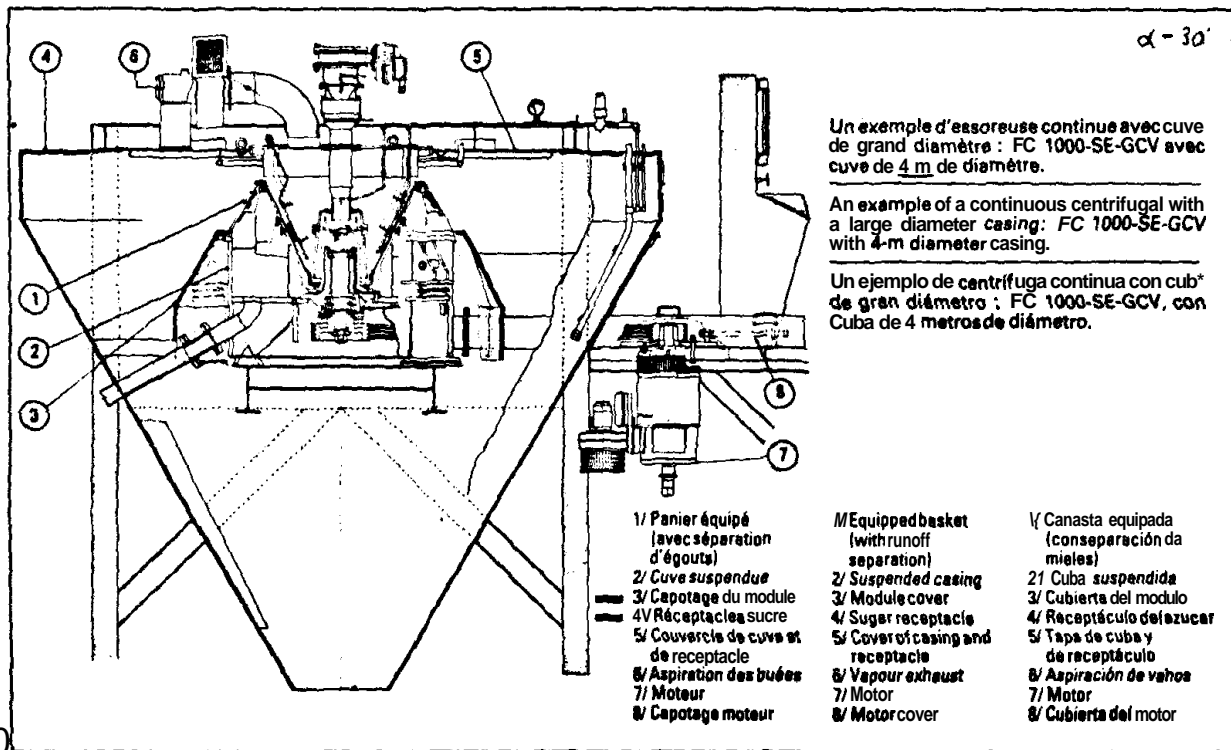
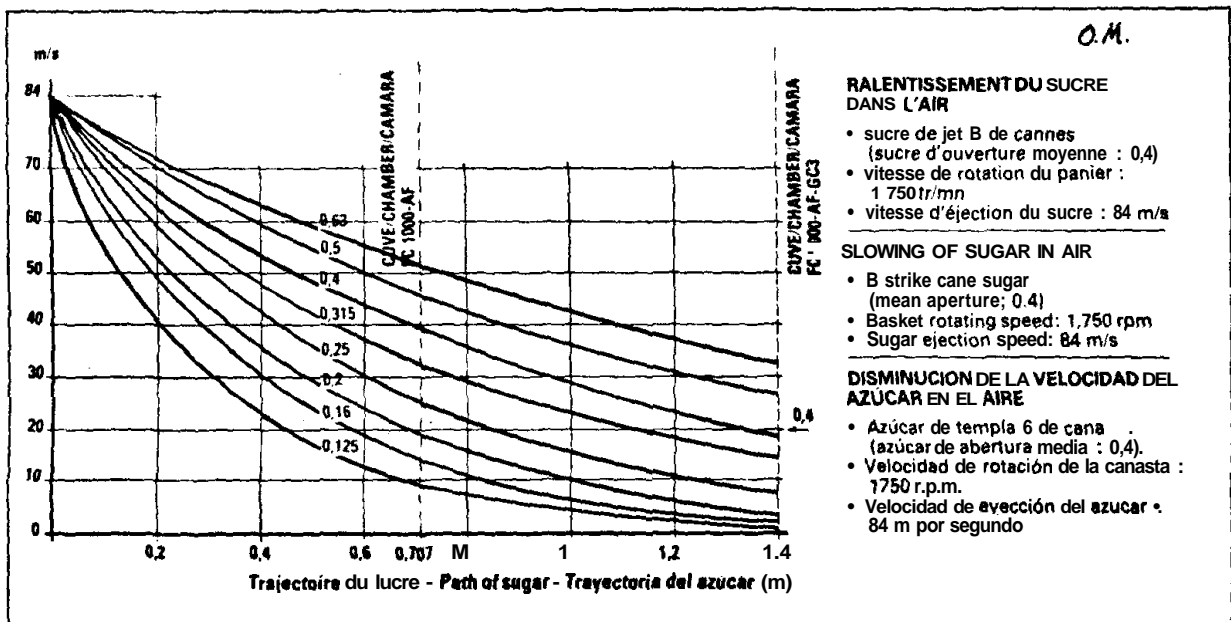


Fig. 35.17. Continuous centrifugal FC1000 GCV. Slowing action of air on crystals as a function of their size and the length of their path.

of 25° , which can treat A massecuite and furnish a commercial sugar with a minimum breakage of crystal (Fig. 35.17). Figure 35.17 shows graphs relating the speed of impact of the crystals to the distance travelled before striking the casing, as a function of their dimensions.

It has been proposed in Reunion to provide the upper edge of the casing of the GCV centrifugal (and continuous machines in general) with a curved surface designed to deaden the shock, soften the violent impact of the crystals, and make them gradually change direction (Fig. 35.18). This system, tried in Europe, has not given the expected results, but it seems that it can be perfected. The idea could be applied in the form of a bladder which, partially inflated, would be of similar shape and which would be inflated automatically at intervals of a few minutes, and would then assume a convex shape and dislodge the layers of crystals which tend to form at that location and constitute a nuisance with centrifugal casings. This has not yet been tried out, but it seems that it might prove satisfactory.

The speed of impact which is liable to break the crystals is of the order of $9-18$ m/s ($30-60$ ft./s). It is often with the object of keeping below this limiting speed that small machines of 0.6 m (24 in.) diameter are used.

Specific work

Repeating in another form the calculation leading to eqn. (35.17): the centrifugal force per unit mass of material or, which amounts to the same thing, the centrifugal force relative to gravity, has the value (eqn. (35.4)):

$$G = \frac{mv^2/r}{mg} \approx \frac{2v^2}{Dg} \quad (35.46)$$

The work necessary (per unit mass) to obtain this separating force G is equal to the kinetic energy per unit mass:

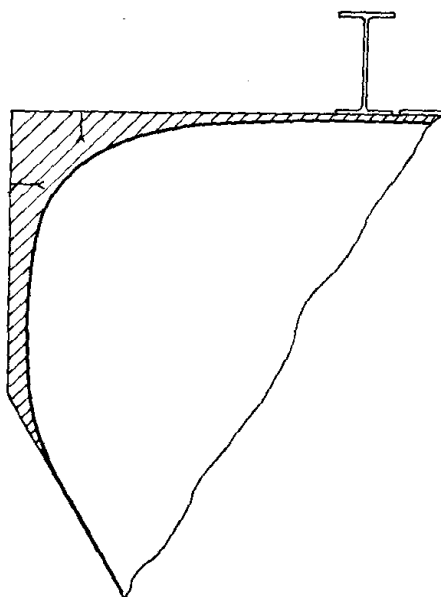


Fig. 35.18. Damping device proposed as shock absorber for crystals.

$$T = \frac{1}{2}mv^2 \tag{35.47}$$

It follows that the work expended per unit separating force, that is, per unit of gravity, has the value:

$$\frac{T}{G} = \frac{\frac{1}{2}mv^2}{2v^2/Dg} = \frac{m g}{4} D \tag{35.48}$$

It is thus proportional to D . It follows that both from the point of view of specific work and from that of avoiding crystal damage, it is of advantage to use small diameters.

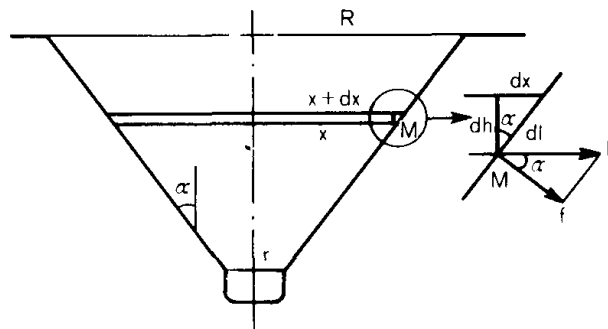


Fig. 35.19. Calculation of capacity.

Capacity

The capacity of a continuous centrifuge is measured by the quantity of massecuite which can be passed through the machine per hour. This quantity depends on the separating capacity of the machine. Now, if we consider an element of mass which we shall take as unity, $dm = 1$, the separating capacity or drying capacity will be proportional to the product of the centrifugal force developed on the unit mass, by the area offered to the total mass of material. We have then (Fig. 35.19):

(a) Component of centrifugal force perpendicular to the screen:

$$f = dm \omega^2 x \cos a = 1 \times \omega^2 x \cos a$$

(b) Area offered = ds ; thus:

$$\text{Capacity} = dq = kfAs = k \omega^2 \cos a x ds$$

Considering the annulus of massecuite of radius x and height dh , situated at the level M , the centrifugal force is the same for all elements dm of this annulus. We shall obtain the total capacity Q by integrating the capacities dq along the cone, from r to R . We have:

$$dq = k \omega^2 \cos a x ds$$

$$ds = 2\pi x \, dl = 2\pi x \frac{dh}{\cos \alpha} = 2\pi x \frac{dx}{\cos a \tan a}$$

Hence:

$$dq = k\omega^2 \cos a x \cdot 2\pi x \frac{dx}{\cos a \tan a} = \frac{2\pi k\omega^2}{\tan a} x^2 \, dx$$

Integrating:

$$Q = \int_r^R \frac{2\pi k\omega^2}{\tan a} x^2 \, dx = \frac{2\pi k\omega^2}{\tan a} \int_r^R x^2 \, dx = \frac{2\pi k\omega^2}{\tan a} \left(\frac{R^3}{3} - \frac{r^3}{3} \right)$$

$$Q = 2k\omega^2 \frac{\pi (R^3 - r^3)}{3 \tan a} \quad (35.49)$$

It will be seen that the fraction occurring in this expression represents the volume V of the portion of the cone forming the centrifugal. We may thus write:

$$Q = 2k\omega^2 V \quad (35.50)$$

Q - capacity of the centrifugal, in tonnes of massecuite per hour

k - numerical coefficient

ω - rotational speed of the machine

V - volume of the frustrum of the cone forming the centrifugal

a - apical angle of *this* cone

R - radius of the upper circle of discharge of the sugar

r - radius of the small lower basin receiving the massecuite.

The capacity of a continuous centrifugal is thus proportional to its volume, to the square of its angular velocity, and to a coefficient which depends on the massecuite being handled (A, B or C), on its consistency and its temperature, and on the percentage of openings in the perforated screen.

Hourly capacity

Eastaughffe¹⁹ considers that:

(a) Capacity is proportional to:

$$q = \frac{\lambda}{M \sqrt{\mu}}$$

(b) Effective area drainage A :

$$A = \pi l \frac{D + d}{2}$$

(c) Effective force F :

$$F = n^2 \frac{\sqrt{D^2 + d^2}}{2} \quad (n \text{ in r.p.m.})$$

(d) Transit time t is proportional to:

$$t = \lambda^1 \sqrt{\frac{1}{\sin \alpha}}$$

Hence capacity in tonnes per hour:

$$Q = \frac{F}{q} = k \frac{2 \sqrt{\frac{D^2 + d^2}{2}} l \frac{(D + d)}{2} \sqrt{r}}{\sqrt{\sin \alpha} M \sqrt{\mu}} \quad (35.51)$$

k = coefficient

l = length along slope of cone = $\frac{R - r}{\sin \alpha}$

$D = 2R$

$d = 2r$

M = molasses removed % massecuite

μ = viscosity of molasses, in poise.

Capacity in massecuite and capacity in sugar

Delden²⁰ has very aptly remarked that the capacity of centrifugals in general, and of continuous centrifugals in particular, when reckoned in weight of sugar per hour, depends on the crystal content of massecuite. The factor limiting the capacity of the machine is the quantity of massecuite fed to it, much more than the output of sugar which it gives. Now, the proportion of crystal has the value (eqn. (33.46)):

$$x \sim \frac{P_s}{100} \frac{P_m - P_e}{P_s - P_e} m$$

The essential factor here is the purity drop ($P_m - P_e$). If, for the same sugar and the same brix of massecuite, we have different purity drops, the capacity of the centrifugal in terms of sugar will vary as a result.

Example. For

$$\left\{ \begin{array}{l} P_s = 96 \\ P_m = 56 \\ B_m = 98 \end{array} \right\} \text{ and comparing the 2 cases } \begin{cases} (1) P_e = 35 \\ (2) P_e = 30 \end{cases}$$

TABLE 35,14

DIMENSIONS AND CAPACITIES OF CONTINUOUS CENTRIFUGALS

Make and Model	Basket					Casing diam.		Power (kW)	Gauze openings (mi- crons)	Capacity (t/h)	Optimal wash (% m.c.)	Steam for purging		
	Angle (deg.)	Top diam.		Screen area		Speed (r.p.m.)	(m)					(ft./in.)	(bar)	(p.s.i.)
		(mm)	(in.)	(m ²)	(sq.ft.)									
FCB FC1000 BP (for C massecuite low grade)	34	981	39	1.098	11.8	1,930	1.75	5 ft. 9 in.	45	60	4.5- 8	2.5	0.4-0.7	6-10
FCB FC1000 AF (for C massecuite and affined C)	30	901	35½	0.989	10.6	1,930* 1,750 1,500	1.75	5 ft. 9 in.	55 45	90 or 130	10 -13	2.5-5.0	0.4-0.8	6-11
FCB FC 1000 SE (for A massecuite and affined B)	25	911	36	1.203	13	1,750* 1,500 1,200	1.75	5 ft. 9 in.	55 45 (ord. c.)	90 or 130	18 -30	3 -5	0.2-0.6	3- 9
FCB FC 1250 (for C massecuite in large factories)	34	1,237	48	1.832	19.7	1,880 (1,600 affining)	2.15	7 ft. 0 in.	90	60 or 90	9 (to 20 in affining)			
B.M.A. K850S	30-34	850	33½	0.8	8.6	1,650- 2,400	1.75	5 ft. 9 in.	30-37 at 1,450 r.p.m.	60 or 90	4-6 (C) 7-10 (B)			
B.M.A. K1100	30-35	1,100	43	1.45	15.6	1,400- 2,000	1.90	6 ft. 3 in.	55-75 (induction)	60 or 90	10-11			
B.M.A. K1500	30	1,500	60	2.32	25	1,700	2.40	8 ft. 0 in.	90	60 or 90	24-30 (A) 10-15 (C)			
Western States CC5	30	940	37	1.07	11.5	2,480	1.66	5 ft. 6 in.	37	2175	4- 8			

*Depending on massecuite purity.

The FC1000 AF centrifugal is generally supplied as GC3, *i.e.* with casing 3 m in diameter, to avoid grain breakage as far as possible. It can also be supplied with remelt incorporated.

The FC1000 SE can also be supplied as GC3 or GCV with 4- or 5-m casing to reduce grain breakage; also aspiration of steam.

We have:

$$(1) \quad x = \frac{96}{100} \cdot \frac{56 - 35}{96 - 35} \cdot 98 = 32.4\%$$

$$(2) \quad x = \frac{96}{100} \cdot \frac{56 - 30}{96 - 30} \cdot 98 = 37.1\%$$

or 15% more in the second case.

It is thus necessary to specify either the purity drop or the crystal content, when stating a capacity in weight of sugar per hour.

Equipment available. Dimensions and capacities of continuous machines offered by Fives Cail – Babcock, B.M.A. and Western States are listed in Table 35.14.

Installation and accessory equipment

Below the centrifugals, the discharge chute receiving the sugar is generally formed simply of two sheets of metal embracing the discharge opening of the centrifugals and sloping to the sugar conveyor. The angle of this chute with the horizontal should be at least 60°, and this applies for both batch and continuous machines.

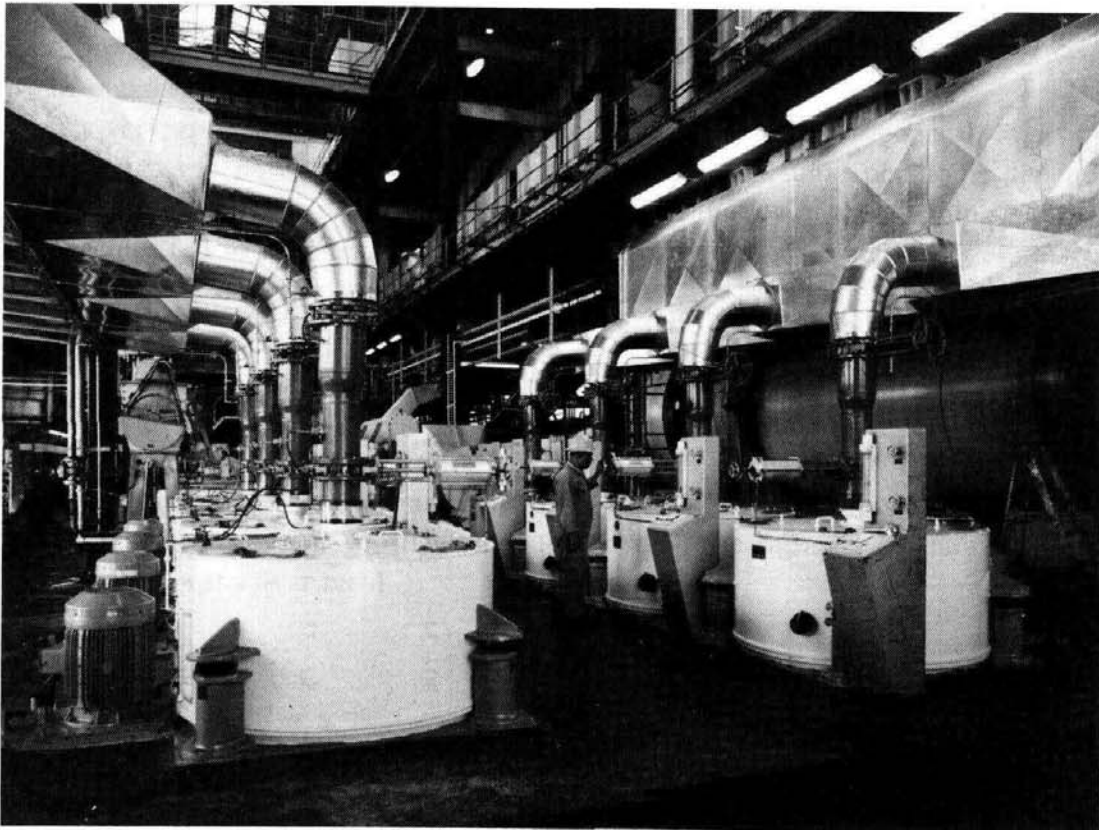


Fig. 35.20. Battery of FC1000 continuous centrifugals (Fives Cail – Babcock).

The discharge opening should be at least 400 mm (16 in.) for 1.015–1.065-mm (40–42-in.) centrifugals, and at least 600 mm (24 in.) for those of 1,220–1,370 mm (48–54 in.).

The screens or gauzes for continuous centrifugals are of chrome nickel sheets with perforations (in mm):

0.06 X 1.66, sheet thickness 0.23–0.25

0.06 x 2.2 , sheet thickness 0.28–0.30

0.09 x 2.3 , sheet thickness 0.24

For brass gauzes, brass of 70% copper, 30% zinc is employed, 0.7 mm in thickness. In Arcap stainless steel, the thickness is 0.5 mm. European suppliers of these gauzes are Krieg and Zivy in France (Ref. N591), and Fontaine at Aachen in Germany.

Exhaustion

The second complaint reported regarding continuous centrifugals, and probably the more disturbing, is that the purity of the molasses obtained is generally higher than that of the molasses obtained by ordinary centrifugals working with the same massecuite*.

This fact has often been verified. It arises mainly from the fact that it is scarcely possible to use continuous centrifugals on C massecuite without a slight water wash. The thickness of the layer of sugar being very small, there is unavoidably a slight dissolution of sugar. It is thus essential to regulate the quantity of wash water with the greatest care.

However, this difference in purity may be kept below 1 point; sometimes it does not exceed $\frac{1}{2}$ point. It should thus be kept in balance with the economies in labour, maintenance and power offered by the continuous centrifugal.

It has been shown²¹ that breakage of grain in continuous centrifugals contributes to increased purity of molasses due to fine broken crystals passing to a great extent through the gauze. For example, the dimension of the crystals in the massecuite was shown to be 128 microns, and the sugar average 107 microns. On the other hand, the perforations of the gauze become worn rapidly, and can increase in 100 days' operation from 60 microns to 160. It is necessary to change the screen when this increase is detected: a gain of more than 2 points in purity may thus be realised.

Washing may be effected in two ways, with continuous machines:

(a) Sometimes (as with **Hein-Lehmann**) by a co-axial arrangement, by introducing the water in the axis of the column of massecuite flowing into the centrifugal.

(b) By radial washing, by a spray pipe placed parallel to a generatrix of the cone. The radial washing may be adjusted and even reduced to zero, but this is difficult with co-axial addition of wash water.

The temperature of the wash water should be of the order of 80–90°C (175–195°F).

* However, F. M. Chapman in personal discussion has expressed the opinion that continuous machines can handle heavier massecuite and so could be operated to give better exhaustion than batch machines. Obviously further investigation is desirable. G.H.J.

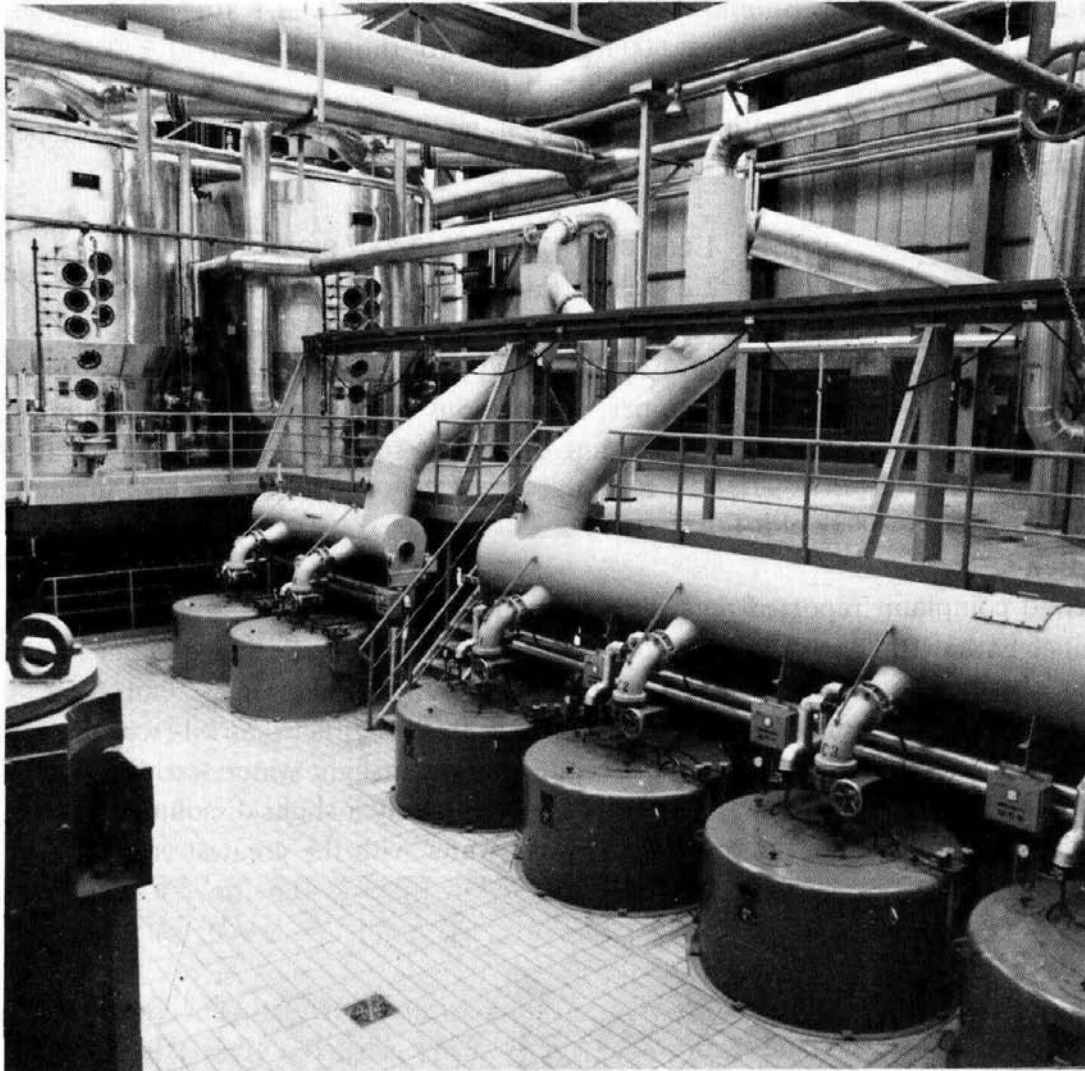


Fig. 35.21. Battery of K 850 S continuous centrifugals (B.M.A.).

With C massecuite, the total wash should be about 2–3% on the weight of massecuite. The optimal co-axial washing is 1.5%; the radial washing may supply 5 of the total wash.

With B massecuite, the wash water may be between 2.5 and 6% on weight of massecuite.

The most frequent cause of poor exhaustion in a continuous centrifugal is reduction of input rate of massecuite while the wash water is left constant. It is preferable to avoid such changes in rate or, better, to eliminate the trouble by adjusting the water to suit the massecuite.

Conclusion

Continuous centrifugals are probably destined to become universal. At present (1982), they have taken over the B and C centrifugal stations. They are only beginning to be tried with A massecuite, where they encounter strong reservations on account of crystal quality. There is every reason to suppose that the progress attending the first installations will permit them to resolve the problems which are encountered at present. We know of two factories already

where the full manufacture is on a continuous basis, in pans and centrifugals, for the three massecuites.

ACCESSORY PROCESSES AND EQUIPMENT

Dilution of massecuite

The density of the massecuite and the viscosity of the molasses often cause difficulties in the handling and fugalling of the massecuites. The simplest means of remedying this is to dilute the massecuite. This is generally carried out in the discharge chute from the crystallisers, as this assures a more even distribution of the water. If the water is added in the crystalliser, it is difficult to obtain an even dilution, and sugar is unavoidably dissolved. In any case this is an unsatisfactory expedient.

Quantity of water to add. It is not necessary to introduce more than 1 or 2% of water by weight of massecuite, or 1 – 3% by volume, otherwise there is a risk of reducing the supersaturation to 1 or less.

Reheating of massecuite

It is very difficult to dilute a massecuite without causing a certain re-resolution of crystals. Fortunately, there exists another means of decreasing viscosity and facilitating the fugalling; this is reheating.

In Hawaii²² it is considered that the viscosity of the massecuite falls by 50% (or, better, that its “fugability” rises by 50%) when the temperature is increased by 5°C (9°F).

The reheating is then carried out, either in the last discs of the crystalliser, where a Werkspoor is installed, or in the mixer – distributor supplying the centrifugals; in the latter case it may be effected by means of a double bottom, or by means of a coil, or again by electrical-resistance heating between the distributor and the centrifugal.

The area of coil required is 30 – 35 m²/m³ (9 – 11 sq.ft./cu.ft.) of massecuite to be fugalled per hour.

These areas are calculated on the assumption that the water used for reheating and circulated in the coil or the heat-exchanger is at a temperature which is not more than 3°C (5°F) above the saturation temperature of the massecuite, that is, the temperature above which crystals commence to dissolve (*cf.* Table 33.1).

Some manufacturers base their designs on larger temperature differences, and consequently supply heating coils in the mixer-distributor of much smaller area, of the order of 3 m²/m³ (1 sq.ft./cu.ft.) of massecuite to be fugalled per hour. However, it must not be forgotten that the low-grade fugal cycle in Hawaii is of the order of 1 hour; if the heating surface is expressed in terms of area per machine of 1,015 X 760 mm (40 X 30 in.), for example, instead of being related to volume of massecuite, the differences given above will be much less.

In any case, it is recommended that the temperature of the water used should not exceed the limit indicated (3°C above the saturation temperature).

For massecuite heaters with coils, a heat-transfer coefficient of 30 – 70 kcal/m²/°C/h (6 – 14 B.T.U./sq.ft./°F/h) is taken, according to the design and conditions of use²³.

Electrical-resistance heaters

Massecuite heating by electrical resistance has been developed in Queensland. It offers

several advantages:

- (a) Simplicity
- (b) Easier and more effective temperature control
- (c) Shorter time at temperature, hence less danger of re-solution
- (d) Excellent adaptation to continuous centrifugals.

The resistance heater (Fig. 35.22) consists of two concentric pipes; the massecuite is passed through the annular space between them. These two pipes form two electrodes between which a potential difference is applied. The massecuite flows by gravity from a mixer above to a distributor running along the battery (in the case of batch centrifugals) or directly to the centrifugals in the case of continuous machines (see Fig. 35.22). The latter arrangement is particularly favourable on account of the regular and continuous flow and the complete elimination of delay between reheating and fugging.

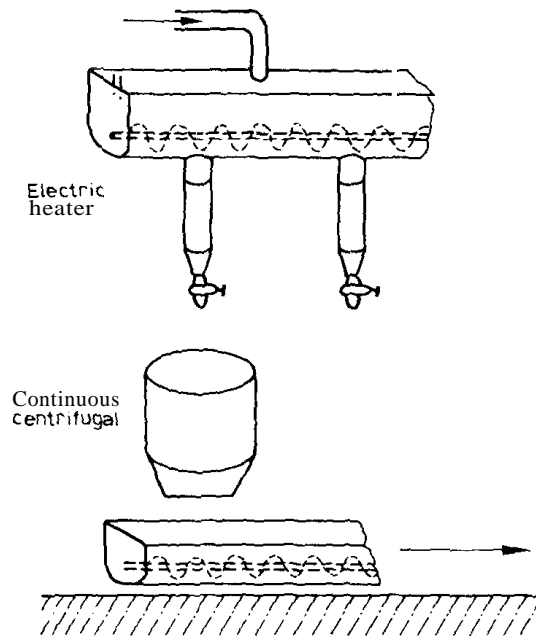


Fig. 35.22. Installation of electrical resistance heater with continuous centrifugal.

Head loss. The loss of head in the reheater is given by our formula (33.18), in which we give:

- (a) The head loss J' not in lb./sq.ft. but in ft.:

$$1 \text{ ft. of massecuite} = 94 \text{ lb./sq.ft.}$$

- (b) The flow of massecuite Q not in cu.ft./s but in cu.ft./hour:

$$1 \text{ cu.ft./s} = 3,600 \text{ cu.ft./hour}$$

- (c) The exterior diameter A and the interior diameter F of the annulus, which we shall call here D and d , now expressed in cm and inches.

We thus have (cf. eqn. (33.19)):

$$j = \frac{J}{L} = \frac{10^8}{10 g \pi (D^2 - d^2) (D - d)^2} \frac{1500 \times 3,600,000}{m/m}$$

$$\left(j = \frac{J}{L} = \frac{128 \times 0.75}{g \pi} \times \frac{\mu Q}{(D^2 - d^2) (D - d)^2} \times \frac{20,736}{94 \times 3,600} \text{ ft/ft} \right)$$

or:

$$J = \frac{0.00575 \mu Q}{(D^2 - d^2) (D - d)^2} \text{ m/m} \quad \left(j = \frac{0.058 \mu Q}{(D^2 - d^2) (D - d)^2} \text{ ft/ft} \right)$$

To take into account the method of determination of viscosity, the film effect, the resistance of the protective grilles which are interposed at the entry above and at the delivery below the heater, as well as the approximations involved, particularly the replacement of the logarithmic mean of the viscosity μ by its geometric mean (see below), Wright and Walkers Ltd. reckon:

$$j = \frac{0.025 \mu Q}{(D^2 - d^2) (D - d)^2} \quad \left(j = \frac{0.25 \mu Q}{(D^2 - d^2) (D - d)^2} \right) \quad (35.52)$$

j ~ loss of head in the heater, in m/m (ft. per ft.) of length

μ — viscosity, in poises (f.p.s. units)

Q - flow of massecuite through the heater, in litres/h (cu.ft./h)

D = exterior diameter of the annular passage, in cm (in.)

d - interior diameter of the annular passage, in cm (in.)

Common dimensions are $D = 38$ cm (15 in.), $d \approx 20$ or 23 cm (8 or 9 in.).

Since it is obviously necessary to maintain flow through the heater, it is obvious that j must be less than 1. The best procedure is to allow a margin of safety for variations in the flow required, for example, 20%, and to take consequently $Q' = 1.2 Q$, thus making $j \approx 1$. The regulating valve below the heater will thus normally not be fully open.

The viscosity of cold massecuite may vary between 350 and 2,000 f.p.s. units (5,000-30,000 poise). A typical order of magnitude is 10,000 poise (700 f.p.s. units). The heater will reduce it to between 2,000 and 4,000 poise. We may take for the mean viscosity the geometric mean between the viscosity at entry and that at exit from the heater. For example, if the massecuite is at 12,000 poise (800 f.p.s.) when cold and 3,000 poise (200 f.p.s.) at the temperature required for fugalling, we may take:

$$\mu = \sqrt{12,000 \times 3,000} = 6,000 \text{ poises} \quad (\mu = \sqrt{800 \times 200} = 400 \text{ f.p.s. units})$$

The Wright and Walkers formula gives, for the valid reasons given above, higher figures than ours.

Power consumption. Assuming the reheating 100% efficient, we have:

$$P = \frac{Q \times 1.5 (t - t_0) c}{860} \quad \text{or} \quad P = \frac{Q \times 94 (t - t_0) c}{3,413} \quad (35.53)$$

P = power consumption required for heating, in kW

Q = massecuite flow, in litres/h (cu.ft./h)

c = specific heat of massecuite = 0.40 – 0.44

t_0 = initial temperature of massecuite, in °C (°F)

t = temperature of massecuite after heating, in °C (°F)

with:

1 kWh = 860 kcal = 3,413 B.T.U.

Density of massecuite = 1.5 kg/litre (94 lb./cu.ft.)

Or:

$$P = 0.00079 Q (t - t_0) \quad (P = 0.012 Q (t - t_0)) \quad (35.54)$$

For example, to reheat 100 cu.ft./h of massecuite from 100 to 120°F, the power consumption would be:

$$P = 0.012 \times 100 \times 20 = 24 \text{ kW (32 h.p.)}$$

Length. Wright²⁴ gives the equation (35.55):

$$L = 3.68 \times 10^6 \frac{P}{kV^2 \cos \phi} \log \frac{D}{d} \quad \left(L = 4.75 \times 10^6 \frac{P}{kV^2 \cos \phi} \log \frac{D}{d} \right) \quad (35.55)$$

L = length of active portion of heater, in m (ft.)

P = power requirement for reheating, in kW, given by eqn. (35.54)

k = specific conductance (logarithmic mean) of the massecuite, in micromhos/cm (in.)

V = voltage applied, in V

$\cos \phi$ = power factor of the supply.

We may take approximately $k = 6.5 (T - 30)$, where T - mean temperature (°C) of the massecuite between entry and exit.

The overall length generally adopted is 168 cm (5 ft. 6 in.). The best method of regulating the temperature reheating is by means of a silicon rectifier.

Equation (32.56) may also supply the value of P for a given heater.

Conductivity. Massecuites normally have a conductivity of 50 – 250 micromhos/cm at the temperature of fugalling, this conductivity increasing rapidly as a function of temperature. Reheating is generally effected by a continuous cylindrical electrode on the inner face of the annulus and a succession of metal rings on the outer circumference distributed over the height of the heater, which are switched on and off by the temperature control in order to

achieve the set temperature. The change of conductivity with temperature means that the current supplied by the last ring is appreciably greater than that supplied by the first.

The mean conductivity of the massecuite between the cold entry and the hot exit is fairly uniform and of the order of 100 micromhos/cm. Heaters may be equipped with ASEA thyristor controls, housed in a suitable cabinet, which regulate the current and consequently the temperature.

Improvement by reheating

Reheating massecuite substantially improves exhaustion. At Central Jaronu²⁵, in Cuba, which at one time was the biggest sugar factory in the world, the installation of reheating led to the following results:

Reduction in number of fugals from 30 to 20

Increase of C sugar purity from 84.73 to 89.43

Reduction of molasses purity from 32.62 to 31.32

Reduction of fugalling cycle from 16.5 to 10 min

The massecuite was heated to 59°C (138°F).

At Raceland²⁶, in Louisiana, the fugalling temperatures of massecuites as quoted by Honig were:

Cooling to 38–40°C (100–104°F) (mother liquor supersaturated)

Reheating to 52–55°C (126–131°F) (mother liquor saturated)

Fugalling at that temperature

Washing with hot water at 60–65°C (140–149°F), under 2 kg/cm² (30 p.s.i.), by fine spray

Quantity of wash water = 3 l per 1,015-mm machine (0.66 gal. per 40-in. machine).

In Hawaii, a temperature of 49°C (120°F) is not exceeded.

The optimal limit of reheating is approximately 50–52°C (122–125°F) in a mixer or mixer-distributor; with electrical-resistance heaters, since the massecuite remains a very short time at high temperature, this can easily be taken as high as 57°C (135°F).

If we consider supersaturation rather than purity, and trace the graph of viscosity as a function of temperature, it is found that it passes through a minimum between 48 and 55°C (119–131°F)²⁸.

Brix of molasses

Pagnier²⁹, in Java, estimated that the optimal concentration for fugalling corresponded to a molasses of 85° refractometric brix. In Java³⁰, however, at the beginning of the Second World War, the average molasses attained a densimetric brix of 95° (dilution 1 + 9).

Mixer-distributor for massecuite

The small mixer-distributor for massecuite placed above the centrifugals, which forms part of the battery, should have a working volume corresponding to the volume of massecuite treated in 15 or 30 min.

The agitator should be rotated at a speed n given by:

$$n = \frac{3.25}{D} \quad \left(n' = \frac{10.7}{D} \right) \quad (35.56)$$

n - rotational speed of the agitator, in r.p.m.

D = width of the trough of the mixer-distributor, in m (ft.) (or diameter of the helical or paddle agitator).

Screw conveyors for massecuite

The massecuite is generally conveyed by a screw conveyor in a horizontal trough. A full screw may be used, but more often the screw consists of a simple metal ribbon held in place by steel strips.

The pitch p of the helix is of the order of $p = 0.7 D$, D being the diameter of the cylinder enclosing the outer edge of the helical ribbon. The speed n of the rotation of screw is generally from 20 to 40 r.p.m., or, better:

$$n = \frac{15}{D} \quad \left(n = \frac{50}{D} \right) \quad (35.57)$$

n = rotational speed of the screw, in r.p.m.

D = diameter of the screw, in m (ft.).

However, Tromp³¹ recommends a speed of 7-8 r.p.m., above which he considers that there will be too much back-flow of a large proportion of the massecuite.

Output. The output of a screw conveyor is given by the formula:

$$V = 60 \frac{\pi D^2}{4} p n k = 47 p n k D^2 \quad (35.58)$$

V = volume of massecuite transported, in m³/h (cu.ft./h)

D = exterior diameter of the helix, in m (ft.)

p = pitch of the helix, in m (ft.)

n = rotational speed of the helix, in r.p.m.

k = coefficient taking into account the extent to which the trough is filled (which should not be above the shaft) and the rate of movement of the material, which does not advance a distance p when the screw makes one revolution.

In general:

D is of the order of 0.3 – 0.6 m (1 – 2 ft.)

p is of the order of 0.7 D

n is of the order of 15 – 40 r.p.m.

We may assume for k :

0.15 — 0.20 for ribbon screws, or a mean value of 0.18

0.25 – 0.30 for a full screw.

Power. For the power required for driving the screw conveyor, we may estimate approximately:

$$P = 1.15 D^2 L \quad (P = 0.043 D^2 L) \quad (35.59)$$

P = power absorbed by the conveyor, in kW (h.p.)

D = exterior diameter of the helix, in m (ft.)

L = length of the helix, in m (ft.).

We must remember to take into account also the efficiency of the speed reducer or the system of drive.

Sugar conveyors

There are two principal types of sugar conveyors:

(a) The screw conveyor

(b) The shaker conveyor, sometimes designated under the name of "grasshopper".

We shall consider the method of designing these, and shall adopt, especially in the second case, Tromp's mode of calculation³¹.

(a) **Screw conveyor.** This conveyor is calculated exactly as for massecuite (*cf. p.* 812). The output is given by:

$$Q = 60 \frac{\pi D^2}{4} p n k d = 47 p n k d D^2 \quad (35.60)$$

Q = weight of sugar transported, in kg/h (lb./h)

D = exterior diameter of the screw, in m (ft.)

p = pitch of the screw, in m (ft.)

n = rotational speed of the screw, in r.p.m.

k - coefficient taking into account the degree of filling of the screw and of backward flow of the sugar

d = apparent density of the sugar, in kg/m³ (lb./cu.ft.).

In general:

p is of the order of 0.7 D

n ranges from 15 to 25 r.p.m. (or $10/D$, D being expressed in m)

d ranges from 800 to 950 kg/m³ (50 – 60 lb./cu.ft.).

We may take for k :

0.10–0.12 for ribbon screws (Tromp gives 0.20)

0.15–0.20 for a full screw.

The power consumption is approximately:

$$P = 0.8 D^2 L \quad (P = 0.03 D^2 L) \quad (35.61)$$

P = power absorbed by the sugar conveyor, in kW (h.p.)

D = exterior diameter of the screw, in m (ft.)

L = length of the screw, in m (ft.).

Tromp recommends:

(1) That the shaft be supported every 3 m (10 ft.), in the trough

(2) That a clearance of 6 mm ($\frac{1}{4}$ in.) be allowed between the screw and the trough, to avoid crushing the crystals

(3) That the sugar which collects and hardens in this space between screw and trough should be removed every week.

Screw conveyors have the disadvantage of crushing the sugar. For this reason, grasshopper conveyors are generally preferred.

(b) Grasshopper conveyor. This conveyor consists of a wide, flat and shallow trough, supported on flexible strips inclined at 60° to the horizontal (Fig. 35.23). The trough is subjected to repeated vibrations by means of an eccentric. The latter has a stroke c of approximately 40 mm ($1\frac{1}{2}$ in.) and rotates at about 300 r.p.m.

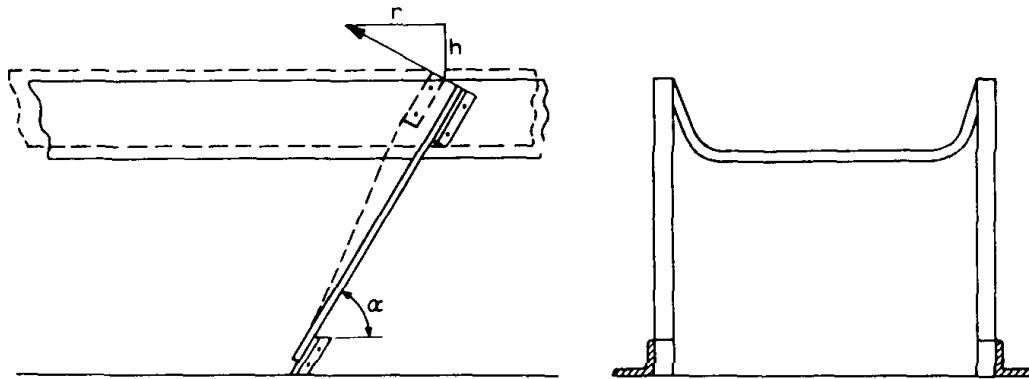


Fig. 35.23. Grasshopper conveyor.

At the moment of the vibration, the grains of sugar situated in the trough make a horizontal jump of:

$$r = \sin 2 \left(\frac{\pi}{2} - \alpha \right) \frac{v^2}{g} \quad (35.62)$$

a = inclination of the flexible strips to the horizontal

g = acceleration due to gravity = 9.81 m/s/s (32.16 ft./s/s)

v = speed of projection of the grain = approximately 1.5 times the mean speed of the eccentric, in m/s (ft./s):

$$v = 1.5 \frac{2cn}{60} = \frac{cn}{20} \quad (35.63)$$

c = stroke of the eccentric, in m (ft.)

n = speed of rotation of the eccentric, in r.p.m.

Now, while the sugar is thus thrown forward, the conveyor returns under it, and it is arranged so that the return stroke corresponds very closely to the time during which the crystals are being thrown forward, in such a way that the sugar thus proceeds again through a distance equal to the return stroke:

$$r' = c \sin a \quad (35.64)$$

whence:

$$A = r + r'$$

We have then:

$$Q = SAnkd \quad (35.65)$$

Q = output of the conveyor, in kg (lb.) of sugar/min

S = useful vertical cross-section of the trough, in m² (sq.ft.)

$A = r + r'$ = movement of the sugar per stroke, in m (ft.)

n = rotational speed of the eccentric, in r.p.m. = number of vibrations/min

k = coefficient of slip (approximately 0.5)

d = apparent density of the sugar, in kg/m³ (800-950) (50-60 lb./cu.ft.)

If necessary, the sugar may be elevated by inclining the conveyor, provided that an angle of 30° with the horizontal is not exceeded.

The best material for the flexible strips is ash timber.

Power. The power requirements of the motor for a grasshopper conveyor may be calculated approximately as follows:

$$P = 2.25 + S \quad \left(P = 3 + \frac{S}{10} \right) \quad (35.66)$$

P = power of the motor, in kW (h.p.)

S = horizontal surface of the trough (= length x width), in m² (sq.ft.).

This equation is useful for horizontal conveyors. If they are inclined and elevate the sugar, it is necessary to add the corresponding work to be done, that is:

$$P' = \frac{QH}{60 \times 100} \quad P' = \frac{QH}{60 \times 550} \quad (35.67)$$

P' = additional power required, in kW (h.p.)

Q = output of the conveyor, in kg (lb.) of sugar/min

H = mean distance in level between the discharge end of the conveyor and the points at which sugar is discharged from the centrifugals, in m (ft.).

Sugar elevators

The sugar is often transported to the dryer by an elevator which may be vertical or inclined. It consists of a chain or a belt carrying buckets, running on a return pulley at the bottom, and driven by a similar pulley at the top of the elevator.

The distance between buckets on the belt is 30–40 cm (12–16 in.). The speed of the belt should be maintained between 18 and 30 m/min (60–100 ft./min). The capacity will be:

$$Q = GVN \quad (35.68)$$

Q = output of sugar, in kg/min (lb./min)

G = weight of sugar conveyed per bucket, in kg (lb.)

V = speed of the belt, in m/min (ft./min)

N = number of buckets/m (buckets/ft.) of belt.

For G , we may take half the weight of water which would fill the bucket on its upward run.

Rubber conveyors may be inclined at 23°

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36. Storage and Drying of Sugar

The commercial sugar leaving the centrifugals, which is to be packed for sale or export, generally has a moisture content of 0.5–2%. Moisture is very detrimental to keeping qualities of the sugar, when it exceeds a certain limit, and particularly when it rises above 1%.

Safety factor

This limit depends on the proportion of impurities present in the sugar. A given percentage of water is more detrimental with a sugar of higher purity. If we put:

$$f = \frac{\% \text{ water}}{100 - \text{polarisation}} \quad (36.1)$$

then if:

/ $<$ 0.3: the sugar will not deteriorate, or will deteriorate only slowly

/ $>$ 0.3: the sugar will deteriorate rapidly.

However, there is no general agreement on the limiting value of this factor f , known as the safety factor of the sugar. In India¹, for example, it is given as:

0.22 for white or refined sugars

0.20 for raw sugars

Generally a figure of 0.25 is used instead of 0.3. Between these values, keeping qualities are considered to be doubtful.

Dilution indicator

A modification of the safety factor is preferred in Australia and is coming into use in some other countries. This is the *Dilution Indicator (D.I.)*:

$$\text{D.I.} = \frac{\% \text{ moisture}}{100 - (\% \text{ pol} + \% \text{ moisture})} \times 100 \quad (36.2)$$

that is, moisture per cent non-pol. It is considered that values of D.I. above 50 (corresponding to a safety factor of 0.33) indicate considerable risk of deterioration.

The author recommends the use of the dilution indicator rather than the safety factor.

Conditions for storage of sugar

Spengler and Bottger² have established that between 10 and 30°C (50 and 86°F) the keeping qualities of sugar are independent of temperature. On the other hand, Webster³ has shown

that, between 31 and 40°C (88 and 104°F) and at humidities ranging from 50 to 78%, it is only at the lowest humidities and temperatures that the safety factor is valid.

Various authors differ on the critical humidity, for which they give values ranging from 50 to 75%. Webster has found the value to depend largely on reducing-sugar content.

In Puerto Rico⁴ it is considered that equilibrium exists between sugar and the atmosphere at a relative humidity of 63%; if the humidity falls below that value, the sugar will lose moisture.

According to Freise⁵, another factor influencing the keeping qualities of sugar is the proportion of oils and waxes contained in the material of the sacks, which renders them less permeable. Thus sugar stored in washed sacks will not keep as well as sugar in new sacks.

SUGAR STORES

Stores for bagged sugar

For good keeping qualities of the sugar in a store, it is necessary to observe the following precautions⁶:

(a) The sugar should be bagged dry, and not too hot. If the sugar is at a temperature above 38°C (100°F), it will harden in the sacks.

(b) The bottom layer of bags should be protected from dampness, by interposing a grating or a bitumen-lined paper.

(c) The bags should be arranged in large stacks, cubical as nearly as possible, to decrease the surface relative to the volume.

(d) The stacks should be covered with bitumen-lined paper, at the top and sides. In Java, it has been found advantageous to cover the stacks of bags with a layer of dry bagasse, designed to protect the stack from radiation from the roof. It is advisable also to paint the roof with aluminium paint to decrease its temperature.

Keeping qualities of sugar are worse:

(a) At higher temperatures

(b) At higher atmospheric humidities.

It would be of advantage to maintain the humidity in the sugar store as uniform and as low as possible. It should not rise above 65%.

Similarly, the sugar store should be kept at a temperature as uniform as possible. It should not be opened except on dry and cooler days.

It will be of advantage to have a recording thermometer and hygrometer in the store.

Volume required. When the sugar is arranged in a closely packed stack, its bulk density may be reckoned at 800 kg/m³ (50 lb./cu.ft.) of stack. It must not be overlooked that it is necessary to leave corridors around the stacks for traffic and for handling the sugar. It is likewise preferable not to stack the sugar close against the walls.

The use of portable conveyors and elevators reduces costs of handling and stacking.

When sugar is stacked in hessian bags, it is advisable to give the sides of the stack a batter of 20°. If paper bags are used, 7½° is sufficient. The height of the stack may be 25 bags in the former case and 40 in the latter⁷.

Bulk storage

Sugar stored in bulk in a silo deteriorates much more slowly than sugar in bags. The surface of the pile absorbs some moisture, but then forms a crust which retards penetration of moisture into the pile. The thicker this crust becomes, the more effectively it retards penetration of moisture. It is possible in this way to keep dry sugar for several years in tropical countries without appreciable loss of titre.

For bulk storage, calculations may be based on a bulk density of $800 - 900 \text{ kg/m}^3$ ($50 - 55 \text{ lb./cu.ft.}$). The sugar becomes considerably compacted in falling on to the pile; at a depth of 6 m (20 ft.), it would retain a density of 1000 kg/m^3 (62 lb./cu.ft.); for a deep silo, one may assume a mean density of 900 kg/m^3 (55 lb./cu.ft.)⁸.

The angle of repose of dry sugar is about $33 - 36^\circ$. It may increase as far as $50 - 53^\circ$ when the sugar is moist. It is the moisture content that determines this angle, and differing values would be found according to the moisture content of the sugar⁹. In Australia¹⁰, at Mourilyan, sugars from five factories investigated varied between 33 and 41° . In Reunion, values ranging from 33 to 36° are found. The silo at Townsville has been built with a roof angle of 35° .

For transport of bulk sugar, rubber belt conveyors should be limited¹¹ to a slope of 20° ; and a value of 16° is recommended. However, Meade¹² indicates a maximum of 22° . For high-quality raw sugars, the author adopts 20° .

For the speed of such conveyors, a value of 2.5 m/s (8 ft./s) may really be used, as this value is definitely exceeded by some manufacturers.

The pressure exerted by sugar in bulk is higher than is normally supposed. Economic designs of silos generally do not exceed 3 m (10 ft.) in the height of side walls if these are not buttressed, and consist mainly of a roof covering a pile of sugar, of prismatic shape (Fig. 36.1).

The sugar is delivered into the bulk store by the belt conveyor installed under the ridge of the roof, and is diverted into the stack either by a movable plough placed above the belt, or by a second belt conveyor which is movable and of length equal to half that of the bulk store, fed at its centre and situated below the other conveyor.

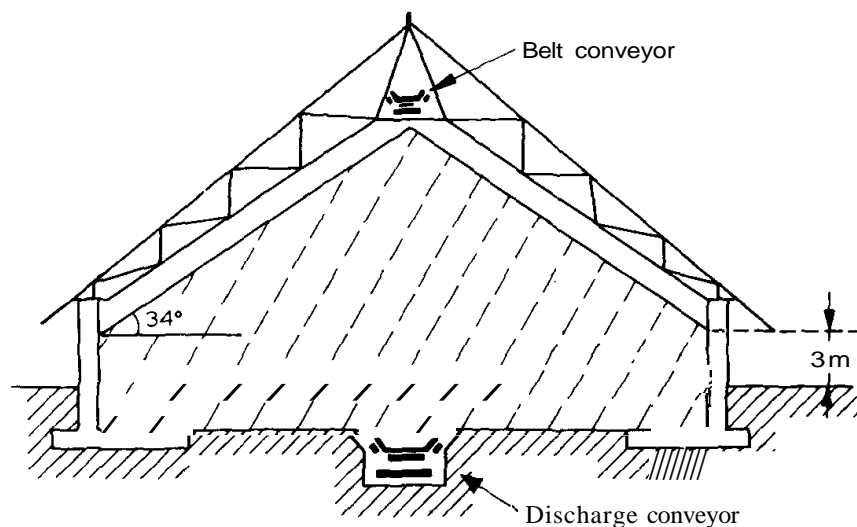


Fig. 36.1. Bulk sugar store.

Recovery of the sugar for loading ships, for example, is done from below, by hoppers which are covered with special slides. The sugar is pushed into the hoppers, picked up by a belt conveyor, and weighed on automatic scales (Servo-Balans or Schenk being well-known makes). A loading boom, moving on rails the length of the wharf, and fitted with an arm ending in a telescopic tube, delivers this sugar to the bottom of the hold. A thrower-distributor throws the sugar into the corners of the hold. The best-known make is Stephens-Adamson, made in France under licence by Boyer. There is also the Lucas Thrower, manufactured by Fletcher and Stewart.

Hoppers. To prevent sugar from adhering to the walls of the hoppers, it is necessary, when these are of quadrangular pyramid form, that the sides should have a slope of 77° , which corresponds to 72° in the corners of the hopper¹³. For a cylindro-conical hopper, 65° is sufficient.

Loss of weight and of test

With raw sugar from a first massecuite, with a good dilution indicator, stored in bags in tropical countries in a coastal climate, the loss in titre is approximately 0.1° (2—4 in recovery) per month. Handling and transport by sea aggravate the loss. When the sugar is already deteriorated, the loss becomes more and more rapid; in other words, the curve of titre as a function of time has a parabolic form and slopes rapidly downwards.

For sugar stored in bulk, it has been found at Kahului¹⁴ (Hawaii) that the loss of polarisation is:

<i>Factor of safety</i>	<i>Number of analyses</i>	<i>Mean loss of polarisation per month</i>
0.15-0.24	125	0.01
0.25-0.30	55	0.02

Specific heat of sugar

The specific heat of crystal sugar, between 15 and 75°C , is approximately¹⁵:

$$c = 0.28 \text{ kcal/kg (or B.T.U./lb.)}$$

or more accurately:

$$c = 0.2775 + 0.000\ 085\ t \qquad (c = 0.276 + 0.000\ 047\ t) \qquad (36.3)$$

t = temperature of sugar, in $^\circ\text{C}$ ($^\circ\text{F}$).

Drying of sugar

The deterioration of sugar is retarded and the loss in test is reduced if the moisture content of the sugar is reduced. The water content of raw sugar is generally within the range 0.5-2%. With a dryer, this may be reduced to between 0.2 and 0.5%. This gives a saving in two directions:

(a) The sugar keeps better

(b) The polarisation and the titre increase immediately, in proportion to the water removed, and if for example the polarisation increases from 97.8 to 98.2°, the financial gain so realised is much greater than the loss of weight due to the water evaporated.

Sugar dryer

Figure 36.2 illustrates a dryer. It consists of an air heater with fan, and is divided into a drying portion and a cooling portion.

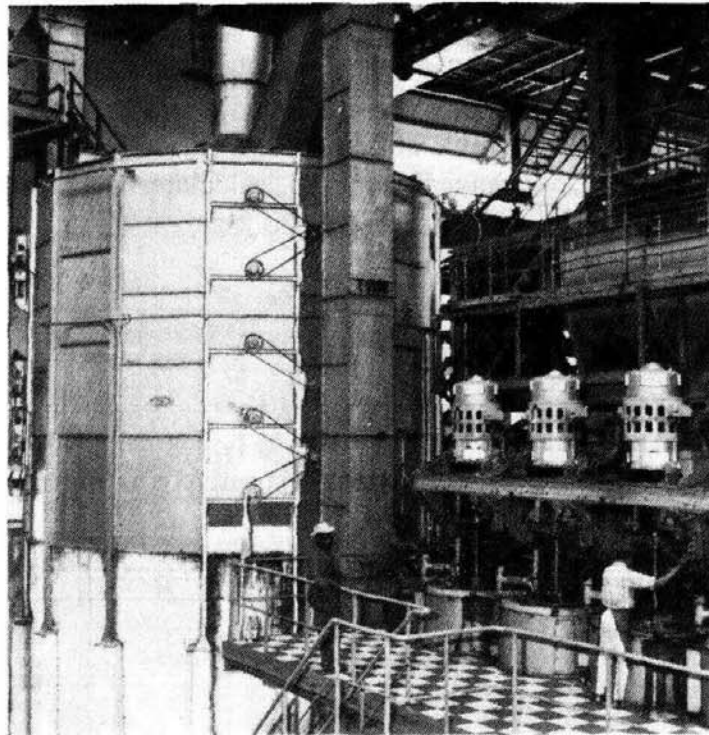


Fig. 36.2. Sugar dryer (Comessa).

Calculations for a sugar dryer. Drying by contact with hot air involves heating the air, to increase its capacity for absorbing water, and bringing it into intimate contact with the sugar, from which it evaporates the moisture.

(A) *Volume of air.* There are two possible methods of circulation:

(a) Countercurrent flow, where the air flows in the opposite direction to the sugar

(b) Parallel flow, where the air and the sugar flow in the same direction.

For safety, the calculation is based on the most unfavourable condition, that is, it is assumed that the ambient air is saturated. On the other hand, the air leaving a dryer is generally not saturated; it is assumed that it has absorbed only:

(a) In the case of countercurrent flow: two-thirds of the quantity of water which it could have absorbed if it had left in a saturated condition

(b) In other cases: one-third only of that quantity.

We have then, in the case of countercurrent circulation:

$$A = \frac{1,000 Ph}{\frac{2}{3} (H_1 - H_0)} = \frac{1,500 Ph}{H_1 - H_0} \quad (36.4)$$

A = weight of air to be passed through the dryer, in kg/h (lb./h)

p = weight of sugar to be dried, in kg/h (lb./h)

h = moisture content of sugar relative to unity (0.01, for example)

H_0 = weight of water vapour contained in saturated air at the temperature t_0 of entry of air to the heater (ambient temperature), in kg/1,000 kg (lb./1,000 lb.)

H_1 = weight of water vapour contained in saturated air at the temperature t_1 of exit from the dryer, in kg/1,000 kg (lb./1,000 lb.).

In the case of circulation other than countercurrent, the coefficient 1,500 in eqn. (36.4) should be replaced by 3,000.

H_1 and H_0 are taken from the graph of (Fig. 36.3).

Knowing the weight of air, we may, if required, calculate from it the volume:

$$V = \frac{A}{a_0 + e_0} \quad (36.5)$$

V = volume of air required, in m³/h (cu.ft./h)

a_0 = density of air at t_0 degrees, in kg/m³ (lb./cu.ft.)

e_0 = weight of vapour contained in saturated air at t_0 degrees, in kg/m³ (lb./cu.ft.).

Figures 36.3 and 36.4 give graphs of e and a respectively as functions of temperature.

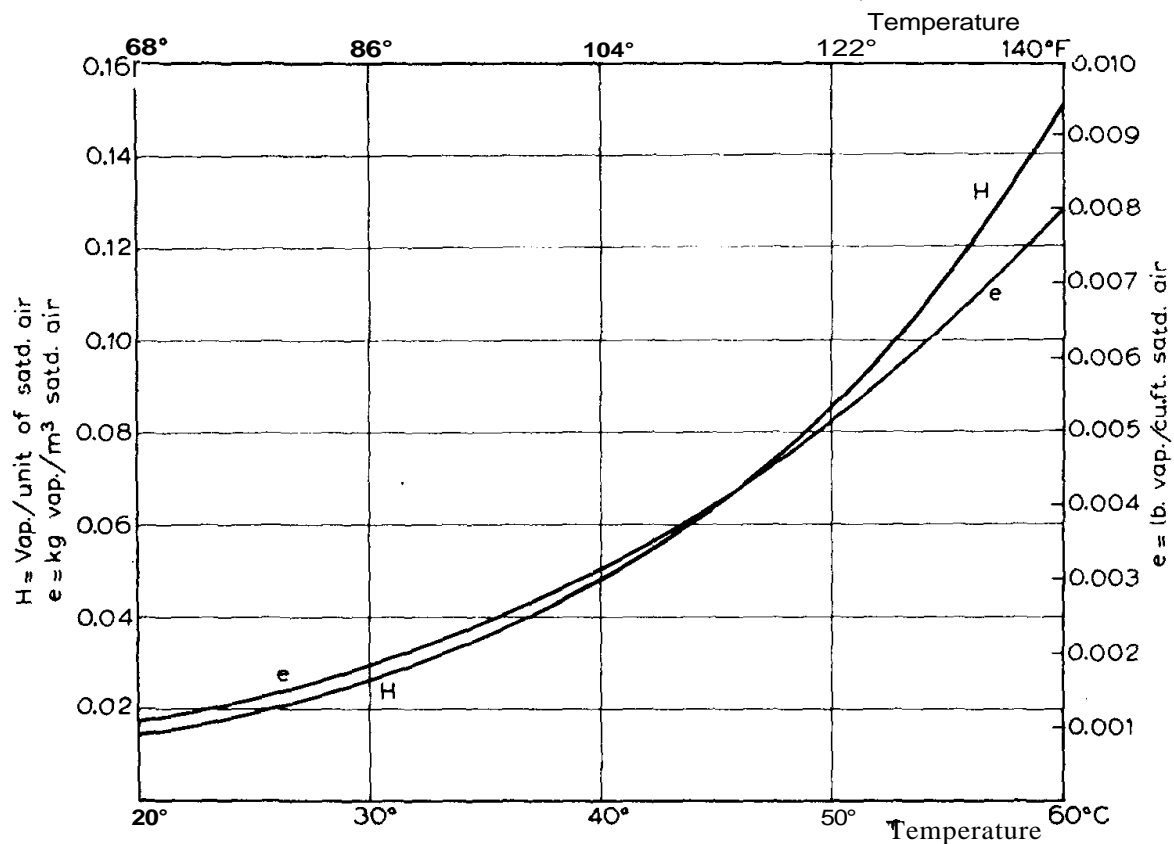


Fig. 36.3. Weight of water vapour in saturated air (per unit weight and per unit volume).

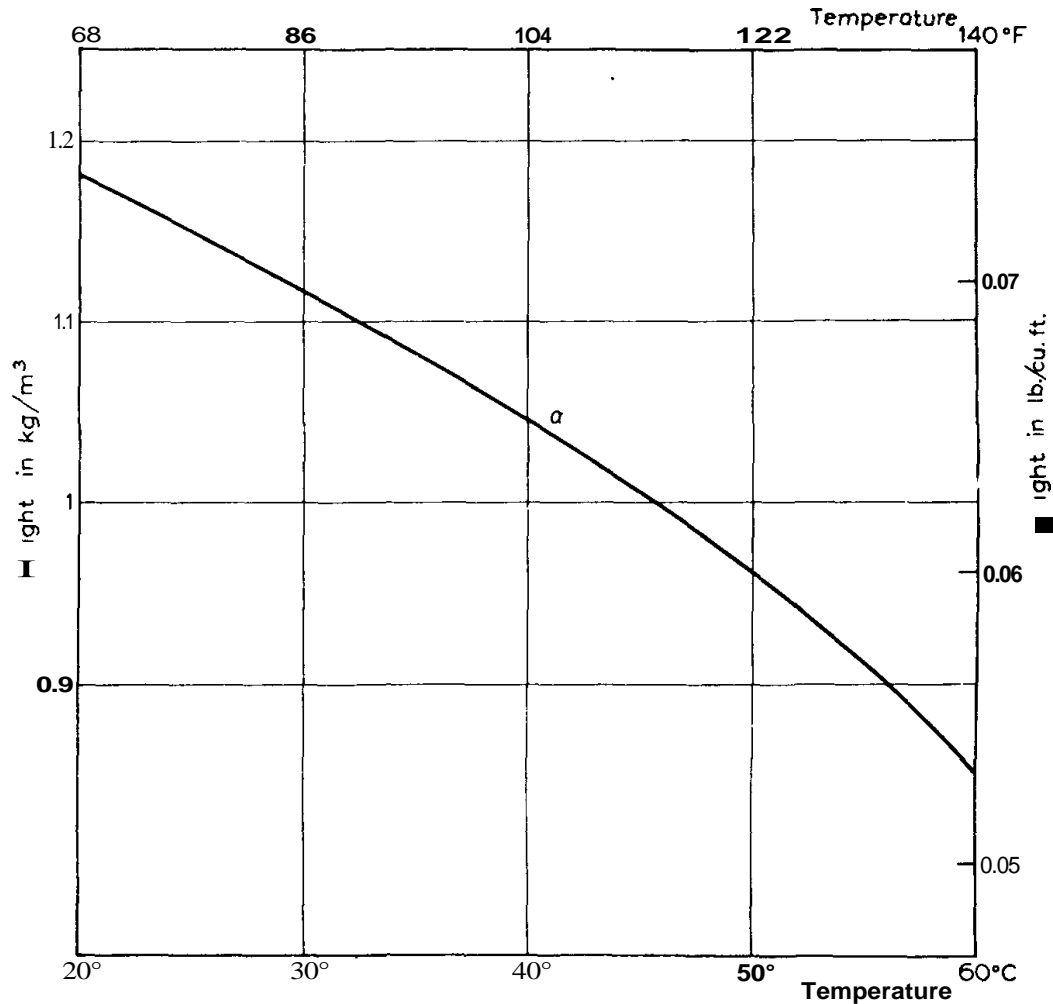


Fig. 36.4. Weight of dry air per unit volume of saturated air.

(B) *Heating surface of the air heater.* The air heater generally consists of tubes, with or without fins, in cast iron or in aluminium, which are supplied with steam at about 5 kg/cm² (70 p.s.i.). The condensate leaves at the bottom.

The heating surface of this small heat exchanger will be given by:

$$S = \frac{M}{k \left(T - \frac{t_1 + t_0}{2} \right)} \quad (36.6)$$

S = heating surface of the air heater, in m² (sq.ft.)

M = quantity of heat to be transmitted, in kcal/h (B.T.U./h)

k = heat-transfer coefficient, in kcal/m²/°C/h (B.T.U./sq.ft./°F/h)

T = temperature of the saturated steam employed, in °C (°F)

t_0 = temperature of cold air entering, in °C (°F)

t_1 = temperature of hot air leaving, in °C (°F).

We shall take:

k = 10 kcal/m²/°C/h (2 B.T.U./sq.ft./°F/h).

The quantity of heat M to be transferred is calculated from the fact that it consists of three principal terms:

(7) The heat necessary to heat the weight A of air:

$$q_1 = Ac (t_1 - t_0) \quad (36.7)$$

c = specific heat of the air = approximately 0.24.

(2) The heat necessary to evaporate the water contained in the sugar (this is generally the most important term):

$$q_2 = Ph [607 + 0.3 t_1 - t_0] \quad (36.8)$$

$$(q_2 = Ph [1,093 + 0.3 (t_1 - 32) - (t_0 - 32)])$$

This quantity q_2 of heat corresponds to that given up by the hot air between T' reached at the outlet from the air heater and t_1 , and which has not been included in q_1 .

(3) The heat necessary to heat the vapour contained in the weight A of air, assumed saturated (this is generally a very small term):

$$(q_3 = AH_0c' (t_1 - t_0)) \quad (36.9)$$

c' = specific heat of the vapour = approx. 0.475.

H_0 is expressed as kg of water/kg of air.

Strictly speaking, it would be necessary to add the heat necessary to heat the sugar. However, since the latter is already at a temperature only slightly below that at which it leaves, this term is generally neglected.

There remains the heat lost to the ambient air. This is taken into account at the same time as the preceding small term, by writing:

$$M = 1.25 (q_1 + q_2 + q_3) \quad (36.10)$$

The air should not be heated to more than 95 - 100°C (203 - 212°F), or 110°C (230°F) according to Tromp¹⁶, in order to avoid damage to the sugar by the high temperature. It is arranged generally that the temperature of the air leaving the heater is between 70 and 95°C (158 and 203°F).

The air velocity is of the order of 1 - 5 m/s (3 - 16 ft./s), but it is reported¹⁷ that velocities greater than 1 m/s involve risk of picking up fine sugar crystals. In practice this speed is often taken up to 2 m/s (6.5 ft./s), a limit which should not be exceeded, reckoned on the hot air leaving the dryer.

The steam consumption will be:

$$Q = \frac{M}{r} \quad (36.11)$$

Q = steam consumption for the air heater, in kg/h (lb./h)

r = latent heat of the steam used (approx. 500 kcal/kg (900 B.T.U./lb.)).

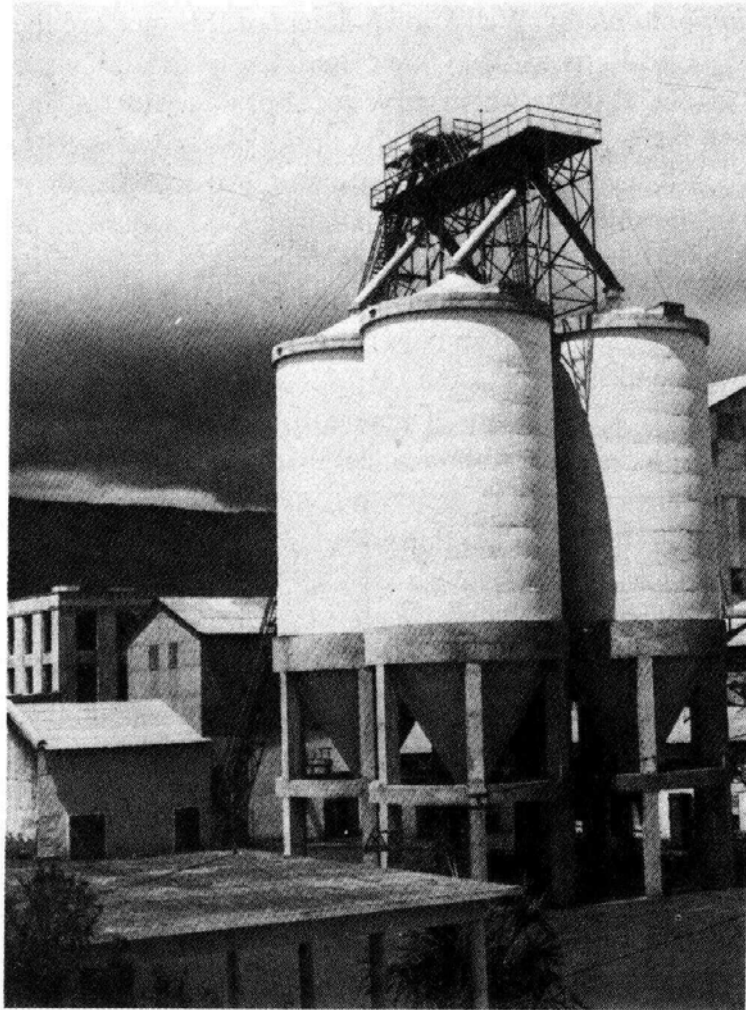


Fig. 36.5. Bulk sugar silo, in four cells.

This steam consumption is generally of the order of 2–3 kg/100 kg of sugar.

The portion of the dryer serving as cooler should reduce the sugar to a temperature of 35–40°C (95–104°F).

Since sugar dust is explosive when it is very fine and mixed with air, it is necessary to prohibit smoking in the sugar-drying room, and it is preferable to install the fan for induced draught rather than forced draught, to avoid risk of blowing this dust into the drying room through leaks in the equipment.

The concentration of powdered sugar in the air above which explosion is possible is 60 g/m³ (0.06 oz./cu.ft.) for powder of 0.1 mm; it falls to 7 g/m³ for very fine powder¹⁸. In Sweden¹⁵ a figure of 17 g/m³ is given, without specifying the fineness.

Types of dryer. There are three principal types of sugar dryer:

- (a) Vertical dryers with superimposed plates
- (b) Fluidised-bed dryers
- (c) Rotating-drum dryers.

Dryers with superimposed plates. Well-known dryers of this type are the French Comessa (Fig. 36.2) and the German Buttner. They are cylindrical in form, contain 20–30 rotating annular plates, with a central space which serves as a passage for the air. These plates are divided into very small sectors, 30 or 40 for the upper plates, 20–30 for the lower ones, separated by slots which themselves are small sectors of 2 or 3°. The sugar deposited on the top plate falls through the slots onto the next plate when it encounters an inclined scraper which acts as a rake. The slots in successive plates are staggered so that the sugar falls only to the next plate and completes almost a full rotation before encountering the next scraper. At a distance ahead of the scraper is an equaliser, which is a scraper set at 20–30 mm (1–1½ in.) above the plates, and which smooths out the hillocks of sugar and gives a uniform layer on the plate. The sugar layer is thus about 25 mm (1 in.) in thickness and its surface is renewed at each drop. From the lowest plate, the sugar is discharged from the dryer by an inclined chute.

An air duct surrounding the plates contains 4 tubular air heaters supplied with steam at 1.5–2 kg/cm² (20–30 p.s.i.). The hot air passes over the plates and is removed by the central chimney; cold air is supplied to the lower plates, the air heaters being limited to the upper three-quarters of the equipment. The interval between the plates is approximately 20 cm (8 in.).

This equipment scarcely damages the crystals, by reason of the low drop from one plate to the next and the fact that only a small proportion of the crystals fall directly on the plate, the remainder falling on to the initial layer of crystals. However, these dryers are bulky, and an accident to a scraper or a plate can mean a serious amount of broken metal.

Fluidised-bed Dryers. The sugar is distributed on a perforated plate; hot air passes through this plate and through the sugar layer. The transmission of heat is very rapid, permitting a very brief time in the dryer. However, the fluidisation and the air flow involve abrasion of the crystals; the equipment also requires a rather high power consumption.

Rotary-drum Dryers. This is the type most widely used. A French make is the Vernon, and the one most widely used is the Roto-Louvre. In general it is cylindrical, and set at a slope of about 5° to the horizontal, which facilitates the movement of the sugar from end to end of the drum, being repeatedly lifted and dropped through the countercurrent air flow. The drum is often arranged to act as both dryer and cooler, the cold air entering at the lower end and leaving near the middle of the drum, where the hot air is introduced. The drum rotates slowly and is fitted with vanes which lift the sugar repeatedly. The drum diameter is 1.5–2 m (5–7 ft.), according to its capacity. The larger diameters have the disadvantage that the repeated fall of the crystals causes abrasion at the corners of the crystals and produces a fine sugar powder which must be collected and recovered; it settles around the dryer, is scattered around the sugar hopper, and even after delivery into storage. To minimise this trouble, B.M.A. divide their dryer so that the crystals fall in a number of small drops, rather than the full diameter of the drum (Fig. 36.6).

This trouble has prompted Fives Cail–Babcock in turn to design a multi-tubular dryer-cooler (Fig. 36.7); this consists of 6 horizontal, parallel dryer tubes surrounding a central support tube of the same diameter, with 6 parallel cooler tubes placed around these six, with

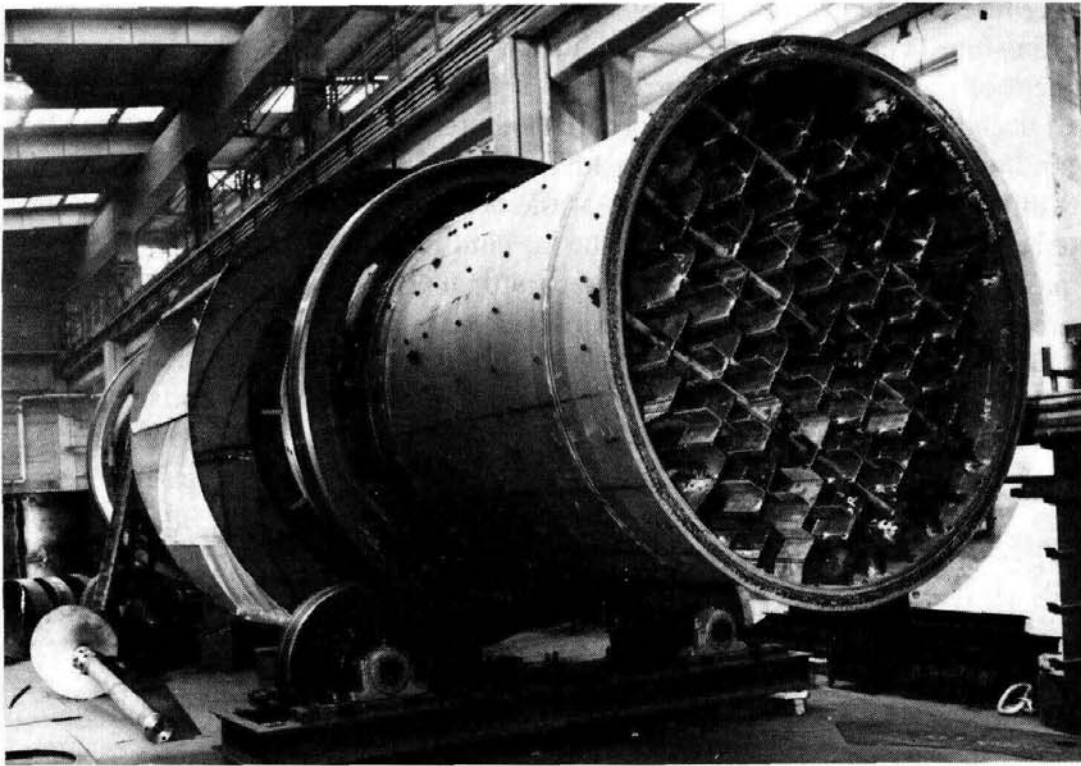


Fig. 36.6. Inner view of a B.M.A. sugar dryer.

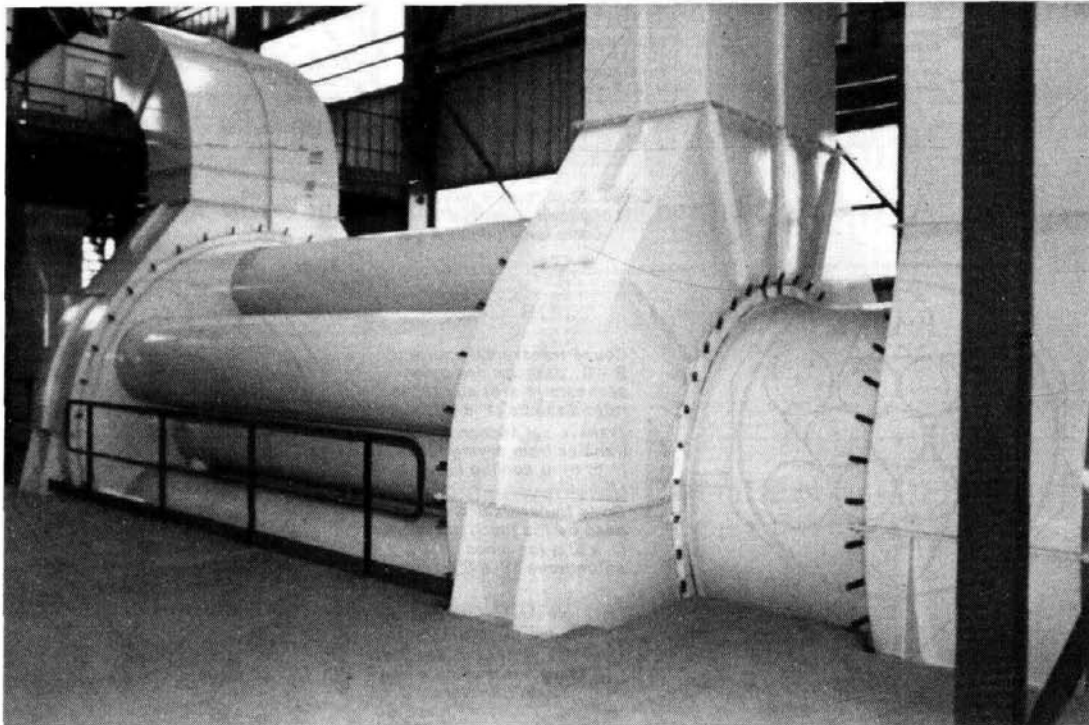


Fig. 36.7. Multitube sugar dryer (Fives Cail - Babcock).

a driving mechanism for rotating the assembly on bearings situated at the two ends of the supporting tube. The twelve tubes are provided with interior baffles which ensure mixing and movement of the sugar, which is introduced at one end of the dryer tubes, and progresses to the far end, where it passes by appropriate chutes into the cooler tubes (Fig. 36.8). The air from the air heater enters the tubes with the sugar and traverses the tubes in parallel current with the sugar. The cold air enters at the delivery end of the cooler tubes, and flows countercurrent to the sugar, leaving by the far end; it mixes with the outgoing hot air, and both pass through a washer or cyclone separator to recover the sugar dust which is then recovered, either dry or in solution, and passed back to the process.

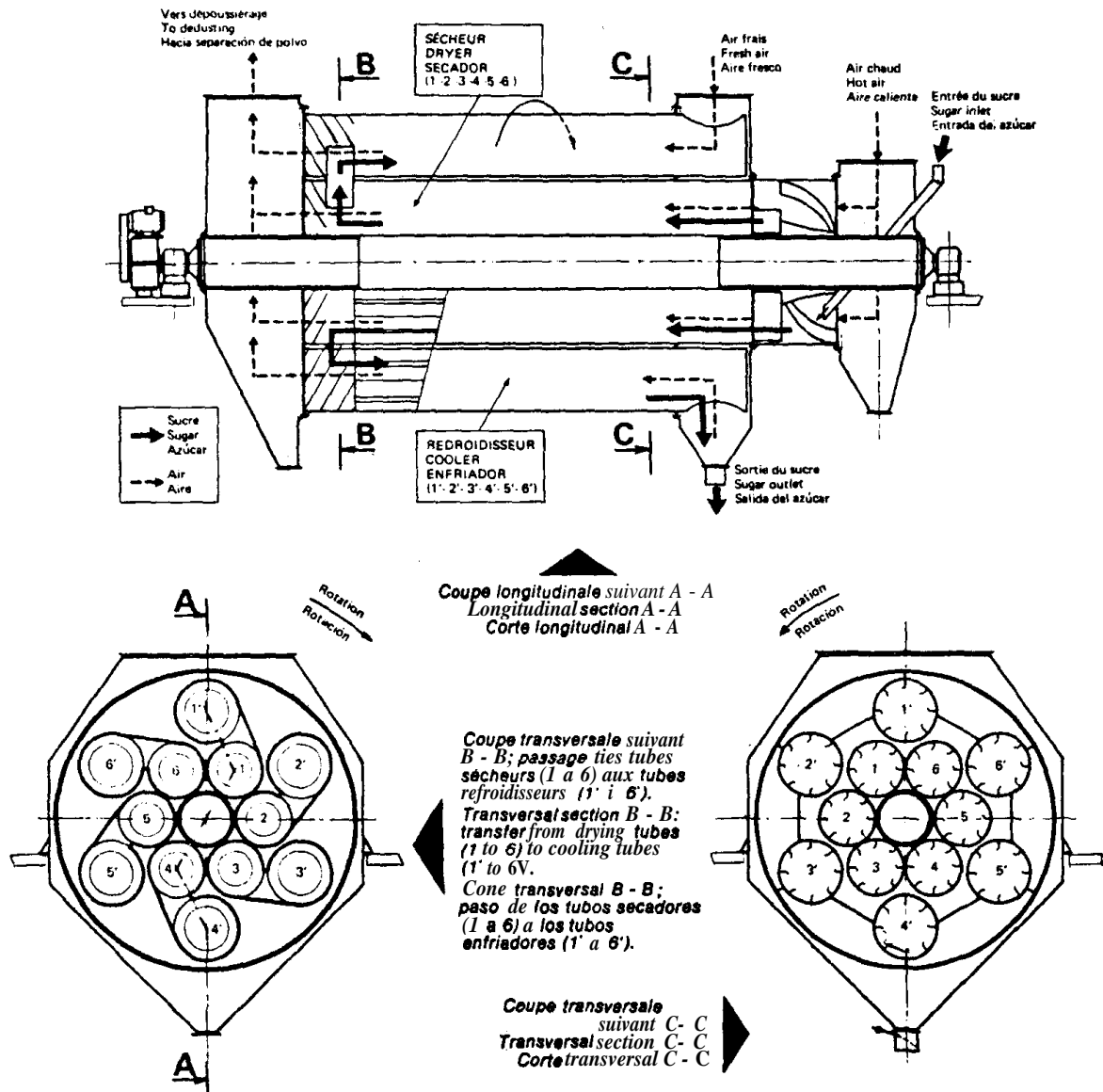


Fig. 36.8. Longitudinal and transversal sections of a multitube dryer (Fives Cail - Babcock).

Power and speed. The speed of rotation of a single-drum dryer is approximately:

$$n = \frac{8-10}{D} \text{ r.p.m.}$$

The speed of the multi-tube is somewhat less. The power consumption of the single-tube type is about 15 kW (20 h.p.), that of the multi-tube about one-third of that figure, since the weight of sugar is much better distributed.

Design of a dryer. We assume the choice of a multi-tube dryer, of Fives Cail-Babcock design (the calculation for a single-drum dryer is much simpler); the dryer is to treat the sugar produced by a factory treating 250 t.c.h., the production of which may reach 30 t/h of sugar leaving the centrifugals at a moisture content of 1%.

The volume of the dryer should be:

$$V = \frac{Qt}{dr} \quad (36.12)$$

V — volume of dryer, in m^3 (cu.ft.)

Q = weight of sugar, in t/h

t = transit time of sugar in the dryer, in h

d = apparent density of loose sugar = 0.80 kg/dm^3 (50 lb./cu.ft.)

r = ratio of filling of the dryer.

For 1% moisture, we accept $t/dr = 1.2 - 2$, generally 1.35. If the moisture content differs from 1% we multiply t by the % moisture (1.5, for example).

We thus have:

$$V = 30 \times 1.35 = 40.5 \text{ m}^3 \quad (1,440 \text{ cu.ft.})$$

This volume must be distributed between 6 tubes, or per tube:

$$v = \frac{40.5}{6} = 6.75 \text{ m}^3 \quad (240 \text{ cu.ft.})$$

The designer proposes tube diameters:

- 711 mm (28 in.) for volume lower than 3.8 m^3 (134 cu.ft.)
- 813 mm (32 in.) for volume of $3.8 - 6 \text{ m}^3$ (134-212 cu.ft.)
- 914 mm (36 in.) for volume of $6 - 8 \text{ m}^3$ (212-282 cu.ft.)
- 1,219 mm (48 in.) for volume more than 12 m^3 (424 cu.ft.)

We take $d = 914 \text{ mm}$ (36 in.). The length L of the tubes will be such that:

$$\frac{\pi d^2}{4} L = 6.75 \quad \text{or:} \quad L = \frac{6.75}{0.656} = 10.3 \text{ m} \quad (34 \text{ ft.})$$

Cooler. For the cooler, we retain the same number of tubes and the same length and cross-sections as for the dryer; with cane, these are sufficient to reduce the sugar to 15°C (27°F) above the ambient temperature, which is the objective generally sought. However, we must check that the air velocities are acceptable. The air heated to 90°C (195°F) drops to about 40°C (105°F) after the dryer. Its density is thus about 1.129 (Table 45.3). The quantity necessary is thus (eqn. (36.4), Fig. 36.3):

$$A = \frac{3,000 \times 30,000 \times 0.01}{49 - 19} = 30,000 \text{ kg/h} \quad (30 \text{ t/h})$$

and its volume is:

$$30,000/1.129 = 26,570 \text{ m}^3 \quad (938,000 \text{ cu.ft.})$$

The cross-section of the dryer is thus:

$$6 \times \frac{\pi \times 0.914^2}{4} = 3.94 \text{ m}^2 \quad (42.4 \text{ sq.ft.})$$

The velocity of hot air leaving is then:

$$\frac{26,570}{3,600 \times 3.94} = 1.87 \text{ m/s} \quad (6.15 \text{ ft./s})$$

which is close to the acceptable limit (2 m/s = 6.5 ft./s) but acceptable, particularly in parallel flow.

At the exit from the cooler, it will be obviously much lower (1.67 m/s = 5.47 ft./s).

Automatic scales

The sugar hopper after the dryer should be preceded or followed by an automatic scale. When the sugar is to be packed into bags, this may be a bagging scale. For bulk sugar, it could be a continuous integrating scale on the belt conveyor, but batch weighing is more accurate and is to be recommended. These must be kept clean, and checked at least twice per day.

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37. Sugar

Density

The true density of the sugar crystal is approximately 1.6 g/cm^3 . In Australia, a figure of 1.58 (98.6 lb./cu.ft.) has been used¹ for commercial sugar. The value for pure sucrose is given as 1.596.

The apparent density of granulated sugar varies considerably according to the form and the regularity of the crystals. It varies generally between 0.8 and 0.9 g/cm^3 (50–55 lb./cu.ft.).

White sugars often go as low as 0.72 (45 lb./cu.ft.)^{1a}. The most frequent densities of raw sugars, or washed raws, are of the order of $0.82 - 0.90$ (51–56 lb./cu.ft.). By vibration during filling, a compaction of 10 - 20% may be obtained, which thus modifies the above values to a corresponding extent.

In a conical pile in bulk, we may generally assume 1% higher in density, for sugar under pressure^{1b}, per 10 m of height of the pile; or 0.7 - 1.8% per 10 m (33 ft.), 1.3 - 3.2% for 20 m (66 ft.). In Hawaii, a figure of 881 kg/m^3 is used.

Solubility of sucrose in water

For a saturated solution we have the following relationships, according to Verhaar²:

$$E = 0.355 (158 - t) \quad (37.1)$$

$$S = \frac{28.162}{158 - t} \quad (37.2)$$

$$s = \frac{10,000}{100 + E} = \frac{10,000}{100 + 0.355 (158 - t)} \quad (37.3)$$

where:

E = water % sucrose
 S = sucrose % water
 s = sucrose % of solution
 t = temperature of the solution, in °C.

} at the condition of saturation

Table 37.1 gives several values of the solubility of sucrose, as determined by Wise and Nicholson³:

TABLE 37.1

SOLUBILITY OF SUCROSE

	<i>Temperature (°C)</i>						
	<i>20 (68°F)</i>	<i>30 (86°F)</i>	<i>40 (104°F)</i>	<i>50 (122°F)</i>	<i>60 (140°F)</i>	<i>70 (158°F)</i>	<i>80 (176°F)</i>
<i>x</i>	66.32	68.20	70.14	72.16	74.24	76.40	78.62
<i>S</i>	196.9	214.5	234.9	259.2	288.2	323.7	367.7

These are the values given by the equation:

$$s = -62.77 + 0.1706 t + 0.000344 t^2 \quad (37.4)$$

Charles⁴ has given slightly different values, corresponding to the equation:

$$s = 64.4 + 0.0725 t + 0.002057 t^2 - 0.000009035 t^3 \quad (37.5)$$

In Poland^{4a} the equation

$$s = 68.415 + 0.09559 t + 0.0004905 t^2$$

has been developed as applicable over the range $89.2^\circ\text{C} \leq t \leq 145.3^\circ\text{C}$.

Contraction

When sugar is dissolved in water, and the solution is diluted, the volume of the resulting solution is always lower than the sum of the volumes of its two components.

In the case of solutions of pure sugar, Paar⁵ states that this contraction is a maximum for a concentration of 57.3%, at which it amounts to 10.3 cm³ per litre (or approximately 1%). For impure solutions the extent of the contraction differs, but is generally greater than for pure solutions.

Filterability

The filterability of commercial raw sugar is an important quality to the refiner. The term refers to the ease or otherwise of filtering in the refinery the liquors produced by dissolving the raw sugars being treated.

The filterability of a given sugar depends on its impurities, and particularly on the following components:

Starch

Waxes

Gums

Iron and other phosphates

Silica

Certain elements such as Ca, Mg, Al, which are important particularly due to the fact that they increase viscosity.

The sugar manufacturer has little means at his disposal for altering these factors, which depend on the nature of the cane rather than on factory operation. One factor by which they can be varied is the washing of the filter cake, after the actual filtration: by restricting this washing and avoiding excess, there is less risk of carrying the waxes on to the manufacture, as these materials are very unfavourable. On the other hand, it is difficult to eliminate starch during the defecation; starch passes through and enters the sugar crystal and is found practically integral with the finished sugar.

Cane juices may contain as much as 460 mg of starch per litre. A fair proportion of this, however, can be eliminated by centrifuging⁶. It has also been suggested in Australia that starch can be eliminated by the action of the natural enzymes in the juice, by maintaining the juice at 70°C (158°F) for a certain time, 10–30 minutes. This procedure has been applied in South Africa. The starch is then depolymerised, while the temperature does not permit the formation of dextran.

In Australia, Briggs⁷ has found a close correlation between filterability of the sugar produced and the pH of the clarified juice; these two quantities vary in the same direction: when the pH dropped from 7.0 to 6.6, the filterability fell from 50 to 36, while an exceptional pH of 6.2 gave a filterability of about 20.

The impurities of the crystal are often found in the centre, when a low-grade sugar such as C sugar has been used as footing for the crystal. Further, the impurities increase from the centre towards the periphery, which is normal, since the purity of the liquor decreases in the pan as the crystal grows. Hence there are two precautions to obtain a sugar of better filterability:

(a) It is necessary to remelt the C sugar (and if necessary the B also) instead of using it as footing for A and B sugars

(b) It is desirable to boil a small grain in order to avoid peripheral impurities: Payne⁸ recommends 0.2 mm for C sugar and 0.6 for A sugar.

Inclusions in the crystal generally occupy 0.2% of its volume⁹, sometimes as much as 0.4%.

Titre

In French countries, the value of the sugar is fixed proportional to its titre, also called "rendement 2—4". This is a formula proposed by Girard in 1876:

$$\text{Titre} = \text{pol} - (2r + 4c) \quad (37.6)$$

r = reducing sugars % on sugar

c = ash % sugar.

The refinery moreover deducts 1% of the titre obtained, before multiplying by the price unit of titre. This 1% is considered to represent the losses in refining other than those caused by reducing sugars and ash.

In Australia, the "net titre", a very analogous figure, is used:

$$\text{Net titre} = \text{pol} - (r + 5c) \quad (37.7)$$

The E.E.C. has accepted an approximate simplification:

$$\text{Titre} = 2P - 100$$

P = pol.

Regularity of grain. The regularity of crystal size is expressed by the coefficient of variation (c.v.). It is given by 100 times the difference between the sizes of screens which retain 16 and 84% of the sample, divided by twice the mean opening (m.a. = mean aperture), that is, the size of the mesh which retains 50% of the sample. A good grain^{1°} has a c.v. of 20 - 30; a poor one, a figure above 40.

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38. Molasses

Density

The true density of molasses is generally of the order of 1.4 – 1.5; but it normally contains fine air bubbles entrained during the fuggalling process and also picked up by friction every time the molasses is discharged in a fine stream into a tank. These air bubbles escape only very slowly, particularly when they are very fine, and it is necessary to take them into account in any figure for density.

In Jamaica¹, it is generally estimated that the included air represents 10 – 15% of the volume of the molasses. The volume of air moreover is greater when the molasses has been handled repeatedly. We should take preferably:

10% for light molasses
15% for heavy molasses

In Antigua², it was customary to subtract 5% of the theoretical weight, but a careful check has indicated that it would sometimes be necessary to allow 17% or more.

It is therefore impossible to have a precise control on a volume basis. This can only be obtained by weighing the molasses.

Several good types of automatic scale are available for weighing molasses. Such a scale requires an automatic correction for tare, to take into account the film of molasses which remains adhering to the tank when it is emptied.

Viscosity

Viscosity is a most important property, since the viscosity which can be handled is the limiting factor in the practical exhaustibility of molasses.

The viscosity of molasses increases rapidly as its temperature decreases. It is estimated³ that the viscosity at temperatures of 50, 45, 40°C is respectively proportional to the figures 1, 2, 3. In other words, the viscosity of the molasses is *approximately* three times as high at 40°C (140°F) as it is at 50°C (122°F).

Viscosity increases' equally rapidly with the brix.

Viscosity also increases with the proportion of air entrapped in the form of fine bubbles in the molasses. For example⁴, with 10% and 20% by volume of entrapped air, the viscosity may be respectively 15% and 50% higher than it would be in the absence of included air. It is thus advisable to avoid discharge of final or other molasses into a tank in a fine stream, as this discharge entrains air at the contact of this fine stream with the liquid in the tank. Molasses should flow by an inclined gutter extending to the bottom of the tank which is to be filled.

Storage

Stored between 30 and 35°C (86 - 95°F), molasses would lose 2 - 3% per year of its fermentable sugars⁵.

If the temperature of storage is increased by 10°C (18°F), this loss would be quadrupled.

Molasses stored in tanks has in several cases undergone explosive decomposition with sudden liberation of heat. The causes of this phenomenon have never been fully unravelled, but it has always followed a more or less lengthy period of heating to 40°C (104°F) or higher. It is thus necessary to watch the molasses in storage to ensure that its temperature does not attain that limit.

Quantity of molasses in manufacture

It is useful to know the quantity of molasses to be obtained from the massecuite and syrups in stock at the end of the week, in order to obtain an accurate figure for sugar lost in molasses.

In each of these products we may calculate the quantity of sugar to be obtained, given by (cf. p. 839):

$$Q = \frac{s(j - m)}{j(s - m)}$$

The corresponding quantity of molasses will thus be:

$$M = 1 - \frac{s(j - m)}{j(s - m)} \quad (38.1)$$

and, equating sugar with sucrose:

$$M = 1 - \frac{j - m}{j(s - m)} = \frac{m(1 - j)}{j(1 - m)} \quad (38.2)$$

Assuming $m = 40$, we have, for example, for the several massecuites:

$$\text{A massecuite, purity} = 85: M = \frac{40(100 - j)}{60j} = \frac{2}{3} \frac{100 - j}{j} = \frac{2 \times 15}{3 \times 85} = 12\%$$

$$\text{B massecuite, purity} = 70: M = \frac{2 \times 30}{5 \times 70} = 28.6\%$$

$$\text{C massecuite, purity} = 58: M = \frac{2 \times 42}{3 \times 58} = 48\%$$

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39. Boiling-House Control

Recovery

The term "boiling house" designates the whole of the factory except the milling station, that is to say, the total of clarification, evaporation, pan station, crystallisers and centrifugals.

Control of the milling plant has been already discussed in Chap. 21, in which the main figure discussed was the extraction, defined as:

$$\text{Extraction} = e = \frac{\text{sucrose in mixed juice \% cane}}{\text{sucrose entering the mills \% cane}}$$

The control of the boiling house commences at the point where control of the milling plant finishes, that is, at the mixed juice.

The efficiency of the boiling house is defined in a manner analogous to that of the mill plant. The term "boiling-house recovery" indicates the ratio of the sucrose obtained in the sugar manufactured to that entering in the mixed juice.

$$\text{Boiling-house recovery} = r = \frac{\text{sucrose obtained in sugar manufactured \% cane}}{\text{sucrose in mixed juice \% cane}}$$

"Overall recovery" is the total efficiency of the factory:

$$\text{Overall recovery} = er = \frac{\text{sucrose obtained in sugar manufactured \% cane}}{\text{sucrose entering the mills \% cane}}$$

Returning to the boiling house, it is seen that the recovery takes into account the following losses:

Loss of sugar in filter cake

Loss of sugar in molasses

Undetermined losses (leakages, entrainment, inversion, etc.).

Noel Deerr's SJM formula

Starting from a primary material, which may be clarified juice, syrup, or massecuite, of purity j , we recover from this, sugar of purity s , in the ratio of Q per unit weight of dry substance in the primary material under consideration; molasses of purity m is also obtained. We shall equate the weight of sucrose present in the primary material to the weight of sucrose recovered in the sugar and molasses obtained. Expressing all quantities in terms of unit weight of dry substance in the primary material, we have:

$$1 \times j = Qs + (1 - Q)m$$

Hence:

$$G = \frac{i - m}{s - m}$$

or:

$$\frac{Qs}{j} = \frac{s(j - m)}{j(s - m)}$$

Now:

Qs = sucrose contained in the sugar obtained

j = sucrose contained in the primary material.

Then:

Qs/j = sucrose which should be extracted from the juice under consideration for unit sucrose contained in that juice, if there is no loss of sucrose during the operation.

In other words, we shall have:

$$e = \frac{s(j - m)}{j(s - m)} \quad (39.1)$$

e = recovery from this operation.

Comments. (1) This formula assumes that true purities are employed. Deerr comments that it may be considered accurate if s , j , and m are expressed in apparent purities, or any other purity, on condition that the three purities are expressed in the same manner. However, this is not accurate, especially for apparent purities; hence it should preferably be used with true purities, or with gravity purities if these are not available.

(2) In the Philippines it has been commented that, in regard to payment of growers on recoverable sugar (and also for comparison for cane varieties), it would be more accurate, instead of using for all cane a mean or fixed value of m , to replace it by a value:

$$m' = (100 - J) 0.4 + K \quad (39.2)$$

J = purity of first expressed juice of the cane analysed

K = a constant, for example, 28.57 (or other suitable local value).

This comment appears logical.

For example, with $K = 33$:

$$(a) J = 88, m' = (0.4 \times 12) + 33 = 37.8$$

$$(b) J = 82, m' = (0.4 \times 18) + 33 = 40.2$$

(3) The value of Q above represents the dry substance in sugar % dry substance in primary material (syrup, for example). We have then:

$$\text{Dry substance in sugar \% dry substance in syrup} = \frac{j - m}{s - m} \quad (39.3)$$

Example.

$$\begin{aligned} j &= 85 \\ s &= 99 \\ m &= 40 \end{aligned}$$

$$\frac{j - m}{s - m} = \frac{85 - 40}{99 - 40} = \frac{45}{59} = 76.27\%$$

Hence dry substance in molasses = 23.73% of dry substance in syrup, and the yield of sugar from a massecuite:

$$e = \frac{j - m}{s - m} B \quad (39.4)$$

B - brix of massecuite.

Winter–Carp formula

Formula (39.1) assumes that there exists an ideal purity of molasses, m_0 , which would represent a perfect performance in the boiling house. The ideal boiling-house recovery is then:

$$e_0 = \frac{s(j - m_0)}{j(s - m_0)}$$

In Java, a value of 28.57 had been adopted for this ideal molasses purity. The sucrose recoverable in percentage then becomes:

$$e_0 = 100 \frac{100(j - 28.57)}{j(100 - 28.57)} = 100 \left(1.4 - \frac{40}{j} \right) \quad (39.5)$$

This is the simple Winter–Carp formula.

We may comment that if we use a standard value of 85 for j in this formula, it becomes:

$$e_0^s = \frac{100(85 - 28.57)}{85(100 - 28.57)} = 93\%$$

Influence of molasses purity on recovery

It is interesting to know what gain could be expected by an improvement (that is to say, a decrease) of 1 point in the purity of the molasses, or inversely what loss would be suffered by an increase of 1 point in the molasses purity.

In eqn. (39.1), substituting $s = 1$, we have:

$$\frac{d\varrho}{dm} = - \frac{1 - j}{j (1 - m)^2} \quad (39.6)$$

Taking for j the standard value of 85, we have:

$$\frac{d\varrho}{dm} = - \frac{0.1765}{(1 - m)^2}$$

$$\text{For } m = 40: \quad d\varrho/dm = - 0.49$$

$$\text{For } m = 30: \quad d\varrho/dm = - 0.36$$

In other words, one point in molasses purity (passing from 36 to 35, for example, or conversely) corresponds to a gain or a loss of about 0.36-0.50% in the recovery, or 0.34–0.46% of the sugar made; say, of the order of 0.4%.

Inversion losses

In the course of the crystallisation process in the factory, that is, between the syrup and the finished sugar, there are inevitably losses by inversion. These depend mainly on the pH and the temperature of the mother liquor. The pH of the massecuites, for example, is of the order of:

A massecuite: 5.8 – 6.8

B massecuite: 5.6-6.5

C massecuite: 5.5 — 6.4

Honig¹ considers that these losses, by the nature of the crystallisation processes themselves, cannot be less than 0.4% of the sucrose present in the syrup. This is an inevitable loss, which is included in the "undetermined losses".

C massecuite per tonne non-sucrose. It appears that the quantity of C massecuite per tonne of non-sugar in mixed juice should be approximately 1,650 l (non-sucrose being calculated as refractometric brix minus sucrose).

Theoretical recovery

It is interesting to compare the actual recovery obtained by the factory with the theoretical recovery which it should be able to obtain. In order to calculate this, losses in filter cake and undetermined losses are assumed as zero, since they are low and could theoretically be reduced to zero. On the other hand, the loss in molasses is substantial (it is generally the highest of the four losses) and unavoidable. Since the purity of the primary material, the mixed juice (say j) is known, and the purities of the two final materials, sugar (5) and molasses (m) are also known, the theoretical recovery is immediately obtained by the formula of Deerr:

$$r_1 = \frac{s(j - n)}{j(s - m)} \times 100 \quad (39.7)$$

If the control is well conducted and the weights are accurate, the theoretical recovery r_1 will obviously be greater than the actual recovery r . The difference between these two figures expresses the losses incurred apart from that in molasses.

Basic recovery

It is, however, equally valuable to compare the loss occurring in molasses with that which would have occurred if it had been possible to pursue the exhaustion of molasses to the lowest purity figure obtainable. To express this we assume a molasses purity m of 28.57 and a sugar of purity $s = 100$, which leads us to the *Winter - Carp* formula:

$$r_b = 100 \left(1.4 - \frac{40}{j} \right) \quad (39.8)$$

j = gravity purity of the mixed juice.

Equivalent Standard Granulated

However, two factories making two qualities of sugar of different polarisations would not have the same efficiency, even in terms of sucrose, since it is much easier to attain a given exhaustion in raw sugar than in refined sugar, for example. Deerr has accordingly created the idea of "Equivalent Standard Granulated" (E.S.G.). This is the quantity of sucrose which could theoretically be obtained from a sugar of purity s , assuming always an ideal purity of final molasses of 28.57. It thus has the value:

$$G = 100 \left(1.4 - \frac{40}{s} \right) \quad (39.9)$$

G = E.S.G.

s = purity of the raw sugar made.

If $s = 99$, for example, $G = 99.60$.

By substituting the E.S.G. for the sugar actually made, or, in other words, by multiplying the quantity of sugar made by G , factories making sugars of different quality are thus put on an equal footing.

If a factory making sugar of purity 99 has a yield of 12, its yield in E.S.G. will therefore be:

$$12 \times 0.9960 = 11.952$$

Recovery E.S.G.

Hence we have the concept of "recovery E.S.G.", which has the object of eliminating the disturbing influence of the sugar quality, and does this by replacing the yield of sucrose in the numerator of the expression for recovery r , by the yield in E.S.G.:

$$r_g = r \times G$$

Reduced recovery E.S.G.

When a factory is handling juice of higher purity, it obtains a higher recovery, as the formula of Deerr shows immediately. The effect of juice purity on recovery is analogous to that of fibre on extraction. The figure "reduced recovery" has accordingly been established, analogous to reduced extraction, choosing as standard purity the Clerget figure of $j = 85$. This has been applied as follows:

To avoid the complication of handling the three losses in the boiling house, the loss in filter cake and undetermined losses have been grouped with the molasses loss, by calculating a fictitious molasses purity under the term "virtual purity" (v), which itself would give the recovery E.S.G. obtained (r_g) if the other losses were nil. In this case we would have:

$$r_g = \frac{100(j - v)}{j(100 - v)} \times 100 \quad (39.10)$$

whence:

$$v = \frac{100j(100 - r_g)}{10,000 - jr_g} \quad (39.11)$$

The recovery which would have been obtained with this virtual molasses purity v and a juice purity at the standard value of 85 would be:

$$r_{85ND} = 100 \times \frac{100(85 - v)}{85(100 - v)} \quad (39.12)$$

or, replacing v by its value (eqn. (39.11)):

$$r_{85ND} = 100 - \frac{3j(100 - r_g)}{17(100 - j)} \quad (39.13)$$

This is the expression which the ISSCT had adopted as international comparison for recovery figures.

Comments. The concept of virtual purity is not useful for deriving this formula (39.13). It is simpler to work as follows: Let:

r_j = recovery of cane obtained with purity j

r_{85} = recovery of cane obtained with purity 85.

In order that these two recovery figures should agree, we write that they should give the same yield of sucrose lost % non-sucrose in mixed juice:

$$\frac{QBj(1 - r)}{QB(1 - j)} = \frac{j(1 - r)}{1 - j}$$

Q = weight of mixed juice % cane

B = brix of mixed juice

j = purity of mixed juice.

We have then:

$$\frac{j(1 - r_j)}{1 - j} = \frac{85(1 - r_{85})}{15}$$

Hence:

$$r_{85} = 1 - \frac{3j(1 - r_j)}{17(1 - j)}$$

Reduced recovery E.S.G. Gundu Rao

The formula of Noel Deerr has been criticized because the correction takes into consideration all clarification and boiling house losses, while the loss in final molasses is the only one influenced by the juice purity. Gundu Rao, desirous of correcting this deficiency, separated the loss in final molasses from the other losses and stated that only the former had to be corrected and brought down to the value it should have been if the juice purity had been 85.

Let r_{85GR} be the reduced recovery so required. The total boiling house losses are:

$$T = 100 - r_g$$

r_g = actual boiling house recovery, ESG.

They include:

(a) The loss in final molasses:

$$a = 100 - 100 \frac{100(J - M)}{J(100 - M)}$$

(b) The other losses (in filter cake, undetermined etc.):

$$b = T - a = 100 - r_g - \left(100 - 100 \frac{100(J - M)}{J(100 - M)} \right)$$

We will reduce the first formula to the value a' which it would have had if the juice purity had been 85. We have then:

$$r_{85GR} = 100 - (b + a')$$

$$r_{85GR} = 100 - \left[100 - r_g - \left(100 - 100 \frac{100(J - M)}{J(100 - M)} \right) + 100 - 100 \frac{100(85 - M)}{85(100 - M)} \right]$$

whence:

$$r_{85GR} = r_g - \frac{100}{J} \frac{100 (J - M)}{(100 - M)} + \frac{100}{85} \frac{100 (85 - M)}{(100 - M)}$$

$$r_{85GR} \sim r_g + \frac{100 M}{100 - M} \cdot \frac{85 - J}{100} \quad (4)$$

The Reduced Recovery ESG Gundu Rao is an improvement on the Reduced Recovery ESG Noel Deerr since it takes into account only the final molasses loss, which is the only loss influenced by the juice purity. It has been criticized, however, for taking into account the whole of the non-sucrose contained in the juice (100 - J), while only the non-sucrose remaining in the molasses should be considered. Another shortcoming proceeds from using the gravity purity M of these molasses which is very different from the true purity M'.

Let us try to find the proportion of non-sucrose left in the final molasses. The respective amounts are:

$$\text{In final molasses: } NS_m = yB_m (100 - M)$$

$$\text{In juice: } NS_j = qB_j (100 - J)$$

On account of the difference between the refractometric Brix B_m and the actual dry solids DS_m of the molasses, it is advisable to replace B_m by DS_m . If DS_m has not been determined, one can merely use the close approximation $DS_m \approx 0.96 B_m$. Similarly the molasses gravity purity M should be substituted for the true purity M', or, if M' is unknown, M should be multiplied by 1.2: $M' = 1.2M$. The proportion of non-sucrose is then:

$$\text{Accurate: } \lambda = \frac{yDS_m (100 - M')}{qB_j (100 - J)} \approx \frac{0.96yB_m (100 - 1.2M)}{qB_j (100 - J)}$$

- X = ratio of final molasses non-sucrose to mixed juice non-sucrose
- y = weight of final molasses % cane
- q = weight of mixed juice % cane
- B_m = refractometric Brix of final molasses
- DS_m = dry solids % final molasses
- B_j = refractometric Brix of mixed juice
- M = gravity purity of final molasses
- M' = true purity of final molasses
- J = gravity purity of mixed juice.

Whence:

$$\begin{aligned} r_{85GR} &= r_g + \frac{100M'}{100 - M} \cdot \frac{85 - J}{0.85J} \cdot \frac{yDS_m(100 - M')}{qB_j(100 - J)} = \\ &= r_g + \frac{100M'(85 - J)yDS_m}{0.85J(100 - J)qB_j} \end{aligned} \quad (39.15)$$

$$\begin{aligned} r_{85GR} - r_g &= \frac{120M}{100 - 1.2M} \cdot \frac{85 - J}{0.85J} \cdot \frac{0.96yB_m(100 - 1.2M)}{qB_j(100 - J)} = \\ &\sim r_g + \frac{115M(85 - J)yB_m}{0.85J(100 - J)qB_j} \end{aligned} \quad (39.16)$$

As this is the boiling house efficiency which must be assessed, we must start from the mixed juice and not from the clear juice, since the purity and dry solids of the latter have already benefited from the quality of the clarification.

ISSCT rules imply the factor 100 instead of 120 in formula (39.16). The factor 115 is an approximation, but leads to a more realistic result.

Example. Let: $r_g = 90$, $q = 100$, $y = 3$, $M = 38$, $B_m = 85$, $DS_m = 81.6$, $l = 88$, $B_j = 14$, $M' = 45$.

$$r_{85GR} = 90 - \frac{100 \times 45 (88 - 85) \cdot 3 \times 81.6}{0.85 \times 88 \times 12 \cdot 100 \times 14} = 90 - 2.63 = 87.37$$

$$r_{85GR} = 90 - \frac{115 \times 38 (88 - 85) \cdot 3 \times 81.6}{0.85 \times 88 \times 12 \cdot 100 \times 14} = 90 - 2.66 = 87.34$$

These figures show the error which would have been made in the correction if the gravity purity had been used instead of the true purity, since the 115 factor would be replaced by 100.

The non-sucrose fraction for the final molasses is around 0.8 in ordinary defecation. With sulphitation and especially carbonatation, it would come down to 0.65, 0.60 or even 0.50^{2a}

General reduced recovery, E.S.G.

This has the value:

$$R = e_{12.5} \times r_{85GR} \quad (39.17)$$

This expression takes into account all the factors which are liable to affect the true overall recovery of a sugar factory: fibre in cane, purity of juice, quality of sugar. The only item which it does not take into account is the particular composition of the impurities, and this must be taken into consideration as qualifying the remark just made.

Boiling-house performance

This figure as a criterion of efficiency is obtained by expressing recovery E.S.G. (eqn. (39.10)) as a fraction of basic recovery (39.8). The latter figure depends on the purity of juice, and thus on the proportion of non-sugars. The boiling-house performance has been criticised in South Africa² as not taking into account the composition of this non-sugar, which can have a marked influence on the boiling-house result, and that country has accordingly decided to discontinue use of the expression.

Non-sucrose

In addition to checking the sucrose throughout the manufacture, it is interesting to follow the non-sugar, from the quantity introduced with the mixed juice.

In South Africa, for example, Douwes Dekker has introduced accounting of the non-sugar in the mutual control of the various factories and draws interesting conclusions from it.³

Non-sucrose is defined as the difference between the brix and the sucrose (or the pol), per 100 of brix.

During the manufacture, the non-sucrose undergoes the following changes:

(a) Part is eliminated in the clarification.

(b) During the clarification, on the other hand, an excess of lime or other material introduced for treating the juice or the syrup leads to an increase in the proportion of non-sugar.

(c) During the clarification and the concentration, inversion tends to increase the non-sucrose at the expense of the sucrose. However, due to excess alkalinity, particularly during the pre-liming, a certain proportion of reducing sugars (and hence of non-sucrose) may be destroyed, and part of this may moreover be eliminated.

If finally the quantity of non-sucrose leaving the factory in sugar and molasses is assessed, Douwes Dekker considers that it would normally represent, in South Africa, 81% of the non-sucrose present in mixed juice. A lower value would generally indicate effective control of manufacture. A higher figure would indicate excess lime, inversion, and, particularly, excess recirculation or remelting in the boiling system adopted.

Comparison of this ratio with losses in filter cake, molasses, and undetermined losses may allow one to detect whether bad work in the boiling house is due to inversion, or to a too high purity of the molasses. It may be considered that the losses in molasses are proportional to this ratio. By reducing this ratio from 90 to 81%, for example, a loss of 9 in molasses may be reduced to 8.1.

These values correspond to gravimetric brix; they would be different if the brix were determined by refractometer.

It is found in South Africa⁴ that, considering the ratio of total non-sugars in molasses and sugars to non-sugars in clarified juice, and considering only those factories utilising refractometric brix, the ratio is approximately 0.83.

Assessment of non-sucrose leaving is particularly useful when the weight of molasses is known accurately. If the weight of molasses for example is too high, an exaggerated figure will be found for non-sucrose leaving; from this an error in the value of undetermined losses may be detected: if these are low, it is because the figure for weight of molasses is too high. We have here also a means of seeing whether excessive undetermined losses are due to entrainment or to inversion.

Brix balance

In the same way, a brix balance may be made, from mixed juice to sugar and molasses.

Reducing-sugar balance

It is useful to work out a reducing-sugar balance, as for non-sucrose and brix balances.

If the reducing sugars in mixed juice are taken as *equal* to 100, Douwes Dekker⁵ considers that, since some is always decomposed in the course of lime addition, there will remain 90 - 95 in the clarified juice. On the other hand, some reducing sugars are formed during the evaporation, and the figure will thus increase to 92 or 97. During the sugar boiling, reducing sugars are both formed and destroyed. Finally, the quantity of reducing sugars in molasses is generally higher than in clarified juice, but Douwes Dekker considers that it should not be higher than in mixed juice.

High undetermined losses, combined with high figures for non-sucrose leaving and for reducing sugars leaving, indicate high losses by inversion.

However, since 95 parts of sucrose give 100 of invert sugar, inversion increases the undetermined losses of sucrose but decreases the undetermined losses of brix.

Standard control system for the sugar factory

We give below the method of *calculation* for factory control according to the recommendations of the International Society of Sugar Cane Technologists.

Data

	<i>Brix</i>	<i>Pol</i>	<i>Sucrose</i>	<i>Apparent purity</i>	<i>Gravity purity</i>	<i>Moisture</i>	<i>Suspended material</i>
Primary juice		S_1	t'				
2nd-mill juice		S_2					
Residual juice				p			
Mixed juice	B	S_m	t		J		f'
Final molasses			z		m		
Final bagasse		u				h	
Filter cake		v					
Raw sugar				s			

	<i>Weight</i>	<i>% on cane</i>
Cane crushed	C	100
Mixed juice obtained	Q	Q
Imbibition water used	W	w
Weight of filter cake	X	
Weight of <i>final</i> molasses	Y	
Sucrose lost in molasses % cane		E
Sucrose recovered % cane		Z

Calculation of material balance for week (or season) by weights of cane and products.

	<i>Total weight</i>	<i>% on cane</i>
1. Corrected weight of mixed juice:	$Q' = Q \frac{100 - f'}{100}$	
2. Brix % bagasse:		$n = 100 \frac{u}{P}$
3. fibre % bagasse:		$g = 100 - h \sim n$
4. Bagasse produced:	$P = C + W - Q$	
5. Bagasse % cane:		$b = 100 \frac{P}{C}$
6. Fibre in bagasse % cane:		$lo = \frac{bg}{100} = \frac{Pg}{C}$
7. Fibre leaving mills:	$F'_0 = \frac{Cf_0}{100} = \frac{Pg}{100}$	
8. Brix passing into mixed juice:	$F' = \frac{Qf'}{100}$	
9. Fibre in cane:	$F = F'_0 + F'$	
10. Fibre % cane:		$f = 100 \frac{F}{C}$
11. Imbibition % fibre:		$100 \frac{W}{F}$
12. Sucrose in bagasse:	$U = \frac{Pu}{100}$	
13. Brix lost in bagasse:	$N = P \frac{n}{100}$	
14. Sucrose in mixed juice:	$T = Q' \frac{t}{100}$	
15. Brix in mixed juice:	$M = Q' \frac{B}{100}$	
16. Sucrose in cane:	$R = T + U$	
17. Absolute juice:	$A = C - F$	
18. Sucrose % absolute juice:		$/ = 100 \frac{R}{A}$
19. Brix of absolute juice:		$a = 100 \frac{M + N}{A}$
20. Purity of absolute juice:		$100 \frac{l}{a}$

21. Absolute juice extracted: $J = 100 \frac{M}{a}$
22. Absolute juice extracted % cane: $k = 100 \frac{I}{C}$
23. Mixed juice (corrected) % cane: $q' = 100 \frac{Q'}{C}$
24. Dilution % cane: $d = q - k$
25. Imbibition water in bagasse % cane: $w - d$
26. Sucrose % cane: $r = 100 \frac{R}{C}$
27. Sucrose (or pol) in cake: $\frac{X_v}{100}$
28. Sucrose in molasses: $\frac{Y_z}{100}$ $E = \frac{Y_z}{100C}$

Mills

29. Sucrose extracted % cane: $c = 100 \frac{T}{C}$
30. Sucrose in bagasse % cane: $100 \frac{U}{C}$
31. Extraction: $e = 100 \frac{T}{R}$
32. Reduced-extraction Mittal Rein: $e_{12.5, 12.5} = 100 - \frac{12.5 (100 - e)}{f} \left(\frac{r}{12.5} \right)^{0.6}$
33. Milling loss (sucrose/fibre): $100 \frac{u}{g}$
34. Extraction coefficient: $100 \frac{100 - e}{f}$
35. First-mill extraction: $e_1 = \frac{(S_m - S_2) S_1 e}{(S_1 - S_2) S_m}$
36. Molasses index: $\frac{100E}{100 \frac{c}{j} - c}$

Boiling house

37. Recovery: $100 \frac{Z}{c}$

38. Theoretical recovery:
$$\frac{s(j - m)}{j(s - m)} \times 100$$
39. Basic recovery:
$$L = 100 \left\{ 1.4 - \frac{40}{j} \right\}$$
40. Equivalent Standard Granulated:
$$G = 100 \left(1.4 - \frac{40}{s} \right)$$
41. Yield E.S.G.:
$$H = \frac{GZ}{100}$$
42. Recovery E.S.G.:
$$K = 100 \frac{H}{c}$$
43. Reduced recovery E.S.G.:
$$r_m = K + \frac{115 m (85 - j)}{0.85 j (100 - j)} \cdot \frac{yB_m}{qB}$$
44. Boiling-house efficiency:
$$100 \frac{K}{L}$$

Overall performance

45. Overall-recovery E.S.G.:
$$100 \frac{H}{r}$$
46. Reduced overall-recovery E.S.G.:
$$12.5, \quad 12.5^p \quad \frac{\times r_{85}}{100}$$

N.B. The Rein correction in the Mittal reduced extraction (No. 32) and our corrected Gundu Rao formula (No. 43) are still not included in the ISSCT recommendation, but appear logical and desirable.

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- 1 P. HONIG, *Principles of Sugar Technology, Vol. II*, Elsevier, Amsterdam, 1959, p. 193.
- 2 C. G. M. PERK, *Proc. 43rd Conf. SASTA, 1969*, p. 6.
- 2a H. S. YANG, T.S.J., 48 (April 1984) 12.
- 3 K. DOUWES DEKKER, *Proc. 37th Conf. SASTA, 1963*, p. 20.
- 4 G. M. PERK, *Proc. 44th Conf. SASTA, 1970*, p. 6.
- 5 K. DOUWES DEKKER, *Proc. 39th Conf. SASTA, 1965*, p. 18.

40. Vacuum Equipment

The multiple effect and the pans in a sugar factory operate under vacuum. Suitable equipment to produce this vacuum is therefore required.

The commonest arrangement of this equipment is as shown in Fig. 40.1. With the aid of an air pump, vacuum is produced in an enclosed vessel called a condenser, which communicates with the vessels to be maintained under vacuum. Cold water is pumped into this condenser to ensure condensation of the vapour coming from the pans or from the multiple effect. The condenser is placed at such a height that the water, after serving to condense the vapour, can flow out by gravity together with the condensed vapour. The condenser is thus a barometric chamber, extended at the bottom by a barometric column dipping into a well open to atmosphere. Since the vacuum approaches within 10 or 20% of perfect vacuum, and

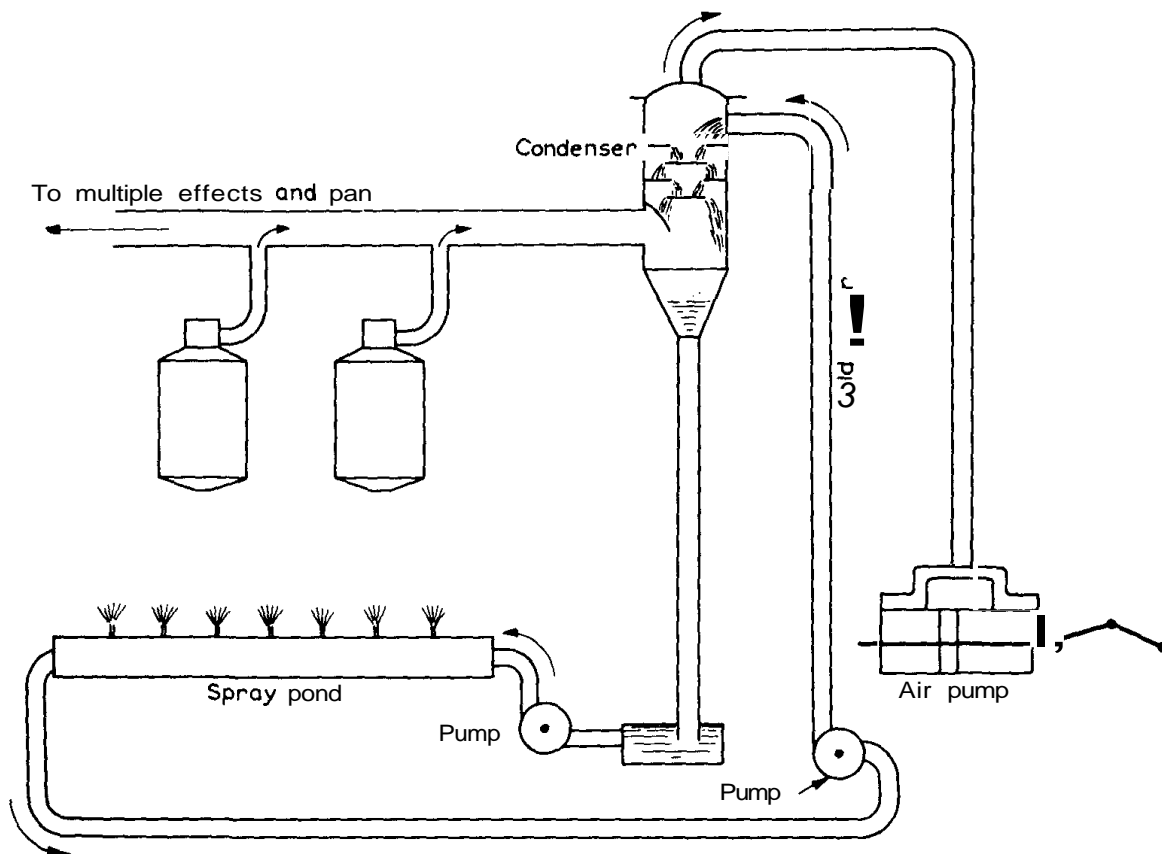


Fig.40.1. Vacuum system.

since the liquid in the barometer is water (and not mercury), the height of the column should be of the order of 10 m (instead of 76 cm):

$$0.76 \text{ m} \times 13.6 = 10.33 \text{ m}$$

where 13.6 = density of mercury relative to water.

When cold water is not available in sufficient quantity, the warm water leaving the column is pumped to a cooling pond, from which it returns at a lower temperature, and is returned to the condenser as cooling water, and so describes a closed cycle.

VACUUM SYSTEMS

The vacuum installation in a sugar factory may be more or less complex.

(1) Central condenser and air pump

The simplest system consists of centralising the whole installation at one point or in one part of the factory. We have then a single air pump, a single pump for cold water, and a single condenser for the whole factory. We have seen (p. 685) the drawbacks of this system, but also have indicated (p. 686) that there are cases where it may be advantageous.

(2) Individual condenser and air pump

Many factories, with good reason, prefer to provide a vacuum installation for each vessel to be served: one for the multiple effect, and one for each vacuum pan. Sometimes, however, two or three pans may be served by the one installation.

(3) Individual condenser with central air pump

Another method is to provide as many individual condensers and barometric columns as desired, and to connect the several condensers to a single central air pump.

In this case, if it is necessary to regulate the vacuum in the several condensers separately, it is necessary:

(a) That the air pump should work at a vacuum higher than the highest vacuum sought in the various condensers and other vessels concerned.

(b) That the various condensers should be connected to the pump by air piping of cross-section no larger than necessary.

(c) That each of these connecting pipes should be provided with a valve the section of which should advantageously be somewhat smaller than that of the pipe and which will serve to regulate the vacuum in the condenser concerned. For a fine and more stable control, a by-pass may also be installed around this valve, with an auxiliary valve of even smaller section.

(d) It is also wise to install a connection with a valve between the air pipe from the condensers which normally operate at lower vacua and the vapour entry from the condensers operating under higher vacuum, which are served by air piping of larger cross-section. Although normally closed, this connection can be very useful in the case of abnormal entry of air into the former condensers (Fig. 40.2).

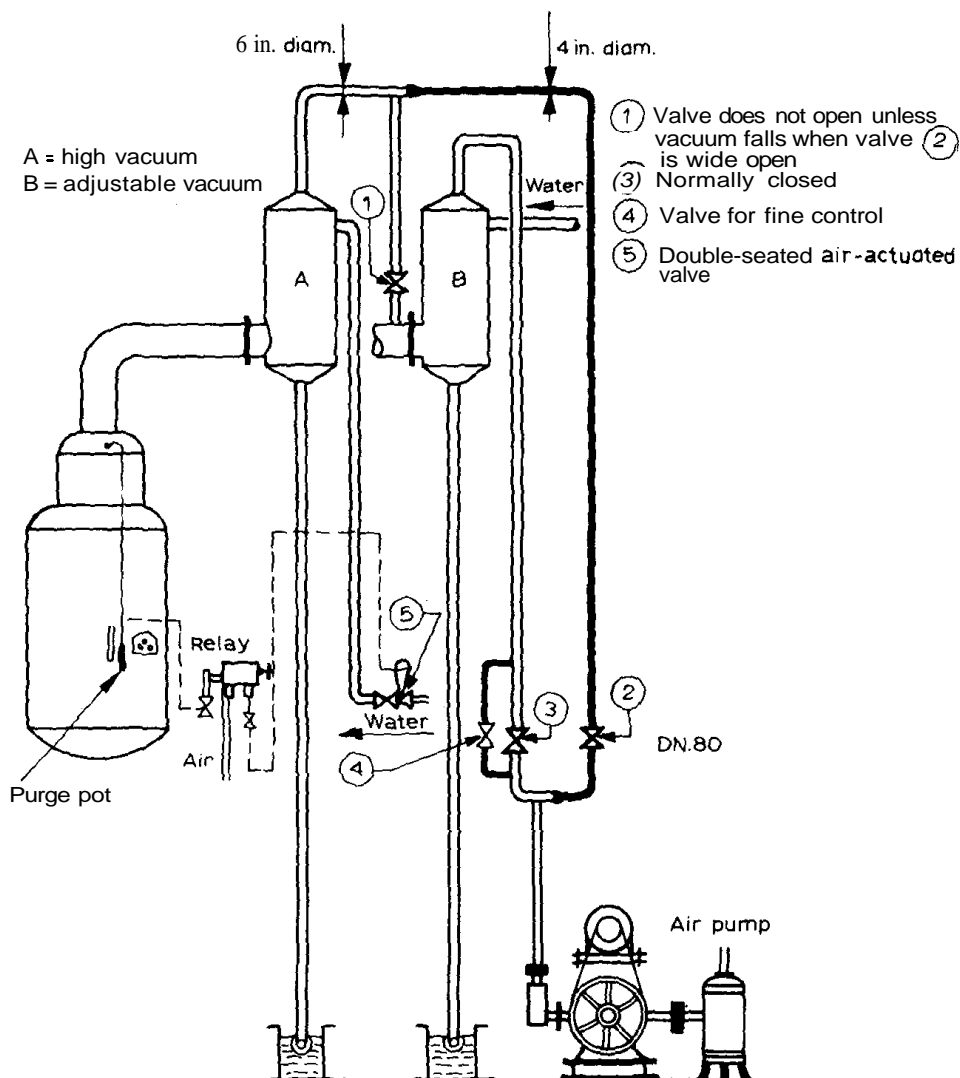


Fig. 40.2. Connection of individual condensers.

General values of the vacuum

It is possible to obtain industrially very high vacua, exceeding 75 cm ($29\frac{1}{2}$ in.) for a barometric pressure of 76 cm (30 in.). However, in the sugar factory, the vacuum sought or found in practice varies between 60 and 68 cm, and generally between 63 and 66 cm.

If the factory is located at an appreciable altitude, the foregoing values should be multiplied by the ratio $H/76$ of the local barometric height to the normal height.

Evacuation of air

Vacuum can only be maintained by continual removal of the air mixed with the vapour and due particularly to the inevitable leaks existing in the vessels and the piping. If evacuation of this air ceased, the air would accumulate, and the vacuum would fall rapidly in spite of the condensation of the vapour.

To this air, originating from leakages and various other sources which we shall see later (p. 868), is added moreover other incondensable gases originating in the juice. We speak of

the mixture (air + other incondensables gases) by designating it sometimes "incondensables", sometimes "air". We shall employ the latter term, on the understanding that it applies to all the incondensable gases accompanying the vapour reaching the condenser. This moreover is more legitimate in the cane sugar factory, since air supplies practically the whole of the incondensable gases, generally 98 – 99%.

We shall study in turn:

- (1) The condenser, or the apparatus for effecting condensation of the vapour.
- (2) The air pump, which ensures removal of the air.
- (3) The barometric column, by which the cooling water, together with the condensed vapour, is discharged.
- (4) The water pumps injecting cold water into the condenser, or returning warm water to the cooling system.
- (5) The cooling system which completes the water cycle by removing from it the heat which it has picked up in the condenser.

THE CONDENSER

The condenser is the closed vessel in which the contact between the vapour to be condensed and the cooling water is effected.

Several types of condenser are distinguished:

- (1) From the point of view of the means of extraction of the air:
 - (a) Wet air condensers, where the mixed air and water are extracted together.
 - (b) Dry air condensers, where the water and the air are removed separately (Figs. 40.4 – 6, 40.8).

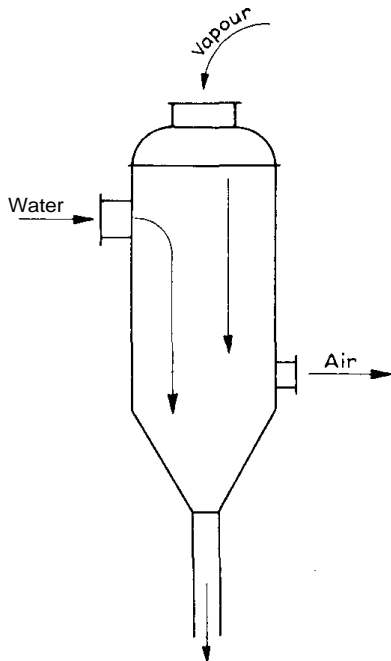


Fig. 40.3. Dry air parallel-current condenser.

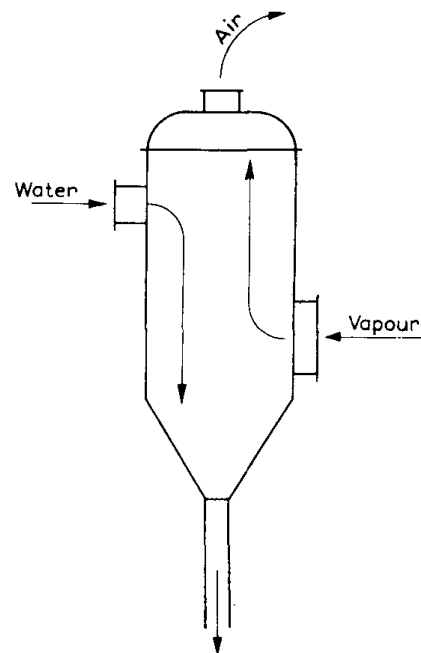


Fig. 40.4. Dry air counter-current condenser.

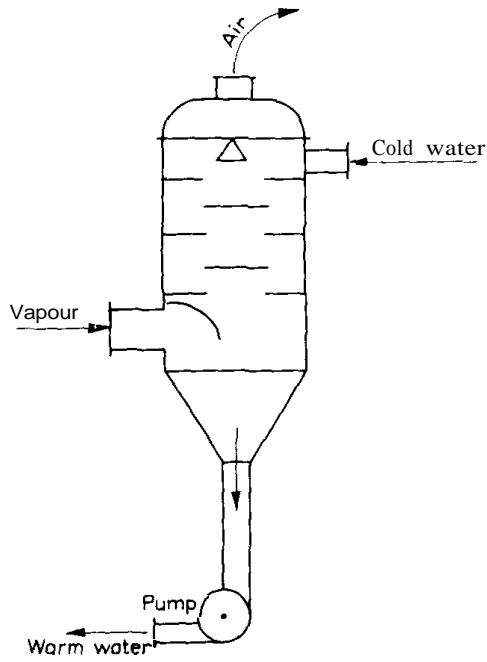


Fig. 40.5. Short counter-current condenser.

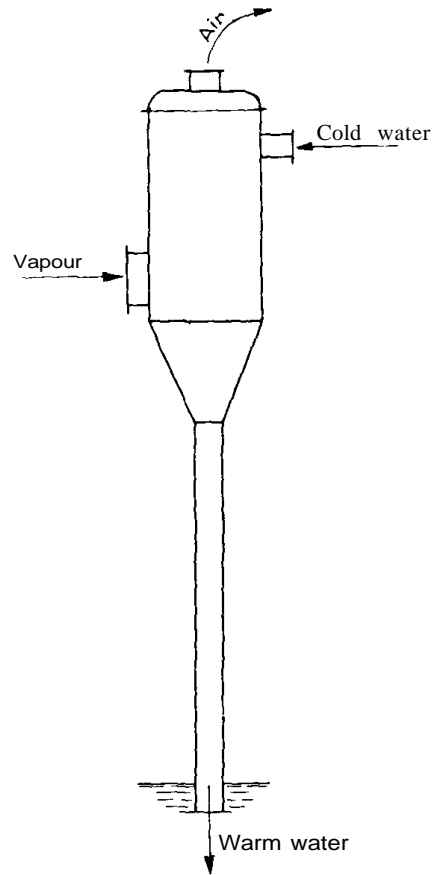


Fig. 40.6. Barometric counter-current condenser.

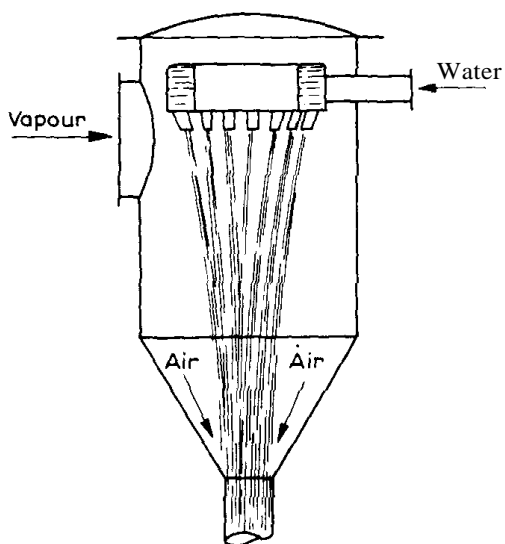
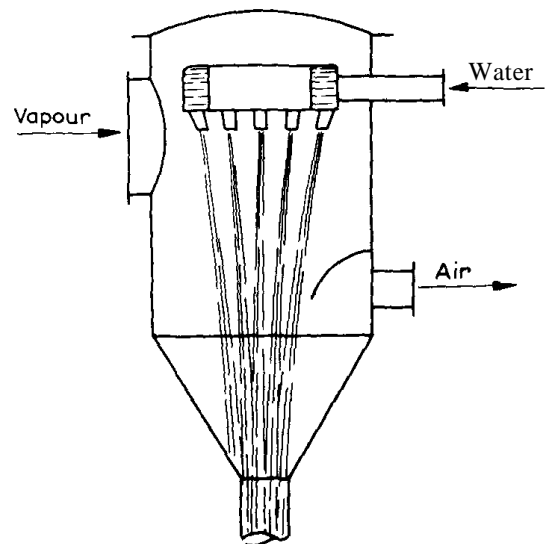
Fig. 40.7. Wet air parallel-current condenser (*jet condenser*).

Fig. 40.8. Dry air parallel-current condenser.

- (2) From the point of view of the means of extraction of the water:
- (a) Short condensers, with extraction by a pump (Fig. 40.5).
 - (b) Long or barometric condensers, with removal of water by gravity (Fig. 40.6).
- (3) From the point of view of the method of air circulation:
- (a) Co-current condensers, sometimes wrongly called parallel current condensers, in which water and air circulate in the same direction (Figs. 40.3, 40.7 and 40.8).
 - (b) Counter-current condensers, in which water and air circulate in opposite directions (Figs. 40.4—6). A counter-current condenser is obviously a dry air condenser.

The types encountered almost exclusively at the present day in the sugar factory are the following:

- (a) Dry air, counter-current, barometric condensers
- (b) Dry air, co-current, barometric condensers
- (c) Wet air, co-current, barometric condensers without air pump (jet condensers).

We shall pay attention more particularly to the dry air counter-current barometric condenser, which is the general type. We shall compare it with the co-current condenser, and shall examine separately jet condensers, which are wet air co-current, barometric condensers. We shall mention briefly certain interesting special types, and shall omit consideration of obsolete types or those which are not of practical interest.

Height of the condenser

The condenser is a device for exchange of heat between vapour and cold water. The exchange will be the more complete as contact between these two fluids is more intimate. Thus the result will depend on the surface offered by the water and on the time of contact. The area of contact depends on the course of the water, and on its degree of subdivision. The time of contact depends mainly on the height of the condenser and the number of stages or of baffles provided to break up the fall of the water into numerous small streams. At each such baffle, the molecules at the surface are replaced, and heat exchange is thus improved. However, the interval between baffles is limited by the necessity of allowing an adequate area for the passage of vapour. Hence, for a given method of breaking up the water, the height of the body of the condenser must exceed a certain minimum, to obtain an adequate efficiency of heat exchange. Deerr¹ gives:

$H = 4.50$ m (15 ft.) for a condenser without baffles.

$H = 3.65$ m (12 ft.) for a condenser with 4 baffles, this number of baffles being considered sufficient and optimum.

Hausbrand² gives the figures reproduced in Table 40.1.

The figures for the last line of this Table are given for 5 or 6 baffles, and are measured from the top plate for distribution of the descending water, in the interior of the condenser.

We may estimate, in a counter-current condenser, for total height of the body, between the cold water entry and the top of the barometric column, for 4 to 6 baffles, the heights given in Table 40.2.

TABLE 40.1

HEIGHT OF BODY OF CONDENSERS

Counter-current wet air condensers to handle 300–3,000 kg (650-6,500 lb.) vapour per hour:	
Without baffles	$H = 1 - 1.8$ m (3-6 ft.)
With 1 baffle	$H = 1 - 1.8$ m (3-6 ft.)
With 2 baffles	$H = 1.25 - 2.25$ m (4-7.5 ft.)
Counter-current dry air condensers to handle 300-12,000 kg/h (650-26,500 lb.) vapour per hour:	
Without baffles	$H = 3$ m (10 ft.) at least
With baffles	$H = 2.4 - 3.6$ m (8-12 ft.)

TABLE 40.2

HEIGHT OF CONDENSER BODIES FOR NORMAL COUNTER-CURRENT CONDENSERS

Capacity, kg vapour/h	500	5,000	20,000	100,000
lb. vapour/h	1,100	11,000	44,000	220,000
H (m)	1.5	2.5	3	6
(ft.)	5	8	10	20

or approximately 2.5 — 3 m (8 — 10 ft.) for the normal capacities in the sugar factory.

Cross-section of the condenser

Whereas the height of the condenser varies only slightly with the capacity in terms of vapour to be condensed (approximately as the cube root, if we choose the last figures given), the cross-section will depend directly on the quantity of vapour to be condensed.

Deerr¹ indicates: $S = 0.16$ m²/tonne (1.7 sq.ft./ton) vapour to be condensed per hour, where S = horizontal cross-section of the body of the condenser.

Hausbrand³ gives the figures of Table 40.3.

TABLE 40.3

CROSS-SECTION OF CONDENSER BODY FOR SMALL CAPACITIES

Vapour to be condensed (kg/h) (lb./h)		Diameter of body			
		Without baffles (mm) (in.)		With baffles (mm) (in.)	
300	650	400	16	500	20
1,000	2,200	550	22	600	24
2,000	4,400	700	28	775	30

Condenser volume

Condenser capacity has also been related to volume, the latter being reckoned as that of the cylindrical portion only, neglecting the upper and lower conical portions.

Strauss^{3a} recommends a volume of 0.75 m³/1,000 kg (12 cu.ft. per 1,000 lb.) of vapour per hour. Stewart^{3b}, however, has determined experimentally that 0.6 m³/1,000 kg (10

cu.ft./1,000 lb.) per h was sufficient in most cases and that, in two designs, 0.4 and 0.3 m³/1,000 kg (6.3 and 4.4 cu.ft./1,000 lb.) per h was adequate. The most efficient designs are those of "rain" type, *e.g.* with water falling from the top from a plate pierced with 3,308 1.3-cm ($\frac{1}{2}$ -in.) holes and a 38-cm (15-in.) central hole, above a small plate, for a condenser 2.4 m (8 ft.) in diameter; or alternatively an ordinary condenser with holes in its plates and annular baffles.

A "barrel volume" has accordingly been recommended in Australia^{3c} of 0.5 m³/1,000 kg (8 cu.ft./1,000 lb.) of vapour to be condensed per hour for the rain type, and 0.6 m³/1,000 kg (10 cu.ft./1,000 lb.) per h for the design with baffles.

It is stated^{3d} that the condensation takes place mainly on the wall of the condenser, and thus it is of advantage for part of the water flow to be discharged down the wall of the condenser. The outlet pipe for the air should be placed in the axis of the condenser.

The cone at the bottom of the barrel should have a slope of 70° to the horizontal. According to Ziegler^{3c}, this arrangement discharges twice as much water as a 30° slope.

Size of vapour pipe

Strictly speaking, the cross-section of the vapour pipe which carries the vapour from the vessel or vessels served to the condenser depends on the absolute pressure of the vapour, that is on the vacuum.

However:

- (1) The vacuum, in the sugar factory, varies only between rather narrow limits.
- (2) If we allow for a certain pressure drop, expressed as percentage of the vapour pressure, this will give vapour velocities which are greater at lower absolute pressures (*cf.* p. 513). Consequently, when the vacuum varies, if it increases for example, the decrease in density of the vapour is compensated, to a certain extent, by the increase in the velocity. It follows that the throughput of the pipe in weight of vapour decreases only slowly as the vacuum increases. For example, we could use the velocities given in Table 40.4.

TABLE 40.4

VARIATION OF VAPOUR VELOCITY WITH PRESSURE

Vacuum <i>H</i>		Normal vapour velocity <i>V</i>		Density of vapour <i>d</i>		Product	
(cm)	(in.)	(m/s)	(ft./s)	(kg/m ³)	(lb./cu.ft.)	<i>V</i> x	<i>d</i>
60	24	45	150	0.140	0.0087	6.3	1.3
65	26	50	165	0.100	0.0062	5	1
70	28	65	210	0.056	0.0035	3.6	0.74

It would therefore generally be sufficient to design the vapour pipe for a velocity of 50 m/s (165 ft./s) at a vacuum of 65 cm (26 in.). The specific weight is then 0.1, and it may be readily calculated that the pipe will still supply the same weight of vapour at a vacuum of 70 cm (28 in.) and at the velocity, still permissible, of 90 m/s (300 ft./s).

The velocities given in Table 40.5 may in fact be approached without great disadvantage.

TABLE 40.5

MAXIMUM VELOCITIES PERMISSIBLE FOR VAPOUR UNDER VACUUM

<i>Vacuum</i>		<i>Max. velocity</i>	
<i>(cm)</i>	<i>(in.)</i>	<i>(m/s)</i>	<i>(ft./s)</i>
55	21.7	70	230
60	23.6	80	262
62	24.4	85	279
63	24.8	88	289
64	25.2	92	302
65	25.6	95	312
66	26	100	328
68	26.8	110	361
70	27.6	120	394

In Australia, it is recommended that the vapour pipe cross-section should be designed to give a vapour velocity of 60 m/s (200 ft./s) or lower. It is considered that the diameter of the condenser need not be greater than that of the vapour entry.

Example. It is required to provide, with a vacuum of 65 cm (25.6 in.), for a flow of 14,400 kg (28,800 lb.) of vapour per hour. It is required to determine the diameter of the vapour pipe passing to the condenser.

We have:

Flow of vapour:

$$Q = \frac{14,400}{3,600} = 4 \text{ kg/s (8.0 lb./s)}$$

Volume of vapour (specific weight = 0.10 kg/m³ (0.0062 lb./cu.ft.)):

$$v = \frac{4}{0.1} = 40 \text{ m}^3/\text{s (1,290 cu.ft./s)}$$

Cross-section of pipe for a velocity of 500 dm/s (165 ft./s):

$$s = \frac{40}{50} = 0.8 \text{ m}^2 \text{ (7.8 sq.ft.)}$$

Diameter of pipe:

$$D = \sqrt{\frac{4s}{\pi}} = 1 \text{ m (3.2 ft.)}$$

Basing the figures on the values suggested above we may then write as a first approximation:

$$D = \frac{\sqrt{Q}}{120} \quad (40.1)$$

D = diameter of the vapour pipe serving the vessel or vessels, in dm (ft.)

Q = flow of vapour from such vessels, in kg/h (lb./h)

Diameter of cooling water pipe

The cooling water may be supplied by gravity or may be supplied by a pump. Its effective head, h , will be equal to the geometrical head at the entry to the condenser, increased by the vacuum in head of water. We have then:

$$V = a\sqrt{2gh} \quad (40.2)$$

V = water velocity in the inlet pipe, in dm/s (ft./s)

a = coefficient depending on the length of the pipe, its bends, valves and other obstructions to flow. In general, a is of the order of 0.5

g = 98 dm/s² (32.16 ft./s²)

h = head of cold water, at entry to condenser, in dm (ft.).

If the water is supplied by gravity, and if the tank from which it comes is at a lower level than the condenser, it will be necessary first to raise vacuum by means of the air pump in order that the water may be drawn into the condenser.

The supply tank should not be at more than 6 m (20 ft.) below the inlet to the condenser, if a minimum head of the order of 3 m (10 ft.) is to be obtained in normal operation (a vacuum of 66 cm (26 in.) of mercury is equal to 9 m (29.5 ft.) of water).

From the velocity V of the water in the pipe, the diameter of the latter may be readily calculated:

$$D = \sqrt{\frac{4Q_1}{\pi V}} \quad (40.3)$$

D = diameter of cooling water pipe, in dm (ft.)

Q_1 = flow of cooling water, in kg/s or l/s (cu.ft./s)

V = velocity of water, in dm/s (ft./s), given by eqn. (40.2).

Water separator

Where the air pump is of reciprocating type, it is absolutely necessary to avoid the risk of water reaching the pump, which would cause serious damage ("water hammer"). Since entrainment or formation of scum may occur, a separator is interposed for safety between the air outlet from the condenser and the air pump (Fig. 40.9). In a design project, what quantity of vapour to be condensed should be assumed as a basis for determining the capacity of the condenser?

Evaporation. The quantity of vapour to be condensed is equal to that leaving the last vessel of the multiple effect, which is readily calculated (*cf.* p. 566).

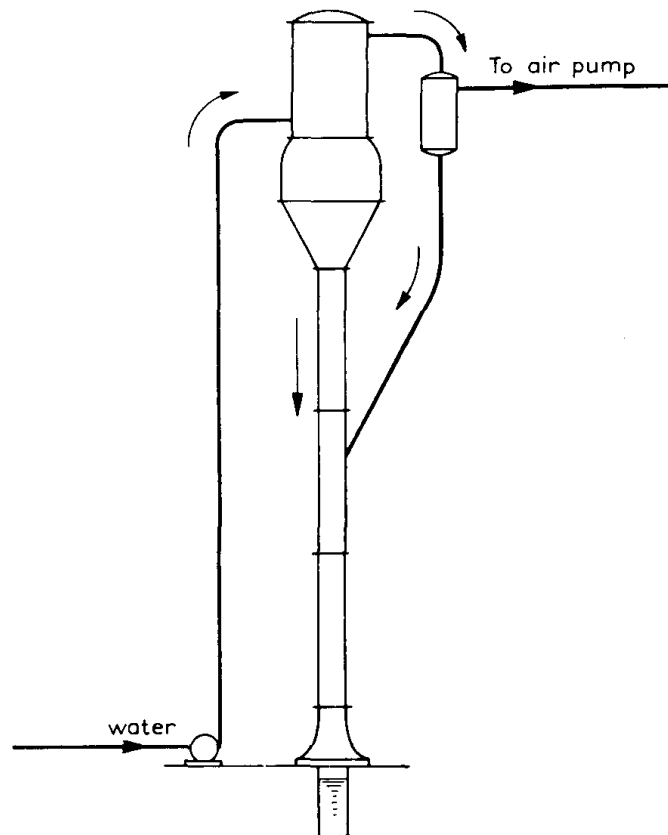


Fig. 40.9. Water separator.

We may estimate in general, in kg vapour/t.c.h., the figures given in Table 40.6.

If a heater condenser is interposed between the last vessel and the condenser, the quantities indicated in the table should be decreased by the quantity condensed in this heater.

With evaporation under pressure, we may use figures below the weight of vapour given by the last line of Table 40.6, but this quantity then becomes very variable, and it is desirable to determine it accurately.

TABLE 40.6

WEIGHT OF VAPOUR FROM THE MULTIPLE EFFECT TO BE CONDENSED (kg/t.C.h.)			
	<i>Triple effect</i>	<i>Quadruple</i>	<i>Quintuple</i>
Without bleeding	225 - 250	180 - 200	150 - 160
With bleeding	200 - 225	160 - 175	125 - 150
With bleeding and thermo-compressor	110 - 175	100 - 150	75 - 120

Pans. (a) *Central condenser.* With central condensation, it will be advisable to allow for a total quantity of vapour from the pans of:

$$125 - 150 \text{ kg (280-336 lb.) vapour to be condensed per t.c.h.} \quad (40.4)$$

the exact value depending on the amount of water returned to the pans (*cf.* p. 665).

This value, moreover, is not uniform: it varies greatly, and shows a marked maximum when a large pan commences a strike.

(*b*) *Individual condenser.* If the condenser serves a single pan, the great variation in evaporation rate during the course of each strike (*cf.* p. 662) completely alters the problem. It is necessary to design the condenser, and similarly the pumps, for the maximum evaporation, which corresponds to the beginning of the strike; and to provide a means of regulating the input of cooling water during the remainder of the strike in proportion to the quantity of vapour to be condensed, which decreases rapidly until the end of the strike.

The dimensions of the condenser will be determined as a function of the maximum evaporation rates given in Table 40.7.

TABLE 40.7

MAXIMUM EVAPORATION AT PANS AT START OF STRIKE		
	(kg/m ² H.S./h)	(lb./sq.ft. H.S./h)
Footing	80	16
A massecuite	60	12
B massecuite	50	10
C massecuite	40	8

If the condenser serves several pans, we may estimate:

- (a) The maximum rate for the largest pan
- (b) The mean rate for the others.

TABLE 40.8

MAXIMUM AND MEAN EVAPORATION RATES FOR PANS			
<i>Max. rate</i>	<i>Mean rate (kg/m²/h)</i>		
	<i>Mechanical circulation</i>	<i>Natural circulation</i>	
Footing	85	50	25
A massecuite	70	40	20
B massecuite	50	30	15
C massecuite	40	20	10

For lb./sq.ft./h, divide these figures by 5.

For this calculation it will be necessary to take more accurate figures for maximum rates than those of Table 40.7. Calculations may be based on the evaporation rates of Table 40.8, expressed in weight of vapour to be condensed per hour and per unit of heating surface (see also Table 33.7).

WEIGHT OF COOLING WATER

The quantity of cooling water necessary for condensation of the vapour is readily found by equating the heat given up by the vapour in condensing to the heat absorbed by the cold water.

Let:

W = weight of cooling water necessary per unit weight of vapour to be condensed

t_1 = temperature of cooling water available, in °C (°F)

t_2 = permissible temperature of warm water, in °C (°F)

t_v - vapour temperature corresponding to the vacuum obtained in the condenser, in °C (°F).

Then:

$$W(t_2 - t_1) = 1 [607 - 0.7(t_v) + t_v - t_2]$$

whence:

$$W = \frac{607 + 0.3 t_v - t_2}{t_2 - t_1} \quad (40.5)$$

Since the difference $t_v - t_2$ is small, especially in a counter-current condenser (of the order of 5°C, for example), and since t_v varies between 62°C ($H = 60$ cm) and 47°C ($H = 68$ cm), and generally lies in the immediate vicinity of 55°C, we see that the numerator of the second member of the expression for W scarcely varies. No great error will be made in adopting a mean value and writing:

$$W = \frac{572}{t_2 - t_1} \quad \left(\frac{1,030}{t_2 - t_1} \right) \quad (40.6)$$

Variation of W as a function of $t_2 - t_1$. Returning to the complete formula (40.5): the value of W depends on $t_2 - t_1$, but is little affected by t_v or t_1 . Hence we shall choose a mean value for the temperatures of vacuum and of cold water, and study the variation of W as a function of $t_2 - t_1$. For example:

$$H = \text{vacuum} = 64 \text{ cm} \quad \text{whence: } t_v = 55^\circ\text{C and } t_1 = 30^\circ\text{C}$$

We have then:

$$W = \frac{623 - t_2}{(2 - t_1)} \quad \left(\frac{1,154 - t_2}{t_2 - t_1} \right)$$

Hence we obtain the curve of Fig. 40.10 giving the variation in W as $(t_2 - t_1)$ varies from zero to a maximum of 25°C (we shall see later that t_2 cannot exceed t_v). We have extended the curve beyond that figure for the case where t_1 is very low, or t_2 very high.

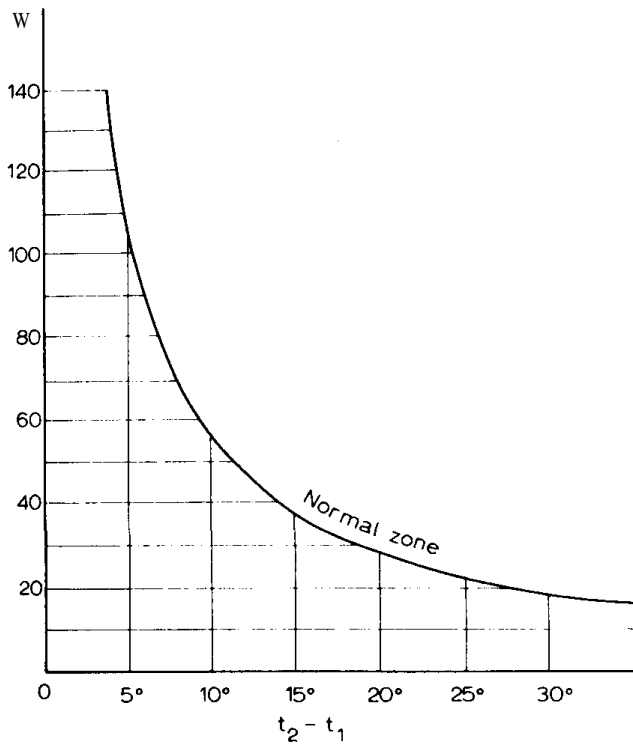


Fig. 40.10. Variation of W as function of $(t_2 - t_1)$.

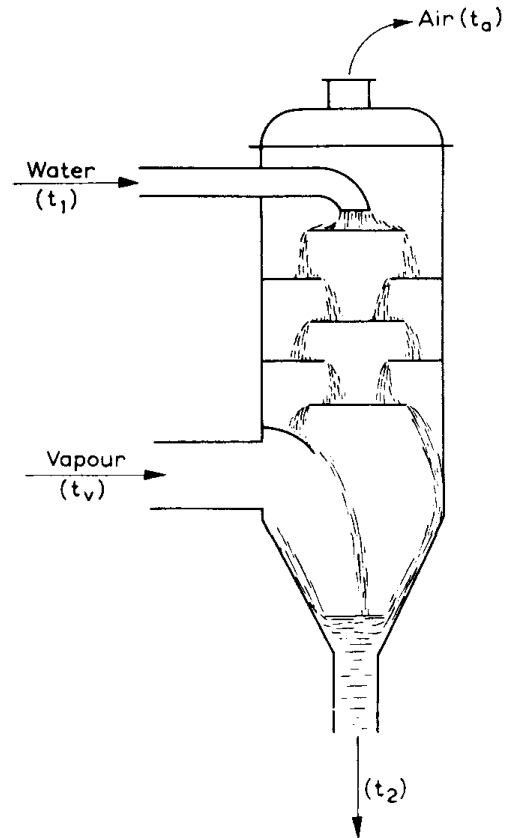


Fig. 40.11. Temperatures in condenser.

It is seen that the quantity of water necessary increases rapidly as $t_2 - t_1$ decreases. On the other hand, it decreases only very slowly when $t_2 - t_1$ is very high. It is of advantage, therefore, to keep to a moderate figure, and especially not to descend below a value of 10°C for $(t_2 - t_1)$.

Considerations on temperatures

Let us consider for example a counter-current barometric condenser (Fig. 40.11). Let:

t_v = temperature of vapour to be condensed, corresponding to the vacuum

t_1 = temperature of cold injection water

t_2 = temperature of hot water leaving the condenser

t_a = temperature of the air at exit from the condenser.

We may make the following comments:

(1) t_2 is lower than or at most equal to t_v . The temperature of the water cannot exceed that of the vapour; otherwise, since the vapour pressure in the condenser corresponds only to t_v , the water which would then be at a higher temperature would vaporise instantly and would give up heat instead of absorbing heat. This is exactly what happens when water (or juice) heated to 105 or 110°C is released from a heater to the atmosphere; it immediately flashes into vapour and drops to 100°C , the maximum temperature at which it can exist under atmospheric pressure.

Since the condenser is a heat exchanger the object of which is to absorb with a minimum of water all the heat content corresponding to the latent heat of the vapour, its efficiency will be the better as t_2 approaches t_v , the ideal being represented by $t_2 = t_v$.

The difference $t_v - t_2$ therefore represents the margin separating the condenser under consideration from the maximum efficiency, or the degree to which it approaches that efficiency. We shall call it the "approach":

$$\text{App} = \text{Approach} = t_v - t_2 \quad (40.7)$$

(a) In a counter-current condenser, the approach varies between 3 and 10°C (5–18°F). As a mean value, we may reckon 5°C (9°F). More accurately, we may take:

$$\text{App} = t_v - t_2 = (0.1 + 0.02a) (t_v - t_1) \quad (40.8)$$

a = proportion of air in the vapour = weight of air % on weight of the air - vapour mixture (cf. Table 40.16).

a generally varies from 0.5 to 2% (cf. Badger⁴; Hausbrand⁵). Hence the approach is of the order of 10–15% of $t_v - t_1$. We shall retain the most frequent value:

$$\text{App} = t_v - t_2 \approx 0.12 (t_v - t_1) \quad (40.9)$$

being careful to note that we must use a figure of 2°C (5°F) in the case where this formula would furnish a value lower than this minimum.

(b) In a co-current condenser, the approach is obviously not as good. Hausbrand⁶ gives:

$$\text{App} = t_v - t_2 \approx 0.66 (t_v - t_1) \quad (40.10)$$

which signifies that the efficiency of a co-current condenser is only about 1/3. However, Webre and Robinson⁷ state that certain types of co-current condenser attain an approach of 4–4.5°C (9–11°F).

Tromp⁸ indicates 5–11°C (12–27°F), with a mean of 8.5°C (20°F), for jet condensers, according to the fluctuations in the quantity of vapour to be condensed.

(2) t_a lies between t_1 and t_2 . The temperature at which the air leaves the condenser, t_a , obviously lies between the temperature at which it enters, which is equal to that of the vapour which carries it, t_v (or close to it), and the temperature of the cold injection water, which is the fluid of lowest temperature within the condenser.

However, the cooling produced by this water makes the outgoing water the hottest fluid, and the air can be no hotter. Its temperature will then lie between t_2 and t_v .

It is found in practice that contact between air and water in the condenser is sufficient to ensure complete equalisation of temperature, according to the system employed. Hence we have:

$$(a) \text{ Counter-current condenser: } t_a = t_1 \quad (40.11)$$

$$(b) \text{ Co-current condenser: } t_a = t_2 \quad (40.12)$$

Comparison between co-current and counter-current condensers

In condenser problems, the data, from the point of view of temperatures, are generally t_1 and t_v : cooling water is available at a certain temperature, and a certain vacuum, which one wishes to maintain, is specified.

Under these conditions, the comparison between the results to be expected from the two types of condenser shows little but advantages in favour of counter-current condensers:

- (1) Its efficiency is higher: it permits the cold water to be heated to a higher temperature t_2 .
- (2) For a given duty, it therefore allows smaller quantities of water. The quantity of heat exchanged M is equal (see eqn. 40.5) to:

$$M = QW(t_2 - t_1) \quad (40.13)$$

Q = weight of vapour to be condensed.

If $(t_2 - t_1)$ decreases, W must increase.

(3) Now, the cooling water brings with it dissolved air, which is released and assumes the gaseous form when exposed to the vacuum in the condenser. Consequently, the duty of the air pump will likewise be increased with a co-current condenser.

(4) However, what especially increases the duty of the air pump in this case is the fact that the co-current condenser supplies air at t_2° , whereas the counter-current condenser supplies air at t_1° , that is, at a lower temperature. It follows: (a) that its specific volume is lower; (b) that the quantity of water vapour which it will carry with it (the air leaving the condenser is obviously saturated) will similarly be smaller.

Finally, the air pump serving a co-current condenser will require a capacity roughly double that of the air pump serving a counter-current condenser receiving the same cooling water and maintaining the same vacuum.

These considerations, however, are theoretical rather than practical and it is appropriate on the other hand to mention the following points:

(7) If we adopt in the co-current condenser a barometric column of sufficiently small cross-section to obtain a high velocity of flow, 21 dm/s (7 ft./s) or more, part of the air will be entrained with the water, and the duty of the air pump will be correspondingly reduced. This phenomenon, moreover, is the basis of jet condensers.

(2) If the cooling water is supplied from a spray pond, and if the latter is fairly generously designed, the cooling water temperature will be slightly lower in the case of the co-current condenser, since the cooling $(t_2 - t_1)$ obtained in the spray pond will decrease only slightly with t_2 . The decrease in t_2 due to co-currents (*cf.* par. 1 above) will be reflected to a great extent in the value of t_1 .

Air and vapour pressures in the condenser

The curtains of water sometimes offer a slight resistance to the passage of the vapour and the air. Adding the loss of head due to the flow of these gases, we may estimate the pressure drop in the condenser at approximately 3 mm (0.1 in.).

If we neglect this slight difference, we may say that the pressure p is the same throughout the body of the condenser.

Now, this pressure p is made up of the sum of the partial pressures of vapour and air:

$$p = p_v + p_a \quad (40.14)$$

p_v = partial pressure of vapour

p_a = partial pressure of air.

The partial pressure of the air, p_a , is very low at the vapour entry. If, for example, the absolute pressure p is 100 mm and if the air forms 1 % of the volume of the mixture, we may consider that 1,000 volumes of mixture contain⁹:

990 vol. of vapour at 100 mm
10 vol. of air at 100 mm

or:

1,000 vol. of mixture at 100 mm

or this may preferably be expressed:

1,000 vol. of vapour at 99 mm
1,000 vol. of air at 1 mm

or:

1,000 vol. of mixture at 100 mm

However, the vapour condenses: its pressure decreases, and that of the air increases (Fig. 40.12). At the outlet, that is, at the point where the air is extracted by the pump, we have air simply saturated with vapour. Its pressure is then:

$$P_s = P - p_v \quad (40.15)$$

p_a = pressure of the air at outlet from the condenser

p = pressure in the condenser corresponding to the vacuum

p_v = vapour tension corresponding to the temperature t_a given by Table 32.1.

AIR PUMPS

Two main types of air pump exist. The oldest and best known at present is the reciprocating pump, analogous in construction and appearance to the steam engine. It is scarcely encountered in new installations and has been abandoned in most sugar countries. It has been studied exhaustively in the second edition of this book (English edition, pp. 811 - 830) and we refer any interested readers to that description. Here we treat only the present type, which is much more practical and economic: the air pump with liquid annulus.

Volume of air to be extracted

The air introduced into the condenser comes from four sources:

- (1) Air contained in the heating steam
- (2) Air and incondensable gases coming from the juice

- (3) Air introduced with the injection water
- (4) Air entering by leakage.

These have already been studied in detail in the second edition. We now summarise these conclusions.

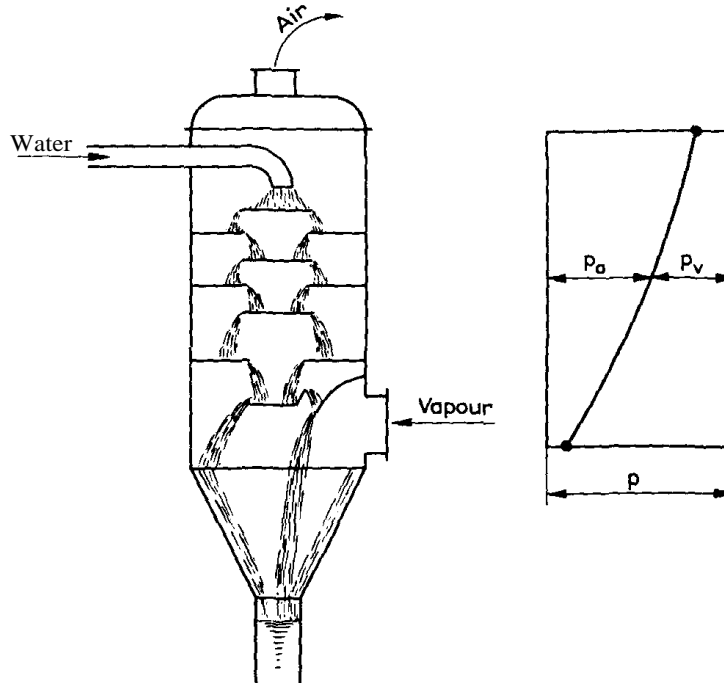


Fig. 40.12. Pressure in condenser.

(1) **Air contained in heating steam.** This originates in the air dissolved in the feed water going to the boilers, which naturally escapes with the steam. Where the incondensibles are rejected into the atmosphere, none of this goes into the condenser. In other cases, where the gases are passed to the vapour space, we may reckon:

$$a_1 \approx 5 - 10 \text{ p.p.m. of steam emitted to the calandria}$$

(2) **Air and incondensibles from the juice.** Dissolved air in the juice has escaped at the flash tank before the clarifier. We have only to take into account the small quantity which has been dissolved in the juice between clarification and evaporation. We may reckon:

- (a) Evaporation: $a_2 = 200 - 250$ p.p.m. of juice entering the multiple effects
- (b) A and B vacuum pans: $a_2 = 100$ p.p.m. of syrup entering the pan
- (c) C pans: $a_2 = 50$ p.p.m. of syrup entering.

(3) **Air introduced with injection water.** Water at atmospheric pressure contains dissolved air which is released when it arrives in the condenser under vacuum. We may reckon, as the proportion of the weight of injection water, depending on the source of the water:

- (a) Calm rivers or ponds: $a_3 = 20 - 30$ p.p.m.
- (b) From mountain streams or cascades: $50 - 150$ p.p.m.

(c) From sea water: 10 – 20 p.p.m.

(d) From a cooling pond or tower: 30 – 40 p.p.m.

(4) Entry from leakages. This is the most important quantity, especially in installations not well maintained, but it is very variable and depends on the care taken in making the joints, and on the total surface of equipment under vacuum and the number of joints and stuffing boxes. We may reckon:

(a) Multiple effects: $a = 3500$ p.p.m. of steam condensed within the last vessel.

(b) Vacuum pans: $a = 500$ p.p.m. per hour for the working volume of the massecuite.

Total figures. The foregoing data on the origin of air in the vapour are useful, but it would be very misleading to use them for design requirements of air pumps. This information is obtained from simple data such as the following, calculated on different bases.

ROTARY LIQUID-ANNULUS AIR PUMPS

As already discussed (p. 868), reciprocating air pumps are cumbersome and expensive. They are now replaced by an ingenious and elegant machine: the rotary air pump with an annulus of water. This pump consists of a shaft carrying radial vanes, rotating in a drum (Fig. 40.13). By means of eccentric placing of the shaft relative to the drum, or by giving the drum an elliptical or oval form, it is arranged that the vanes enclose a space at first increasing, connected to the suction, and then decreasing for compression of the air and subse-

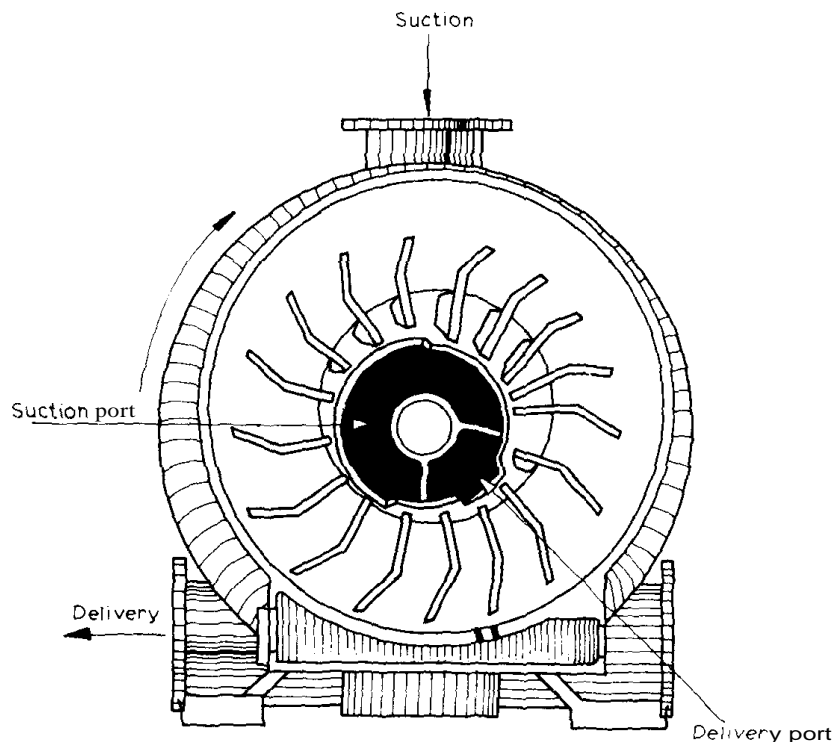


Fig. 40.13. Principle of liquid-annulus air pump.

quent connection to the delivery. This space is limited on the outside by an annulus of water lining the drum and held against the walls by centrifugal force. Part of the water leaves with the compressed air, but can be recovered and re-cycled. It also serves to remove the heat of compression and so cool the drum.

These pumps run, according to their dimensions, at speeds between 250 and 1,500 r.p.m. They are driven by an electric motor, either directly coupled or by Vee-belts. Their volumetric efficiency is of the order of 0.65 — 0.75. They were originally built in two stages, but at the present day single-stage pumps are made which can give a vacuum greater than 70 cm (27.5 in.).

These pumps take up little space, consume very little lubricant and require only little attention. They cause scarcely any vibration and so do not require foundations; they may be installed on the pan floor, and thus the vacuum piping can be substantially shortened. Entry of water with the air is no disadvantage, thus the water separator may be omitted.

Compared with jet condensers, they require only very little water, and this quantity of water can be proportioned to the weight of vapour to be condensed, and thus can be substantially decreased from the beginning to the end of a strike, when used with a vacuum pan. They are consequently adaptable for automatic vacuum control and give an excellent solution for individual vacuum. They thus offer at the same time economy in first cost, maintenance cost and power consumption.

Relative to steam ejector condensers, they consume only about one-eighth of the equivalent energy, giving at the outset much better efficiency, and the vacuum produced is much more independent of the quantity of water and of vapour supplied. They are greatly superior where air leaks are substantial, and where there is a shortage of water or of steam.

They may be used to supply vacuum for rotary filters of the Oliver type, using a special, lower-priced model.

TABLE 40.9

VOLUME OF AIR TO BE REMOVED PER UNIT OF VAPOUR, IN l/kg (cu.ft./lb.) OF VAPOUR

For central condenser:	$V = 100 - 110$	($V = 1.6-2$)
For multiple effect:	$V = 80-100$	($V = 1.3- 1.6$)
For vacuum pan:	$V = 20 - 25$	($V = \backslash -1.3$)

For vacuum pan, the figures are for maximum rate at commencement of the strike.

TABLE 40.10

WEIGHT OF VAPOUR FROM A MULTIPLE EFFECT

	<i>Triple</i>		<i>Quadruple</i>		<i>Quintuple</i>	
	<i>kg/t.c.h.</i>	<i>lb./t.c.h.</i>	<i>kg/t.c.h.</i>	<i>lb./t.c.h.</i>	<i>kg/t.c.h.</i>	<i>lb./t.c.h.</i>
Without bleeding	225-250	500-550	180-200	400-450	150 - 160	336-360
With bleeding	200-225	450-500	160-175	360-400	125-150	280-336
Bleeding and vapour comp.	110-175	250-400	100-150	225-336	75-120	170-270

TABLE 40.11

MAXIMUM AND MEAN EVAPORATION RATE FOR PANS

	<i>Maximum rate</i>				<i>Mean rate</i>			
	<i>Author's figures</i>		<i>Nash figures</i>		<i>Mechanical circulation</i>		<i>Natural circulation</i>	
Footing	85	(17)	83	(17)	50	(10)	25	(5)
A m.c.	70	(14)	68	(14)	40	(8)	20	(4)
B m.c.	60	(12)	68	(14)	35	(7)	15	(3)
C m.c.	40	(8)	39	(8)	20	(4)	10	(2)

More accurate figures would be obtained by allowing for temperature of the condenser water, which is important.

Calculations for liquid-annulus air pumps

Calculations of these pumps are based on the volume V to be extracted, obtained by means of Tables 40.9 or 40.10, with the aid of Table 40.11 for vacuum pans.

A simplified and rapid calculation may also be made in the following manner.

A very high vacuum is generally not desired, due to the disadvantages which have been discussed: danger of entrainments in the evaporators, difficulty in ensuring satisfactory performance with vacuum pans. The author would recommend that it is advisable to keep to 66 cm (26 in.); even if it is not intended to work above 62 or 64 cm (24 or 25 in.), this value ensures a useful safety margin:

$$\begin{aligned}
 p &= 66 \text{ cm (26 in.) of vacuum} = 10 \text{ cm (4 in.) absolute pressure} \\
 &= 0.136 \text{ kg/cm}^2 \text{ (1.93 p.s.i.a.) (cf. Table 32.1).}
 \end{aligned}$$

The air temperature to be considered here is no longer that of the air leaving the condenser (eqn. 40.11), but that of the water used for forming the liquid annulus. Actually, the contact of the air with the annulus is sufficiently intimate to reduce the air to a temperature very close to that of the liquid. Therein lies one of the advantages of these pumps; the coldest water in the factory may be used for them, since they consume very little 2–4 l/s (25–50 Imp.gal./min), thus profiting by a specific volume of the air substantially lower than that corresponding to the temperature of the cold water from the condenser, which often rises to 30–36°C (86–97°F).

Example. If only an order of magnitude is required for a preliminary calculation, with a good margin of safety, we may be content with using Tables 40.9 and 40.10. For example, a vacuum pump is required to serve a quadruple effect with vapour bleeding, for a factory of 200 t.c.h. Table 40.10 gives: 170 kg of vapour per t.c.h., Table 40.9: 90 l of air per kg of vapour.

Hence: 170 x 90 = 15,300 l/t.c.h. (550 cu.ft./t.c.h.), and for 200 t.c.h.:

$$200 \times 15,300 = 3,060,000 \text{ l/h} = 51 \text{ m}^3/\text{min} \text{ (1,830 cu.ft./min)}$$

However, generally we have more complete data. Assuming:

TABLE 40.12

VOLUME OF AIR TO EXTRACT FOR 66 Cm (26 in.) VACUUM ABS. PRESS. 10 Cm (4 in.)

	<i>Temperature of condenser water</i>				
	20 °C	25 °C	30 °C	35 °C	
Multiple effect	123	136	178	236	l/min per t/E juice
A and B pans	16.2	18.7	24.5	32.8	l/mir per m ²
C pans	11.5	13.5	17.8	22.4	l/min per m ²

For vacuum other than 66 cm. Nash gives the coefficients shown in Table 40.12 as applicable to the volumes of air in this table.

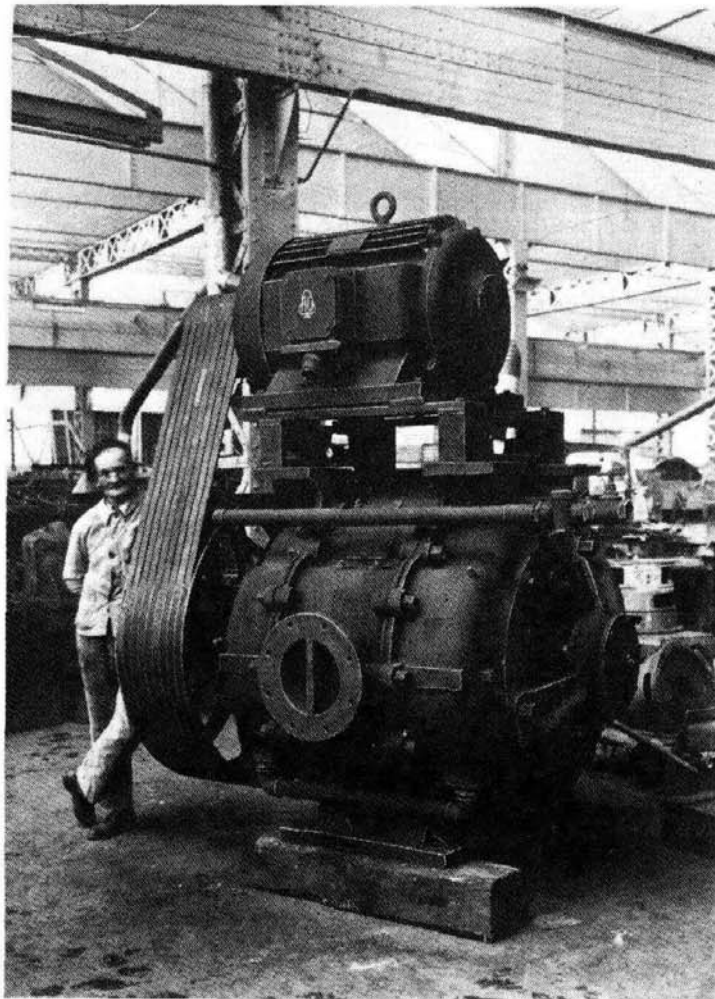


Fig. 40.14. "Hydro B15" liquid-annulus air pump.

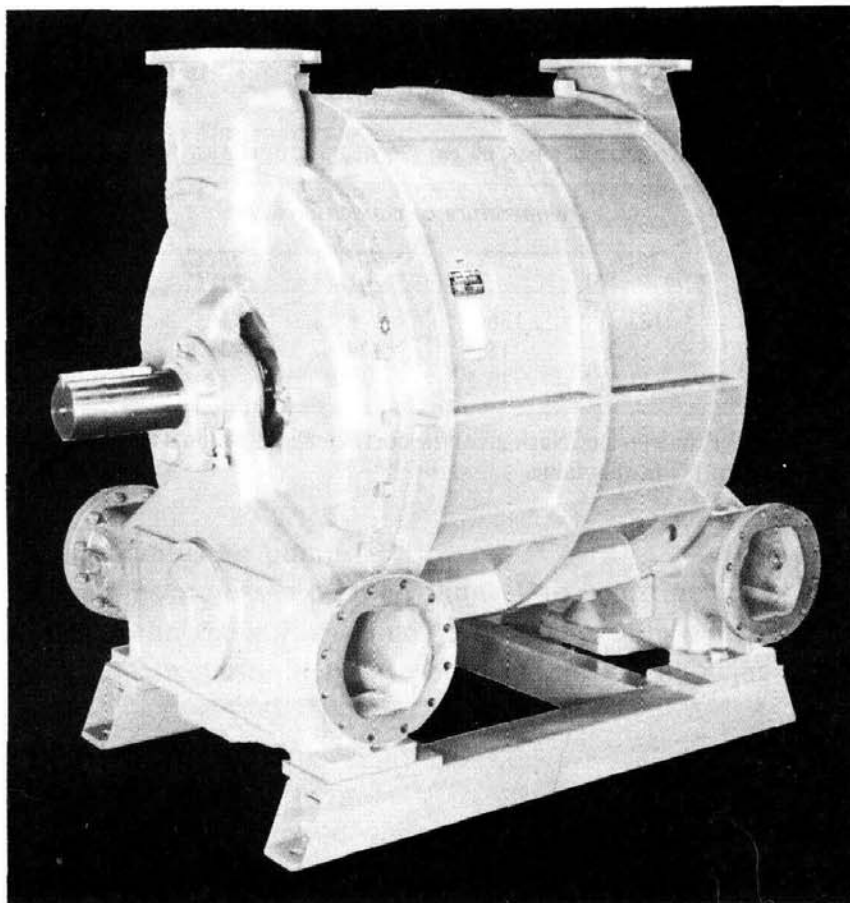


Fig. 40.15. Nash liquid-annulus air pump.

TABLE 40.13

COEFFICIENTS APPLICABLE TO VOLUMES OF TABLE 40.12 FOR VACUUM OTHER THAN 66 cm

Temperature (°C)	Vacuum (cm)			
	64	66	68	70
20	0.80	1	1.31	1.94
25	0.79	1	1.34	2.08
30	0.77	1	1.40	2.26

Nash also takes into account the difference d between the condenser water and the water forming the liquid annulus, for which the coldest water available is used (see Table 40.14).

Crushing rate:	200 t.c.h.
Mixed juice % cane:	100%
Vapour from last vessel:	22,000 kg/h
Vacuum required:	66 cm
Water temperature to condenser:	30°C
Water supply to pump annulus:	25°C

We require a "Hydro" or Nash pump.

Calculation: Volume of air to be extracted (Table 40.12), 178 l/min x 200 t.c.h. = 35.6 m³/min (1,284 cu.ft./min).

TABLE 40.14

COEFFICIENT TO APPLY TO VOLUMES OF TABLE 40.12 AS FUNCTION OF TEMPERATURE DIFFERENCE *d* BETWEEN CONDENSER COOLING WATER AND ANNULUS COOL WATER

<i>Vacuum (cm)</i>	<i>d (°C)</i>					
	3	5	6	8	10	12
64	1.050	1.088	1.097	1.111	1.126	1.140
66	.067	1.104	1.112	1.127	1.141	1.156
68	1.084	1.120	1.128	1.143	1.157	1.172

For continuous pans, the quantity of vapour varies greatly depending on whether agitating steam ("jigger steam") is used. Table 40.15 gives appropriate figures.

Taking into account the 5° gained due to the annulus water, a B. 13 pump (Tables 40.14 and 40.17) could extract 32 x 1.104 = 35.3 mVmin (1.130 x 1.104 = 1.250 cu.ft./min). This could thus suffice if need be, but a B. 15 would be advisable to attain a good margin of security. Similarly, a Nash 2000 (100 h.p.) would be suitable (Table 40.18).

Central condenser. Figures for a central vacuum system have been included in the foregoing tables. This is, in fact, still an exceptional case at present (1982), but it is probable that it will become more and more frequent in the future. This is the case where the factory is

TABLE 40.15

WEIGHT OF VAPOUR FROM A CONTINUOUS PAN

	<i>kg/t .c.h.</i>		<i>kg/m² heating surface^b</i>	
	<i>Without agitation</i>	<i>With agitation</i>	<i>Without agitation</i>	<i>With agitation</i>
A m.c.	60	90	14	20
B m.c.	30	45	12	18
C m.c.	20	30	8	12

^a For lb./t.c.h. multiply by 2.2

^b For lb./sq.ft. multiply by 0.2.

TABLE 40.16

WEIGHT OF AIR TO EXTRACT IN g PER kg OF VAPOUR TO BE CONDENSED

Central vacuum	8 - 10
Multiple effects	6 - 8
Vacuum pans	5 - 6

entirely equipped for continuous operation, *i.e.* where all the vacuum pans (including high grades) are continuous pans. There is then nothing to disturb the regularity of the vacuum, and the motives of economy and simplicity act in favour of a central vacuum system.

For reasons previously discussed (pp. 506, 610, 649), in this case a slight difference will be maintained between the vacuum for the evaporators, which would be fixed at about 64 cm, and that for the vacuum pans, which would be for example 66 cm, with an automatic valve in the vacuum line to maintain this difference in vacuum between the two.

From weight of air to extract. It is in fact more accurate to base calculations on the weight rather than the volume of air to be handled, since this volume varies with the vacuum for a given weight. Taking the weight as a basis, we avoid the approximations made in the preceding tables, depending on the vacuum (assumed in the tables as 66 cm) and the temperature (Table 40.16).

TABLE 40.17

CAPACITY AND POWER OF "HYDRO" NEYRPIC PUMPS TYPE B

Model	R.p.m.	Motor power (kW)	Water (m^3/h) (cu.ft./h)	750	100	80	60	Abs. press mm Hg Vacuum mm Hg Vacuum in. Hg
				610	660	680	700	
				24	26	26.8	27.6	
B.7	740	9	2 70	6.20	6.20	6.10	5.45	mVmin
				220	220	215	190	cu.ft./min
				8.25	8	7.50	6.95	kW
B.7 ^b	740	15	4 140	8.25	8.10	7.90	7.60	mVmin
				290	285	275	270	cu.ft./min
				11.20	10	9.40	8.75	kW
B.9	580	15	2.5 90	10.30	10.30	10.20	9.10	mVmin
				365	365	360	320	cu.ft./min
				13.80	13.50	13	12	kW
B.10	500	23	3 100	15.20	15.10	14.50	13.70	mVmin
				535	535	510	485	cu.ft./min
				20.20	20	19.10	17.80	kW
B.12	406	30	3.5 125	21	19.30	17.75	16	mVmin
				740	680	625	565	cu.ft./min
				26.50	25.25	24.40	22.10	kW
B.13	340	45	5 175	32	32	30.50	26	mVmin
				1,130	1,130	1,090	920	cu.ft./min
				41.20	38.20	35	32	kW
B.15	270	75	10 350	51	51	51	45.50	mVmin
				1,800	1,800	1,800	1,600	cu.ft./min
				63.50	61	59.50	54.50	kW
B.15 ^b	280	120	13 460	75	70	66	57.50	mVmin
				2,650	2,470	2,330	2,030	cu.ft./min
				101.5	97	92.50	88	kW

The quantity of water forming the liquid annulus is approximately 2.5–4 l/m³ of air aspirated (2.5–4 cu.ft. per 1000 cu.ft.).

TABLE 40.18

POWER AND CAPACITY OF NASH VACUUM PUMPS (TYPE CL)

No.	H.p.	Capacity in m^3 (cu.ft.) per min					
		51 cm (20 in.)	61 cm (24 in.)	66 cm (26 in.)	66 cm (26 in.)	66 cm (26 in.)	66 cm (26 in.)
200	7.5	4	(140)	3	(100)	3	(100)
	10	5	(175)	4	(140)	4	(140)
400	20	10	(350)	9	(320)	8	(280)
	25	12	(425)	11	(390)	9	(320)
	30	13	(460)	12	(425)	10	(350)
700	30	16	(565)	15	(530)	13	(460)
	40	19	(670)	17	(600)	15	(530)
	50	21	(740)	19	(670)	17	(600)
1,000	50	27	(950)	23	(810)	21	(740)
	60	30	(1,060)	26	(920)	25	(880)
2,000	75	42	(1,480)	40	(1,400)	26	(920)
	100	53	(1,870)	50	(1,765)	45	(1,590)
	125	60	(2,120)	57	(2,010)	52	(1,835)
3,000	125	70	(2,470)	64	(2,260)	58	(2,050)
	150	80	(2,825)	80	(2,825)	68	(2,400)
4,000	150	90	(3,180)	80	(2,825)	67	(2,365)
	200	110	(3,880)	106	(3,740)	94	(3,320)
6,000	250	147	(5,190)	125	(4,430)	130	(4,590)
	300	164	(5,790)	160	(5,650)	150	(5,300)
9,000	400	226	(7,780)	215	(7,590)	210	(7,415)
	450	250	(8,830)	240	(8,475)	230	(8,120)
	500	266	(9,400)	—	—	—	—

The volume of water for the liquid annulus should be approximately 2–2.5 l/m^3 of air pumped.

Knowing the weight of air, it converts to volume by applying Mariotte's Law:

$$v = \frac{RT_a}{p_a}$$

v = specific volume of the air under absolute pressure p_a and temperature T_a , in l/g
 R = constant for perfect gases = 0.002927 for metric units (0.3697 in British units).

T_a = air temperature in degrees Kelvin = 273 + t_a

t_a = temperature of air leaving the condenser, given by formulae (40.11) and (40.12), in $^{\circ}C$

p_a - partial pressure of the air, in kg/cm^2 by formula (40.15) and Table 32.1

The specific volume v also gives the total volume $V = Av$

A = total weight of air to extract = aQ

Q - weight of vapour to be condensed, in kg/h .

The volumetric efficiency of the pump must be taken into account (this is taken into account in the tables for capacities of Neyrpic Hydro and Nash pumps (Table 40.18)).

Example. Required to determine capacity of air pump to serve a vacuum pan under the following conditions (discontinuous pan):

Pan for a m.c. at 60 m³ (1700 cu.ft.) and 360 m² (3.875 sq.ft.)

Vacuum desired, 66 cm (26 in.)

Temperature of water to the condenser, 30°C

Temperature of water to the pump, 27°C

Calculation. At the beginning of the strike, the pan gives (Table 40.8) 70 kg of vapour per m² of heating surface, or:

$$360 \times 70 = 25,200 \text{ kg/h}$$

Weight of air to extract (Table 40.16):

$$25,200 \times 6 \text{ g/kg} = 151,200 \text{ g/h}$$

Temperature of air at entry to the pump:

$$t_a = t_1 = 30^\circ\text{C}$$

Pressure of air at that point (formula 40.15, Table 32.1):

$$p_a = p - p_r = 0.136 - 0.0435 = 0.0925 \text{ kg/cm}^2$$

Specific volume of this air:

$$v = 0.002927 \times \frac{273 + 30}{0.0925} = 9.6 \text{ l/g} \quad \left(v = 0.3697 \times \frac{459 + 95}{1.273} = 161 \text{ cu.ft./lb} \right)$$

Volume of air to extract:

$$V = 151,200 \times 9.6 = 1,451,520 \text{ l/hr.} = 24.2 \text{ m}^3/\text{min.}$$

A B.13 pump at 66 cm handles 32 mVmin (1.130 cu.ft./min). With the 3° lower temperature of water to the pump, (Table 40.14): 32 x 1.067 = 34 mVmin (1.200 cu.ft./min).

Similarly, a Nash 2000 (75 h.p.) which handles: 26 x 1.067 = 27.7 mVmin (982 cu.ft./min).

Measurement of air leakages

The value of leakages of air into vessels under vacuum may readily be ascertained, by working in the following manner.

The total volume of vessels under vacuum is calculated, down to the level of the water in the barometric column, and including the total interior volume of any pans which are connected. The incondensable gas valves on the pipe lines are carefully closed. Vacuum is raised, then

the air pump is shut down. The time is then determined for the vacuum to fall by 5 - 10 cm (2 - 4 in.), for example.

Knowing the total volume, and the absolute pressure at the beginning and end of this period, the weight of air at the start and finish is readily deduced, and hence the weight of air entering per minute.

It is known that the weight of air entering is independent of the pressure in the vessel, provided that this pressure does not reach:

$$p_c = 0.528 P \tag{40.16}$$

p_c = critical downstream pressure

P = upstream pressure = atmospheric pressure.

Or:

$$p_c = 0.528 \times 76 = 40 \text{ cm (16 in.)} \tag{40.17}$$

In other words, the quantity of air passing through any leak or orifice is independent of the downstream pressure p and depends only on the upstream pressure P , provided the downstream pressure remains lower than $p_c = 0.528 P$. In the case which concerns us here, so long as the vacuum remains greater than:

$$H_c = 76 - 40 = 36 \text{ cm (14 in.)} \tag{40.18}$$

Example. Assuming a vessel of capacity $V = 50 \text{ m}^3$ (2,000 cu.ft.), in which the vacuum takes 36 min to drop from 60 to 50 cm (24 - 20 in.).

The pressure (Table 32.1) corresponding to a vacuum of:

60 cm (24 in.) is $p_a = 0.218 \text{ kg/cm}^2$ (2.91 p.s.i.)

50 cm (20 in.) is $p_a = 0.354 \text{ kg/cm}^2$ (4.87 p.s.i.)

If the temperature of the air has been read with a thermometer at the beginning and of this 36-min period, and has been found to be equal to 20 and 25°C (70 and 80°F), respectively, the specific volume of the air at the beginning (v_1) and at the end (v_2) of this period:

$$v_1 = 0.002927 \frac{273 + 20}{0.218} = 3.93 \text{ m}^3/\text{kg} \text{ (67.2 cu.ft./lb.)}$$

$$v_2 = 0.002927 \frac{273 + 25}{0.354} = 2.46 \text{ m}^3/\text{kg} \text{ (40.9 cu.ft./lb.)}$$

Hence the weight of air contained in the vessel at the beginning and end of the 36-min period:

$$P_a = \frac{50}{3.93} = 12.7 \text{ kg (29.8 lb.)}$$

$$P_z = \frac{V}{v_z} = \frac{50}{2.46} = 20.3 \text{ kg (48.9 lb.)}$$

$$P_z - P_a = 7.6 \text{ kg (19.1 lb.)}$$

The leakage then amounts to:

$$7.6 \times \frac{60}{36} = 12.7 \text{ kg/h} \quad (32 \text{ lb./h})$$

THE BAROMETRIC COLUMN

There are two dimensions of importance in the barometric column (Fig. 40.16).

- (a) Its height
- (b) Its cross-section.

Height of the barometric column. The height of the barometric column should be reckoned from the level of the Torricellian well at the foot of the column up to the junction with the body of the condenser.

To facilitate the entry of the water into the column, Hausbrand¹⁰ recommends that it be connected to the condenser by a conical junction-piece of 30° angle, that is, making an angle of 60° with the horizontal; we have seen (p. 859) that Ziegler recommends at least 70°. This cone is not necessary if the condenser body itself terminates in a definite conical section.

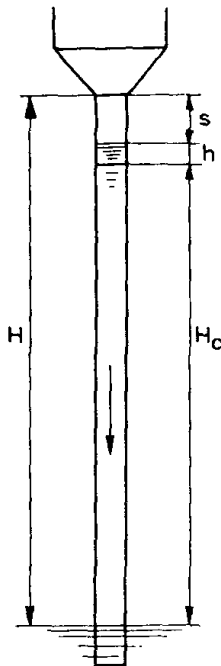


Fig. 40.16. Height of the barometric column.

The height of the column represents the sum of 3 terms:

- (a) H_0 = head of water corresponding to the maximum vacuum in the condenser
- (b) h = head necessary to give the desired downward velocity of the water
- (c) s = margin of safety as provision for abrupt variations in level of the water in the column.

$$H = H_0 + h + s \tag{40.19}$$

(7) H_0 . We may take as lower limit for the temperature of the outgoing water a minimum of 30°C (86°F). The corresponding vacuum is 73 cm (28.7 in.), which is thus a maximum in the tropics. Taking into account the specific volume of the water at that temperature (1.004) and the margin necessary to allow for a possible rise in barometric pressure to 78 cm (30.7 in.), we shall have:

$$H_0 = 10.33 \times \frac{73}{76} \times 1.004 \times \frac{78}{76} = 10.20 \text{ m (33.3 ft.)}$$

(2) h . The head is given by

$$h = (1 + \alpha) \frac{V^2}{2g} \tag{40.20}$$

h = head of water, in m (ft.), necessary to maintain flow in the column, at velocity V

V = velocity of flow in the column, in m/s (ft./s)

g = 9.8 m/s² (32.16 ft./s²)

α = coefficient, values of which, after Hausbrand¹¹, are given in Table 40.19.

TABLE 40.19

VALUE OF COEFFICIENT α

Column	Diameter of column, mm (in.)				
	50 (2)	100 (4)	200 (8)	400 (16)	800 (32)
Straight	6.1	2.6	1.2	0.6	0.4
With double 90° bend at base	6.7	3.2	1.8	1.2	1

(3) s . This is a simple margin of safety as provision for sudden variations in vacuum or in flow of water. We may take in general:

$$s = 0.5 \text{ m (1.5 ft.)} \tag{40.21}$$

Example. To calculate the height of a barometric column for:

V = velocity of flow of water 1 m/s (3.0 ft./s)

D = interior diameter of the column 25 cm (10 in.)

assuming that there is no curve at the bottom of this column.

We have

- (1) $H_0 = 10.20 \text{ m} \quad 33.3 \text{ ft.}$
 - (2) $h = 0.10 \text{ m} \quad 0.3 \text{ ft.}$
 - (3) $s = 0.50 \text{ m} \quad 1.5 \text{ ft.}$
- Whence: $H = 10.80 \text{ m} \quad 35.1 \text{ ft.}$

It is obviously permissible to modify the margin 5 in the case where the lower portion of the condenser may be allowed to fill with water without any disadvantage.

Cross-section of the column. The cross-section of the barometric column requires simply to have sufficient cross-section to assure discharge of the water.

The water to be discharged consists of:

- (a) The cooling water supplied to the condenser
- (b) The vapour which it has condensed

or:

$$(W + 1) \text{ kg/kg (lb./lb.) vapour condensed}$$

Previously, a velocity of approximately 1 m/s (3 ft./s) was general, and rates greater than 1.5 – 1.8 m/s (5 – 6 ft./s) were avoided. However, Webre and Robinson¹² considered that barometric columns were generally much too large, and saw no advantage in choosing velocities of flow lower than 2 – 3 m/s (7 – 9 ft./s). By adopting such values, there is obtained in effect the entrainment of a certain proportion of air, so decreasing the work required of the air pump in a proportion which is quite appreciable.

Moreover, air sometimes accumulates at the bottom of the condenser, forming a pocket which is detrimental to exchange of heat between vapour and water. The higher velocity, which entrains part of the air, avoids this drawback.

The diameter of the column is given by:

$$\frac{\pi D^2}{4} \times \frac{Q(W + 1)}{3,600} = \frac{V}{62.4} \times 3,600 \tag{40.22}$$

D = diameter of the column, in dm (ft.)

V = velocity of flow of water in the column, in dm/s (ft./s)

Q = weight of vapour to be condensed, in kg/h (lb./h)

W = ratio of weight of injection water to weight of vapour condensed, given by eqn. (40.5).

Hence:

$$D = \sqrt{\frac{Q(W + 1)}{2827 V}} = \left(\frac{Q(W + 1)}{1226 V} \right)^{1/2} \tag{40.23}$$

The well at the foot of the column should have a minimum volume of $1\frac{1}{2}$ times the volume of the column.

WATER PUMPS

In general, to serve the condenser, two water pumps are required:

(a) The injection water pump, delivering water from the cooling system to the top of the condenser

(b) The hot water pump, taking water from the Torricellian well below the condenser to the cooling system.

Capacity

We already know the capacity necessary for these pumps:

(a) Injection water pump:

$$Q_i = QW \tag{40.24}$$

(b) Warm water pump:

$$Q_2 = Q(W + 1) \tag{40.25}$$

Q_1 = flow of injection water, in l/h (lb./h)

Q_2 — flow of warm water, in l/h (lb./h)

Q - weight of vapour to be condensed, in kg/h (lb./h)

W = ratio of weight of cold water to weight of vapour to be condensed, given by eqn. (40.5) or (40.6).

Delivery head

The delivery head of which the pump must be capable in normal operation at full discharge is readily calculated.

Example. (a) Injection water pump.

Gravity head at delivery, for example:	15 m	50 ft.
Deduct: Vacuum of 66 cm x 13.6:	9 m	30 ft.
	6 m	20 ft.
Add: Maximum suction, for example:	1.5 m	5 ft.
Losses of head (to be calculated), for example:	1.5 m	5 ft.
Margin of 20%:	2 m	6 ft.
	11 m	36 ft.

The losses of head are calculated by eqn. (45.3).

The injection water pump should be able to deliver water when first started up, when vacuum has not been established. In the preceding example:

Gravity head:	15	m	50	ft.
Add: Maximum suction:	1.5	m	5	ft.
Losses of head:	0.5	m	1	ft.
	17	m	56	ft.

The characteristic for the pump $H = f(Q_1)$ should then in this case begin at: $H = 18$ m (60 ft.) for $Q_1 = 0$ and should give $H = 11 - 12$ m (36-40 ft.) for the normal output Q_1 given by eqn. (40.24).

(b) *Warm water pump.* The operation of the sprays requires a certain head at the nozzles, generally 5 – 6 m (16 — 20 ft.). For example:

Head required:	6	m	20	ft.
Add: Vertical height of the nozzles above the pump, for example:	4	m	13	ft.
Suction from the condenser well to the pump, for example:	1	m	3	ft.
Loss of head between pump and nozzle (to be calculated), for example:	2	m	6	ft.
	13	m	42	ft.
Margin of 20%:	2.5	m	8	ft.
	15.5	m	50	ft.

The characteristic curves for such pumps, giving H as a function of capacity Q_1 , generally show a distinct drop in head as discharge increases (Fig. 40.17).

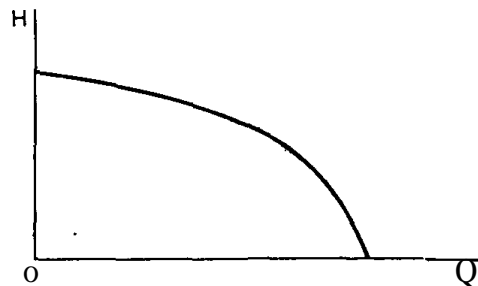


Fig. 40.17. Graph of operation of condenser water pump.

Efficiency

These large pumps generally have an efficiency of the order of 75 – 80%.

Their power requirement is given by:

$$P = \frac{HQ_1}{\eta} \quad \left(\frac{HQ_1}{550 \eta} \right) \quad (40.26)$$

- P = power consumed by the pump, in h.p.
- H = delivery head, in m (ft.) of water
- Q_1 = output, in l/s (lb./s)
- η = efficiency of the pump.

SPRAY PONDS

Factories which have an ample supply of water can allow the warm water from the condenser to run to waste. Accordingly, they require only one pump for the condenser: the injection water pump. Again, certain factories receive the cooling water at a sufficient head to feed the condenser directly, or have a reservoir at sufficient head above the condensers. These are particularly fortunate conditions, but infrequent.

Quantity of water required for condensers

The minimum quantity Q_1 of cold water required for condensation amounts to:

$$Q_1 = QW \quad (40.27)$$

- Q_1 = minimum quantity of cold water required for condensers, in l/h (lb./h)
- Q = weight of vapour to be condensed in the factory, in kg/h (lb./h)
- W = ratio of weight of cooling water to weight of vapour to be condensed.

Or, from eqns. (40.6) and (40.9), and assuming counter-current condensers are used:

$$Q_1 = \frac{572 Q}{t_2 - t_1} = 0.88 \frac{572 Q}{(t - t_1)} = \frac{650 Q}{t_v - t_1} \quad \left(\frac{1,170 Q}{t_v - t_1} \right) \quad (40.28)$$

t_v = temperature corresponding to the vacuum, in °C (°F)

t_1 = temperature of cooling water available, in °C (°F).

In general a vacuum of 65 cm (25.5 in.) is considered sufficient, hence:

$t_v = 54^\circ\text{C}$ (129°F); and in the common case where the water comes from mountain streams or from deep flowing rivers, we have frequently, in tropical countries: $t_1 = 21^\circ\text{C}$ (70°F). We have then:

$$Q_1 = \frac{650 Q}{54 - 21} = 20 Q \quad (40.29)$$

Table 40.6 and eqn. (40.4) then allow for a ready determination of the minimum quantity of water required per tonne of cane. With 300 kg (672 lb.) of vapour to be condensed per tonne of cane, it will be, for example:

$$Q_1 = 20 \times 300 = 6,000 \text{ l/t.c.} \quad (40.30)$$

To allow a comfortable margin and to provide for momentary variations, it would be necessary to allow:

(a) with central vacuum system:

$$Q_1 = 30 Q \quad (40.31)$$

or in the case chosen:

$$Q_1 = 10,000 \text{ l/t.c. (20,000 lb./t.c.)} \quad (40.32)$$

(b) with individual condensers:

$$Q_1 = 60 Q \quad (40.33)$$

or in the case chosen:

$$Q_1 = 20,000 \text{ l/t.c. (40,000 lb./t.c.)} \quad (40.34)$$

Cooling systems

If the required quantity of water is not available from spring or river water (or from the sea, in the case of factories situated on the coast), it is necessary to resort to a cooling system to which the warm water leaving the condenser is pumped, and which cools it for re-use.

Types of cooling system

There are three principal types of cooling system:

(a) The spray pond, in which the warm water is broken up into a spray by means of nozzles.

(b) The cooling tower, generally packed with brushwood or wooden laths. This is a very large tower in which the hot water encounters a natural draught which promotes cooling of the water by convection and evaporation. Roughly, we may reckon that 10 - 15% of the heat removed is by convection, and 85 - 90% by evaporation. Generally, there will be about 1 kg of water evaporated per 600 kcal removed (1 lb of water/1080 B.T.U.). This system is little utilised in the sugar factory.

(c) Forced draught cooling tower, which follows the same principle, but with draught provided by fans.

The spray pond (Fig. 40.18) consists simply of a large flat basin above which the warm water is distributed by pipes on which the spray nozzles are arranged at regular intervals. When the water is thus brought into intimate contact with the atmosphere, a small portion of it is evaporated. The evaporation and the contact of the ambient air with the fine drops of water produce the required degree of cooling.

Limit of cooling possible

The lowest temperature to which the water can be cooled by contact with the air is the temperature indicated by the wet-bulb thermometer.

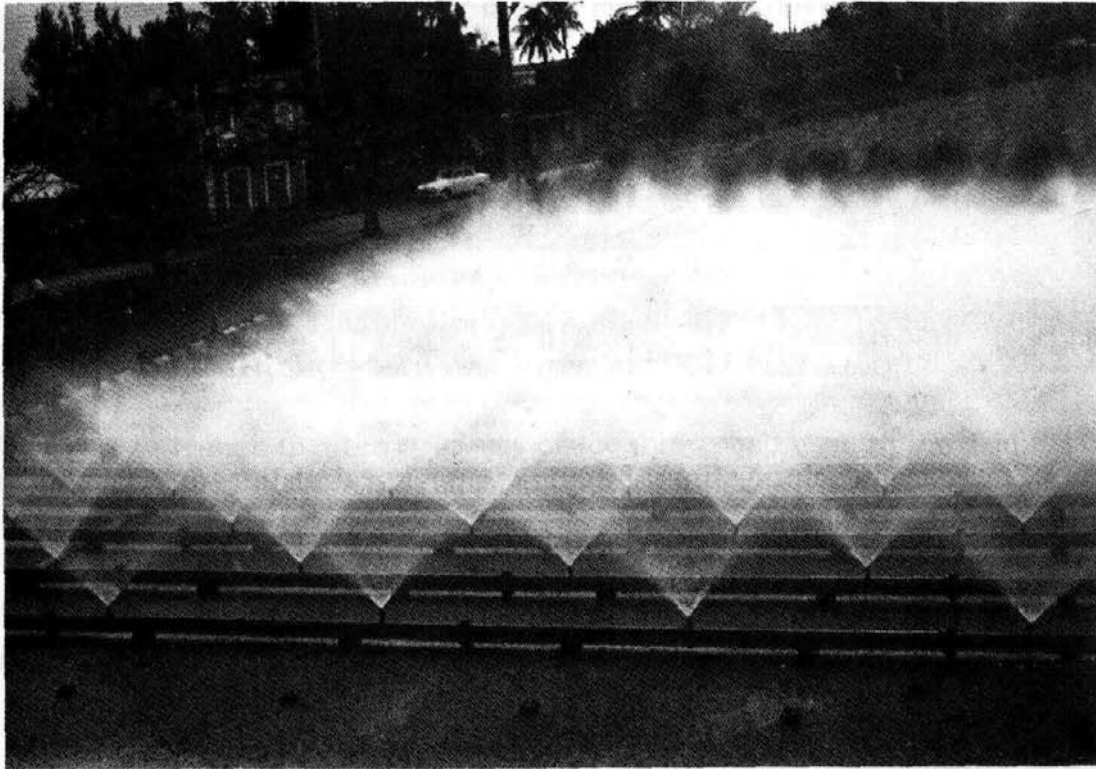


Fig. 40.18. Spray pond.

If this temperature were reached, we should have an efficiency of 100% in the cooling. Thus the efficiency e of the cooling system may be defined as:

$$e = \frac{t_2 - t_1}{t_2 - t_0} \quad (40.35)$$

t_2 = temperature of the warm water entering the spray pond

t_1 = temperature of the cold water leaving the spray pond

t_0 = wet-bulb temperature.

Generally, efficiencies range from 50 to 70%, averaging 60%.

Since the use of the wet-bulb thermometer is not widespread, it will be remembered more simply that a good spray pond should cool the water within 1 or 2°C of the temperature of the ambient air, provided that it has been very generously designed.

Spray pond

Depth. The spray pond itself should have a depth of approximately 1 m (3 ft.), as recommended, for example, in Australia¹³.

There is no advantage in exceeding a depth of 1.1 m (3 ft. 6 in.), since the increase in the mass of water in the circuit has only a negligible influence on the cooling. The surface area only is important.

Area of the pond. The area of the pond is important on account of the necessity of arrang-

ing the nozzles so that the curtains of water formed by them do not overlap or interfere with each other, and so that air may circulate between the sprays.

The area of the pond is deduced from the quantity of water which it can treat per hour per unit area of the pond. **Tromp**¹⁴ suggests 120 lb./sq.ft./h, **Webre and Robinson**¹⁵ 150 lb./sq.ft./h. Other figures given are in kg/m²/h (lb./sq.ft./h):

	<i>Minimum</i>	<i>Maximum</i>	<i>Mean</i>
Hawaii ¹⁶	700 (145)	820(168)	760(156)
Queensland ¹³	650(133)	825(170)	740 (152)

Among French manufacturing firms, **Fives Lille** bases its calculations on 800 kg/mVh (164 lb./sq.ft./h).

We consider that the best value to adopt is:

$$\frac{Q_2}{S} = 750 \text{ kg/mVh (154 lb./sq.ft./h)} \quad (40.36)$$

Q_2 = weight of hot water to be treated, in kg/h (lb./h), given by eqn. (40.25)

S - area of the pond, in m² (sq.ft.), not including the excess of the margins of safety (40.38) over the half intervals a and b (40.37).

There is no advantage in adopting an excessive surface area for the pond^{17a}. The water in the pond is heated by the sun, especially in regions where it is not protected by the cones of the sprays, and this solar heating is proportional to the exposed surface.

Disposition of the nozzles. A system of piping is generally arranged above the pond, with diameters constant or progressively decreasing; on these pipes are placed the nozzles in a staggered arrangement (Fig. 40.19).

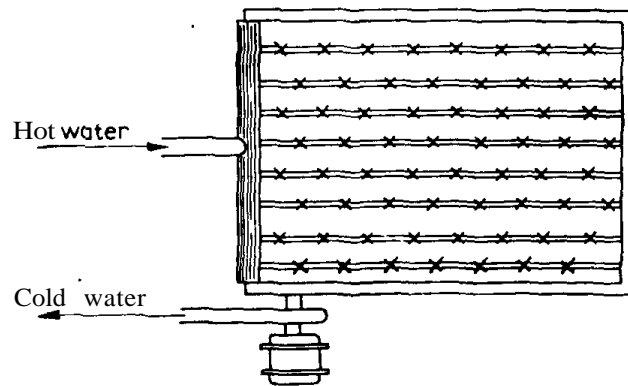


Fig. 40.19. Spray pond. Arrangement of nozzles.

The spacing between pipes, and distance between nozzles on each pipe, will depend on the size of the cone of water given by each nozzle. With ordinary types, we may take approximately:

$$\text{and } \left. \begin{aligned} a &= 2.5 - 4 \text{ m (8 - 13 ft.) between pipes} \\ b &= 2-2.5 \text{ m (7 - 8 ft.) between nozzles on each pipe} \end{aligned} \right\} (40.37)$$

The curbstone of the low wall enclosing the pond should be sloped towards the inner side. A sufficient distance between the peripheral nozzles and the outer edge of the wall will be arranged as follows:

$$\left. \begin{aligned} \text{Windward side} & 3-4 \text{ m (10 - 13 ft.)} \\ \text{Leeward side} & 6-8 \text{ m (20 - 25 ft.)} \end{aligned} \right\} (40.38)$$

It is relevant to remark that the efficiency of the spray pond is much lower in the centre than at the sides. For this reason, it is advantageous to make the pond long and narrow; this obviously increases the cost.

Nozzles. Types. There are many types of nozzle. Most of them aim to give a water spray the form of a hollow cone, either by forming this cone at the start by the shape of the nozzle, or by giving to the water a centrifugal or turbulent movement.

In Queensland^{17b}, comparisons have been made between three types of nozzle: conical jet, vortex and evolule types, and the conical jet nozzles have been found superior, as regards capacity and efficiency.

Figure 40.20 shows a good design of nozzle.

A good spray nozzle should be of simple design, readily dismantled, and should not have any portion with a cross-section smaller than that of the outlet orifice, which might catch foreign bodies in the water and block the nozzle. Alternatively, such a portion should be plac-

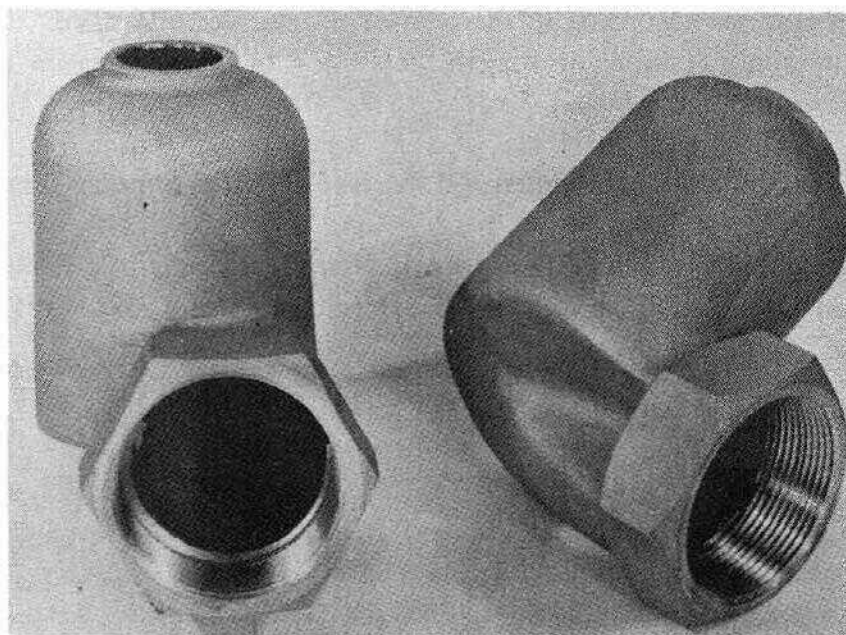


Fig. 40.20. Nozzle For spray pond (A. Gauthier).

ed in such a way that the foreign body can be readily removed by quickly dismantling the nozzle.

Pressure. The pressure generally utilised at the nozzles varies from 400 to 700 g/cm² (6–10 p.s.i.), say 4–7 m (13–23 ft.) of water. In Queensland^{17b} a pressure of 10 p.s.i. has been recommended; Tromp¹⁴ estimates that the optimum pressure is 500 g/cm² (7 p.s.i.) or 5 m (16 ft.) of water, and such a figure is generally favoured.

Capacity. The quantity of water delivered by a nozzle is given by:

$$q = 3,600 \frac{\sqrt{2gH}}{10,000} \quad \left(\frac{3,600}{144} \sqrt{2gH} \right)$$

or:

$$q = 0.36cs \sqrt{2gH} \quad (25 \text{ cs } \sqrt{2gH}) \quad (40.39)$$

q — capacity of the nozzle, in m³/h (cu.ft./h)

c = contraction coefficient

s = cross-section of the delivery orifice of the nozzle, in cm² (sq.in.)

g = 9.8 m/s² (32.16 ft./s²)

H - pressure of water at the nozzle, in m (ft.) of water.

Or again:

$$q = 1.6 \text{ cs } \sqrt{H} \quad (200 \text{ cs } \sqrt{H}) \quad (40.40)$$

in the same units.

The coefficient c varies according to the type of nozzle. We may take:

(a) Nozzles giving jets of water normal to the section of the orifice $c = 0.5 - 0.7$

(b) Nozzles of vortex type $c = 0.4 - 0.5$.

Nozzles with unit capacity of the order of 5 mVh (170 cu.ft./h) under a head of 5 m (16 ft.) are generally used.

We may comment that the area of the spray pond corresponding to one nozzle should be proportional to its capacity, if the ratio Q_2/S determined by eqn. (40.36) is used, say:

$$\frac{Q_2}{S} = \frac{q}{ab} = 750 \text{ kg/m}^2/\text{h} \text{ (154 lb./sq.ft./h)} \quad (40.41)$$

Hence:

$$ab = \frac{q}{750} \quad \left(\frac{q}{154} \right) \quad (40.42)$$

a = spacing between the pipes carrying the nozzles, in m (ft.)

b = distance between nozzles on each pipe, in m (ft.)
 q = output of each nozzle, in kg/h (lb./h).

Number of nozzles. The number N of nozzles is obtained immediately:

$$N = \frac{Q_2}{q} \tag{40.43}$$

N = number of nozzles for the spray pond
 Q_2 = quantity of warm water to be cooled, in mVh (cu.ft./h or lb./h), given by eqn. (40.25)
 q = capacity of warm water of each nozzle, in mVh (cu.ft./h or lb./h).

Loss of water. A small proportion of the water is evaporated due to the breaking up of the water into drops by the sprays. Further, the wind carries away some of the fine drops thrown up.

Leading authors attribute to these losses the values given in Table 40.20.

TABLE 40.20
 LOSS OF WATER FROM SPRAY POND (%)

	<i>Evaporation</i>	<i>Entrainment</i>	<i>Total</i>
Deerr ¹⁸			3-5
Webre and Robinson ¹⁹			3
Tromp ²⁰	2.6	3	5.6

The total loss should be of the order of 3-5%, according to the strength of the wind. Now, if 3 – 5% of the water leaves the circuit, during the cooling, there returns to the condenser:

$$\frac{100}{W} \%$$

since W kg of cold water condense 1 kg of vapour from the juice.

The nett loss is then:

$$\epsilon = \varphi - \frac{100}{W} \tag{40.44}$$

e = nett loss of water, % of water in circuit Q_2
 φ = approximate total loss by evaporation and entrainment, % of Q_2
 W = ratio between weight of cold water and weight of vapour to be condensed.

Since the quantity of water in circuit must not diminish, it is therefore necessary to add to the circuit a make-up quantity equal to the nett loss e (Fig. 40.21).

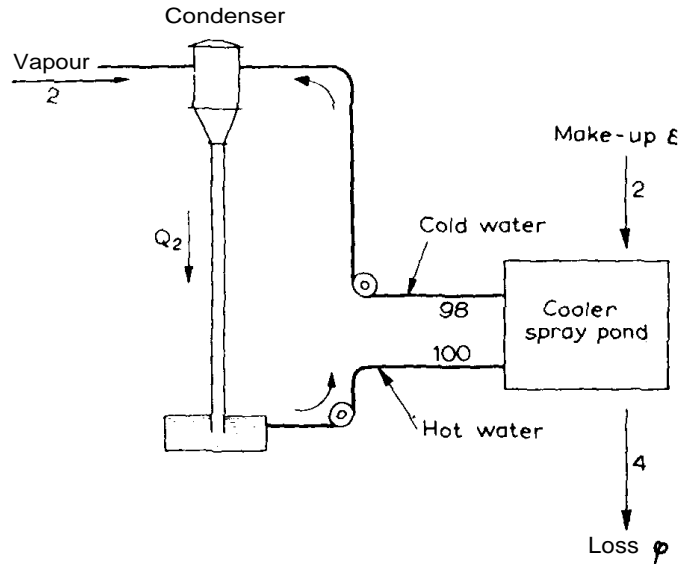


Fig. 40.21. Condenser water cycle.

We may comment that if it were possible to reduce the gross loss in the cooling system to the value $1/W$, the make-up would become unnecessary; say:

$$\varphi_0 = \frac{100}{W} \quad (40.45)$$

Unfortunately, it is difficult to achieve this, even with a good spray pond and a low vacuum. For some factories at which the supply of cold water falls to very low levels during the dry season, it is necessary to make every effort to approach this goal. The best method is to increase the safety margin at the wall of the pond on the leeward side (40.38) and to block up carefully all leaks which may appear. It is also necessary to take care that other waters in the factory (cooling water at mill bearings, or water used in the oil cooler of the turbo set; condensate from later effects, etc.) should not be sent to the drain or wasted.

The quantity of make-up water necessary will be:

$$E = \frac{QW\epsilon}{100} \quad (40.46)$$

Example. With:

$Q = 300$ kg (672 pounds) of vapour to be condensed per tonne of cane

$W = 50$

$\epsilon = 2\%$

we shall have:

$$E = 300 \text{ l/t.c. (672 lb./t.c.)}$$

If, as in this case, $W\epsilon/100 = 1$, we shall have $E \approx Q$. That is, the make-up water would be approximately equal to the weight of vapour to be condensed.

Forced-draught cooling towers

These take the form of a large rectangular concrete structure. The water is pumped to the top and broken up into sprays; it falls as rain on to successive stages of boards in the form of louvres of wood, polystyrene or metal, with notches and ribs to break up the flow. The water flows from one stage to the next and finally arrives at the pond which forms the base of the structure, from which it is taken by the cold water pump.

The efficiency of this device depends essentially on climatic conditions, in particular the wet-bulb temperature and the relative humidity of the ambient air. The quantity of water required is calculated as indicated on p. 864, formulae (40.5) and (40.6). It is of the order of 10 – 15 m³/t.c.h.

The wet-bulb thermometer indicates, according to atmospheric humidity, 3 – 5°C lower than the dry-bulb temperature. Calculations are made on the assumption of 75 - 80% relative humidity. The drop in temperature of the water is approximately 12 – 15°C.

The fans are of axial flow type of large diameter (3 — 8 m in general: 10 — 26 ft.). The cooling assembly is divided into several sections (2 – 6 generally). Power consumption is approximately 75 - 110 kW/1000 m³/h (2 – 3 kW/cu.ft.) according to the heat quantities involved, or 5-6 kW/million kcal (1.25 – 1.50 kW/million B.T.U.).

Losses of water are approximately 0.5 kg/tonne of circulated water by entrainment, and 2-3.5% by evaporation. The approach (difference between the temperature of the cold water and the wet-bulb temperature) may be as low as 2.5°C. French makers of this equipment are Hamon, Scam, and Jacir.

The advantage of this cooling equipment is its relatively low space requirement, compared with spray ponds. The disadvantage is the power consumption, roughly triple that of spray ponds, when water pumps are included.

Water requirements of the factory

Since the condenser water is one of the large water requirements of the sugar factory, this is a convenient point to discuss the water requirements in general^{21a}.

(1) <i>Boilers.</i> Taking the total steam requirements as 450 kg/t.c. and expressing the capacity of the factory in t.c.h. as A. Water for the boiler plant must be assumed as 10 – 15%, normally 10%. We have then	0.05 A
(2) <i>Washing of cane.</i> Depends on conditions	
(3) <i>Cooling mill bearings;</i> assuming 5 mills, say	0.20 A
(4) <i>Imbibition.</i> Assuming 30%	0.30 A
(5) <i>Filtration.</i> Assuming water 100% on cake (p. 482)	0.04 A
(6) <i>Cooling A massecuites:</i> with a Werkspoor, we may reckon 1.5 1/1 of m.c. and 130 1/1 of m.c./t.c. = 0.130 × 1.5	0.20 A
<i>Cooling B m.c.</i> Similarly: 0.060 × 1.5	0.09 A
<i>Cooling C m.c.</i> Similarly: 0.040 × 1.5	0.06 A
Movement water for pans	0.02 A
(7) <i>Water for coolers:</i> See above	0.30 A
(8) <i>Washing at centrifugals</i>	0.01 A

	1.27 A

RAIN TYPE CONDENSERS

These are condensers^{21b,c} in which the water enters at the top on to a perforated plate which is furnished with a flanged edge. The size of the holes is not of great importance: a diameter of 15 mm may be adopted. The water then falls as rain through the whole height of the condenser. The vapour to be condensed arrives at the bottom. It is necessary that the lateral flange on the plate, which forms a bowl, should also be perforated so that the water flows in lateral jets which strike the wall of the condenser. A good proportion of the condensation is actually effected on the wall. The openings in the plate should indeed be calculated so that, when the condenser is working at full capacity, there remains an excess of water overflowing the flange of the plate.

The annular space between the empty plate and the wall of the condenser should be sufficient to allow the full flow of water to pass, in case the holes become totally obstructed. Otherwise, water would enter into the vapour pipe.

It is very important to allow sufficient height between the entry plate and the top of the vapour-entry pipe. A minimum of 1.8 m is recommended, though 1.5 m may sometimes be sufficient; however, a height of 2.40 m is generally adopted.

The water should enter axially at the top of the condenser. The central part of the plate should remain unperforated, for a diameter 10 cm greater than that of the water pipe. It is equally important to arrange the incondensable-gas pipe at the top of the condenser.

The total area of the holes should be three times that of the water inlet pipe. The pipe should be placed near the centre of the condenser, and its area should be equal to 0.16% of the cross-section of the condenser, thus its diameter some 6% that of the condenser. This type of condenser permits of an approach of the order of 2–5°C, generally 3°C.

A number of tubes will be placed passing through the plate, and serving for the passage of incondensables. Their total cross-section should exceed that of the incondensable outlet pipe. The cross-section of the condenser should be 0.1 — 0.2 m²/t/h of vapour to be condensed.

CONDENSER-HEATERS

The raintype of condenser may be used for heating juices and so replacing the tubular heaters previously described (Chapter 30). It is essentially a matter of feeding juice to the condensers instead of cold water; the juice must be screened very thoroughly so that particles of solid matter do not obstruct the perforations of the condenser plate. This system has several advantages:

- (a) It eliminates the tubular heaters, which are much more expensive than the cylindrical condenser bodies.
- (b) It eliminates the cleaning of heater tubes, which is expensive both in labour and tools.
- (c) It reduces the quantities of hot water to be sent to the coolers, which involve two large pumps, large piping and a large cooler.

On the other hand, it has a disadvantage: it greatly increases the volume of juice to be evaporated, which is increased by the quantity of vapour condensed in the condenser-heaters.

It is appropriate in this connection, to note that their efficiency is 100%, as against 95% for tubular heaters.

We give an interesting example of calculations for evaporation with heating of the juices by condenser-heaters. Consider a factory to be constructed, very modern, of 100 t.c.h. capacity with mixed juice 100% on cane, carrying out all its heating with vapour from the juice, including vacuum pans, and equipped with an evaporator set composed of:

1st vessel with downward flow:	giving vapour of: 130°C
2nd vessel ordinary:	giving vapour of: 118°C
3rd vessel:	giving vapour of: 105°C
4th vessel:	giving vapour of: 85°C

with the following scale of heating by condenser-heaters:

No. 1 on 4th vessel:	30- 60°C
No. 2 on 3rd vessel:	60- 80°C
No. 3 on 2nd vessel:	80 - 100°C
No. 4 on 1st vessel:	100 - 120°C

The massecurtes being heated as follows:

A m.c.	140 l/t.c.	X 1.47	x 0.275	X 1.2	= 68 kg/t.c.	= 6,800 kg/h from 2nd V.
B m.c.	54 l/t.c.	X 1.47	x 0.275	x 1.2	= 26 kg/t.c.	= 2,600 kg/h from 3rd V.
C m.c.	37 l/t.c.	X 1.47	x 0.36	x 1.2	= 23 kg/t.c.	= 2,300 kg/h from 3rd V.

or:

6,800 kg	of vapour from 2nd vessel	(15,232 lb./h)
4,900 kg/h	of vapour from 3rd vessel	(10,976 lb./h)

The quantities of vapour required for condensation in the four condenser-heaters will be:

$$\text{NO. 1. } q_4 = \frac{100,000 \times 0.9 (60 - 30)}{548} = 4\,927 \text{ kg/h}$$

$$\text{No. 2. } q_3 = \frac{100,000 \times 0.9 (80 - 60)}{536} = 3.358 \text{ kg/h}$$

$$\text{No. 3. } q_2 = \frac{100,000 \times 0.9 (100 - 80)}{521} = 3.415 \text{ kg/h}$$

$$\text{No. 4. } q_1 = \frac{100,000 \times 0.9 (120 - 100)}{519} = 3.468 \text{ kg/h}$$

15,168 kg/h (33,977 lb./h)

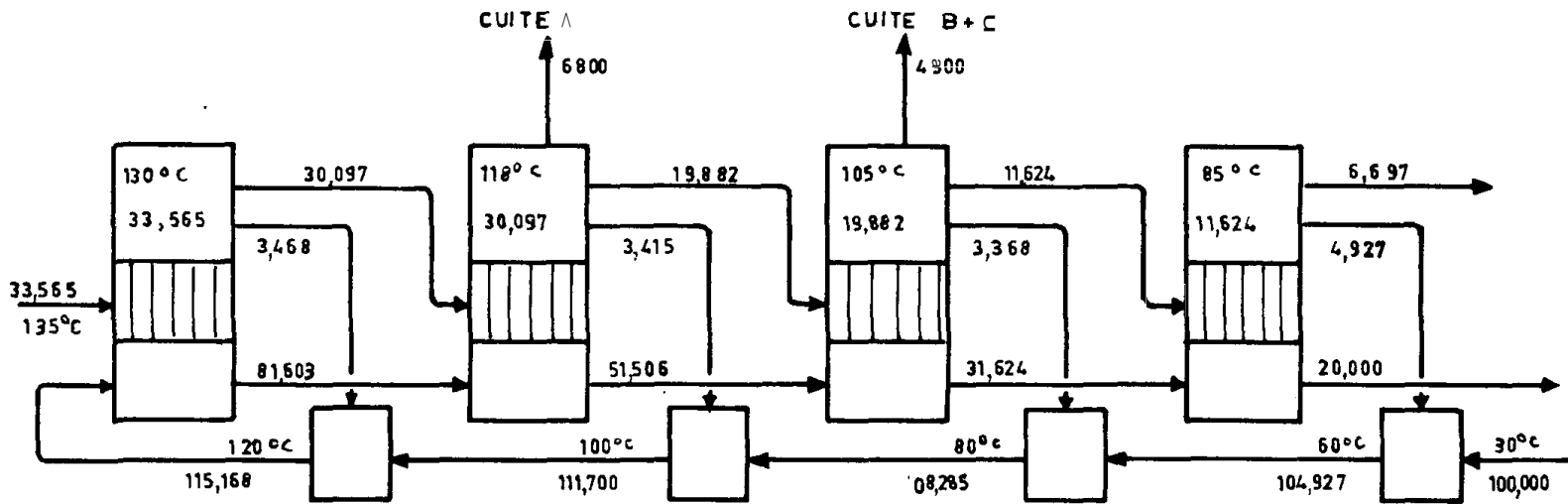


Fig. 40.22. Quadruple effect partly under pressure with bleeding for pans and direct contact heaters (100 t.c.h. factory, metric units).

This quantity of water must be evaporated completely.

Designating by x (Fig. 40.22) the quantity of vapour leaving the last vessel to the general condenser, we have the following disposition of the evaporation:

$$\begin{array}{r}
 \text{4th vessel: } x + 4,927 \\
 \text{3rd vessel: } x + 4,927 + 4,900 + 3,358 \\
 \text{2nd vessel: } x + 4,927 + 4,900 + 3,358 + 6,800 + 3,415 \\
 \text{1st vessel: } x + 4,927 + 4,900 + 3,358 + 6,800 + 3,415 + 3,468 \\
 \hline
 80,000 + 15,168 = 4x + 19,708 + 14,700 + 10,074 + 13,600 + 6,830 + 3,468
 \end{array}$$

$$x = \frac{95,168 - 68,380}{4} = \frac{26,788}{4} \approx 6,697 \text{ kg/h (15,000 lb./h)}$$

Hence the individual evaporations:

$$\begin{array}{r}
 \text{4th vessel: } 6,697 + 4,927 = 11,624 \\
 \text{3rd vessel: } 11,624 + 4,900 + 3,358 = 19,882 \\
 \text{2nd vessel: } 19,882 + 6,800 + 3,415 = 30,097 \\
 \text{1st vessel: } 30,097 + 3,468 = 33,565 \\
 \hline
 95,168 \text{ kg/h (213,176 lb./h)}
 \end{array}$$

The heating by condenser-heaters thus involves the factory in a 19% additional evaporation load, and consequently it requires a multiple effect 19% larger; but it consumes only a total of 336 kg steam per t.c.h., heating and pans included, if a complete solution is adopted such as described above.

JET CONDENSERS

Principle

Jet condensers were first placed on the market by Schutte-Koerting about 1930. They are based on the dynamic effect of jets of water which, penetrating into the body of water in the barometric column, entrain with them, by friction, the air contained in the condenser. If the cross-section of the barometric column is small enough to ensure a suitable velocity, the bubbles of air do not rise into the condenser and are evacuated to the well at the foot of the column.

It is thus possible to operate a condenser system without an air pump. This eliminates an expensive and cumbersome machine, with its operating maintenance costs.

The design of the condenser is similar in all points to that of an ordinary co-current condenser (Fig. 40.23). It is even simpler, due to the elimination of baffles, which would impair the velocity of the jets.

Only the design of the nozzles and the direction of the jets require some care, in order to

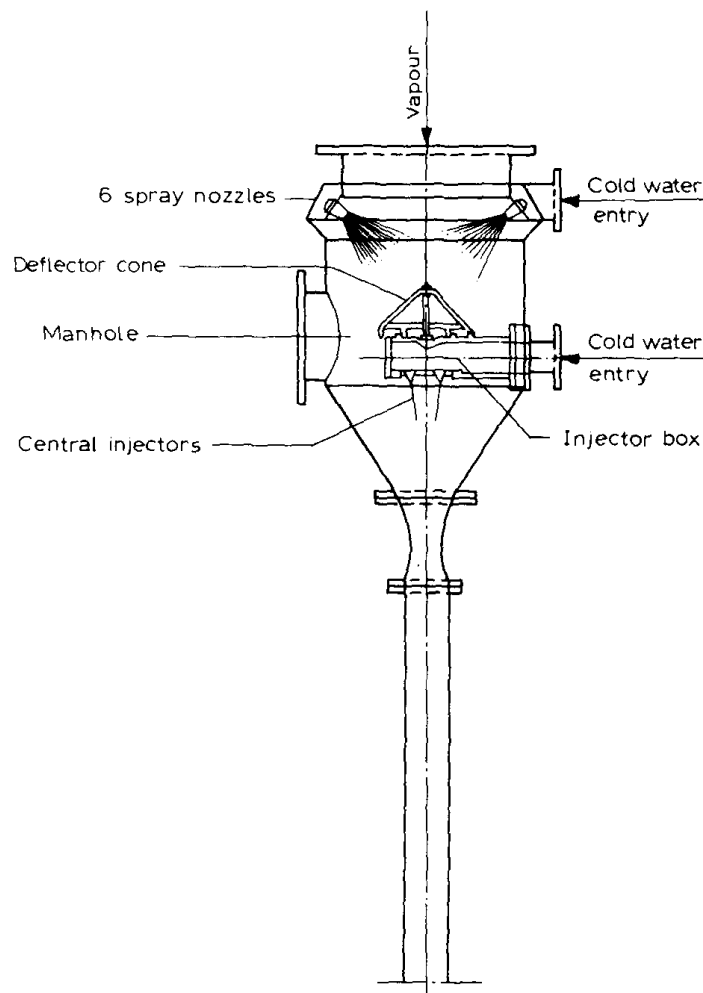


Fig. 40.23. Multi-jet condenser (Fives Cail-Babcock).

obtain clean and smooth jets of water, to ensure the correct direction of the jets, to achieve the necessary entrainment of the air and an effective penetration into the water in the column.

Calculations

Calculations for jet condensers are made exactly as for ordinary condensers (*cf.* p. 857 *et seq.*). The only points of difference are the following:

- (a) **Approach.** The approach is not given by eqn. (40.10). It depends on:
- (1) The magnitude of air leakages. The greater these are, the worse will be the approach.
 - (2) The vacuum desired. The higher the vacuum, the more difficult the approach.
 - (3) The dimensions of the condenser. The larger the condenser, the better the approach.

We may take as approach the figure given by the following empirical formula:

$$\text{App} = 0.3 (10 + R) \frac{79 - H}{76 - H} \frac{4,000 - Q_1}{4,000} \quad (40.47)$$

$$\left(0.25 (22 + R) \frac{31.2 - H}{30 - H} \frac{140,000 - Q_1}{140,000} \right)$$

- App = approach = $t_v - t_2$ = difference between the temperature of the hot water leaving the condenser and that of the vapour entering, in °C (°F)
- R = leakages of air into the vessel served by the condenser, in kg (lb.) air/h
- H = vacuum to be maintained in the condenser, in cm (in.) of mercury
- Q_1 = quantity of cooling water given by eqn. (40.24), but expressed in m³/h (cu.ft./h) (the formula is valid only for $Q_1 \leq 2,000$ mVh (70,000 cu.ft./h)).

The figures for air leakage will be determined by the procedure explained on p. 878. For a normal installation, the following figures may be used:

- R = 10 kg/h (22 lb./h) as normal minimum
- R = 15 kg/h (33 lb./h) as an average value
- R = 20 kg/h (44 lb./h) as a high value

However, leakages appreciably greater are sometimes encountered.

Tromp²² quotes an approach of 4-5°C (8°F) for a jet condenser; more generally⁸, a range of 7-15°C (12-27°F), with a mean of 11°C (20°F) is reported, according to the fluctuations in the quantity of vapour to be condensed.

(b) Water pressure. Jet condensers operate under a relatively low water pressure. This pressure, somewhat higher in older designs, has been reduced to a very low figure in more recent designs. In fact, certain condensers operate with the water reaching the nozzles under a slight vacuum; but this is exceptional.

The water pressure should be a function of the air leakage. It will be prudent to base estimates on the minimum values given in Table 40.21, *i.e.* approximately 0.1 m water/kg/h (0.15 ft./lb./h) of air leakages. These figures are gauge pressures, *i.e.* pressures above atmospheric.

It follows that, when a vacuum exists in the condenser, the water enters the condenser with a pressure for example of:

$$1.5 + 9 = 10.5 \text{ m (35 ft.)}$$

9 m (30 ft.) being the pressure corresponding to 66 cm (26 in.) of mercury.

TABLE 40.21

PRESSURE OF WATER AT NOZZLES OF JET CONDENSERS

	<i>Air leakages</i> kg/h	<i>Pressure at nozzles</i> kg/cm ² m water	
Low	10	0.1	1.0
Moderate	15	0.15	1.5
High	20	0.20	2.0

However, it is necessary to assure the required pressure at starting up, when the vacuum has not yet been established. A value of 400 g/cm² = 4 m (6 p.s.i. = 14 ft.) of water would

be sufficient. The characteristic curve of the injection water pump should therefore permit of a small discharge with a pressure of 4 m (14 ft.) at the top of the condenser. This is the pressure head which should be adopted for specifications of the pump, and which should be added to the geometric height of the nozzles, without allowing for vacuum (*cf.* p. 883).

It will be advisable to provide the suction of the pump with an efficient screen, so that pieces of bagasse or other foreign bodies will not be liable to block the nozzles.

Table 40.21 is applicable for normal temperatures of cooling water. If it is necessary to use cooling water at a temperature of t greater than or equal to 35°C (95°F), it will be advisable to increase the pressure at the nozzles substantially.

(c) **Barometric column.** For ease of construction, the barometric column is generally designed so as to obtain a constant loss of head. According to the formula of Darcy:

$$J = 4b \frac{V^2}{D} \quad (40.48)$$

J = loss of head in a nozzle, *e.g.* in m/m (ft./ft.)

b = coefficient varying slightly with the diameter

V = velocity of water in the nozzle, in m/s (ft./s)

D = diameter of the pipe, in m (ft.).

Neglecting the variations in the coefficient b , and in using for it a mean value of 0.0004 (0.00012), we have:

$$J = 0.0016 \frac{V^2}{D} \quad \left(0.0005 \frac{V^2}{D} \right) \quad (40.49)$$

For jet condensers, we generally take:

$$J = 0.064 \quad \text{or} \quad \frac{V^2}{D} = 40 \quad (130) \quad (40.50)$$

or expressing V in m/s (ft./s) and D in dm (in.):

$$V = 2\sqrt{D} \quad (3.3\sqrt{D}) \quad (40.51)$$

V = apparent velocity, in m/s (ft./s), of the water flowing down the column, neglecting the air entrained with it.

D = interior diameter of the column, in dm (in.).

The velocity will thus reach 2 m/s (6.6 ft./s) in a column of 10 cm (4 in.) diam., and 4 m/s (13.2 ft./s) in one of 40 cm (16 in.) diam. These velocities are sufficient to promote entrainment of the air bubbles, but they should not be allowed to fall below half or two-thirds of these values, otherwise entrainment of the air will not be ensured.

Use of jet condensers

Jet condensers require vessels well protected against air leakages, to which they are much more sensitive than condensers fitted with an air pump. However, when this condition is fulfilled, they function very well, and the elimination of the air pump constitutes an economy.

They will not be employed for a central condensing system on account of the greater extent of air leakages in this case. However, they constitute a good solution with individual condensers.

They are less flexible than condensers with air pumps, as regards the quantity of injection water, which can vary only between appreciably narrower limits. The vacuum which they give is similarly less uniform, and varies slightly with the temperature of the hot water, which in turn depends on the quantity of vapour to be condensed. However, this drawback has no great importance in the case of individual vacuum. They have a high water consumption and require more powerful pumps with consequent high electric power demand.

STEAM-JET EJECTOR CONDENSERS

Apart from jet condensers, there is another method of eliminating the air pump: this consists of evacuating the air by means of an ejector condenser actuated by steam.

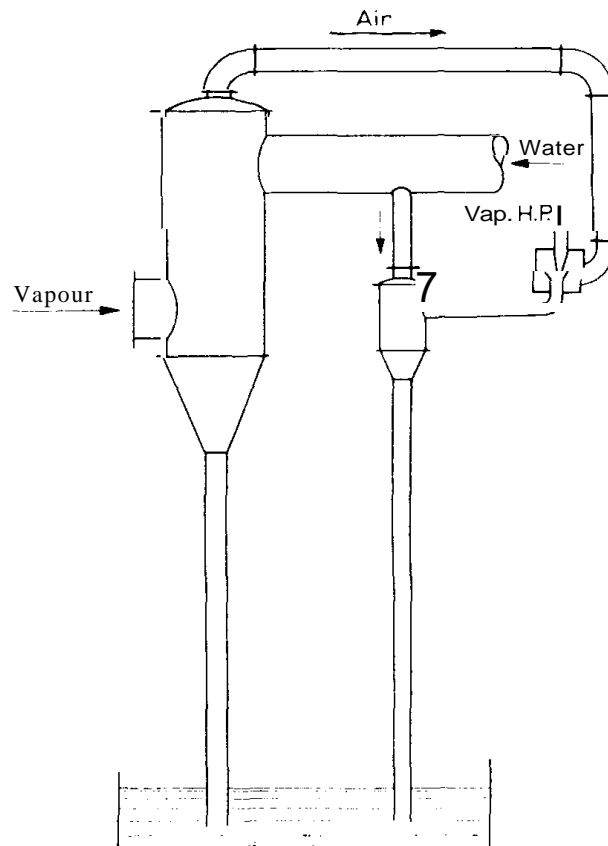


Fig. 40.24. Ejector-condenser.

This device is connected to the air outlet of the condenser (Fig. 40.24). Medium-pressure steam, $5 - 10 \text{ kg/m}^2$ ($70 - 140 \text{ p.s.i.}$), is admitted to a nozzle and entrains the air by friction towards the well at the foot of the column, directly to atmosphere, or to a second small condenser which condenses the steam.

The advantages of this system over the air pump are²³:

- (1) Space occupied is less and practically negligible.
- (2) No foundations are required.
- (3) There are no moving parts. Hence no lubrication, no stuffing boxes, no wear, no parts to replace (other than the nozzle, which is insignificant and lasts several years).
- (4) Cost is much lower, about half.

Moreover, the volumetric efficiency of an air pump decreases rapidly as the vacuum increases, whereas an ejector condenser is designed for a given vacuum, for which its efficiency will be an optimum.

On the other hand, it follows that, at starting up, its efficiency will be low since it has not yet attained the desired vacuum. Hence a long delay before the desired vacuum is attained, at starting up.

This disadvantage is sometimes overcome by the installation of a special ejector for starting up, which is used only for raising vacuum in the pan and which is shut down when the vacuum reaches about 50 cm (20 in.), after starting up the ejector for normal service.

The steam ejector condenser has another disadvantage: when the condenser water is sent to a spray pond or cooling tower, its temperature is increased due to the heat of condensation of the injected steam.

Number of stages

The air may be extracted in two stages: (1) from the large condenser to the small one; (2) from the small condenser to another still smaller. In the latter case it is termed a two-stage ejector.

General practice employs:

(a) Single-stage ejectors for vacua up to 66 cm (26 in.), used for vacuum filters or as starting-up ejectors.

(b) Two-stage ejectors for vacua up to 75 cm (29.5 in.). These are generally used for multiple effects and vacuum pans.

The normal installation thus consists of the first-stage ejector, drawing vapour from the condenser and discharging into a second small condenser, or "inter-condenser"; then the second-stage ejector drawing from the inter-condenser and discharging to atmosphere, preferably by a pipe discharging into the well at the base of the barometric column which thus receives water from the three columns: those of the two condensers and that of the last ejector.

Steam consumption

The steam consumption is proportional to the quantity of air extracted. In order to obtain a suitable steam consumption, it is necessary to use actuating steam at a pressure between 5 and 10 kg/cm^2 (70 and 140 p.s.i.). Steam consumption decreases as the pressure of the actuating steam is increased. Above 10 kg/cm^2 (140 p.s.i.), the saving becomes smaller and

smaller. Below 5 kg/cm^2 (70 p.s.i.), the consumption increases very rapidly.

The heat content of the expanded actuating steam may *be* recovered by collecting it for use in a heat exchanger. However, the steam consumption of the ejector increases very rapidly with back pressure (see Table 40.22).

TABLE 40.22

RELATIVE STEAM CONSUMPTION OF EJECTOR CONDENSERS WITH DIFFERENT
BACK PRESSURES (after Hellmer²⁴)

Back pressure (g/cm^2)	Relative steam consumption
0	100
200	106
300	125
400	140
500	160
600	200
700	400

The steam consumption of an ejector increases with the vacuum sought. According to the vacuum, *the* pressure and the number of stages, it varies from 2 to 10 kg/kg (lb./lb.) air extracted. With a vacuum of 66 cm (26 in.), steam at 6 kg/cm^2 (85 p.s.i.), a back pressure of 50 g/cm^2 (0.7 p.s.i.), and a two-stage ejector, we may reckon approximately:

For the first ejector:	2.5 kg of vapour	} per kg of dry air extracted
For the second:	6 kg of vapour	
Or total:	8.5 kg of vapour	

this air moreover being saturated with water at the temperature t_1 .

Approximately, calculations may be based on a total steam consumption at 6-10 kg/cm^2 (85-140 p.s.i.), of the order of 200 kg/h (450 lb./h) for a vacuum pan, and 300-400 kg/h (650-900 lb./h) for a multiple effect. This consumption obviously depends not only on the vacuum, on the steam pressure employed and the number of stages, but also on the dimensions of the pan, on the number of effects and the temperature of the cold water used in the condenser. It also depends on the material of construction of the vessels, all leakages being greater *in* the cast iron vessel, generally estimated at double those with a corresponding vessel of steel.

Application

Steam ejector condensers are of interest:

- When air leakages are small
- When the vacuum is not too high
- When it is difficult to find space for an air pump
- When an excess of steam is available.

It is recommended that ejectors of generous capacity should be provided. The increase in

cost is only slight, and they are not likely to become rapidly under capacity with increasing rates.

It is advisable to provide efficient insulation for the pipe carrying the actuating steam, and to insert an efficient water separator just before the entry of this steam into the injector. In the same way the actuating steam should pass through a filter or a screen arranged to retain possible foreign bodies, and in particular particles of rust coming from the piping.

It is desirable that this actuating steam should be slightly superheated, so as to avoid the risk of being wet and saturated at the nozzle. The ideal would be to have dry saturated steam, but this condition is obviously difficult to maintain and a slight superheat will maintain a safety margin. An excess of superheat would be harmful as it would decrease the density and consequently the entraining effect of the steam. Wet steam, on the other hand, is also undesirable, since the droplets of water will cause erosion of the nozzle.

As with thermo-compressors, an ejector functions efficiently only at the steam pressure for which it has been designed. Care should therefore be taken to maintain its pressure at the design figure, and for that purpose to install a pressure gauge upstream of the ejector.

To avoid increasing the back pressure to no purpose, care will be taken to see that the discharge pipe into the well at the foot of the column is as short and as direct as possible, and that it is not continued too far below the safety level of the well.

Jet condensers for external extraction of air

Instead of steam, extraction of air may be effected by means of a jet of water. This is indeed

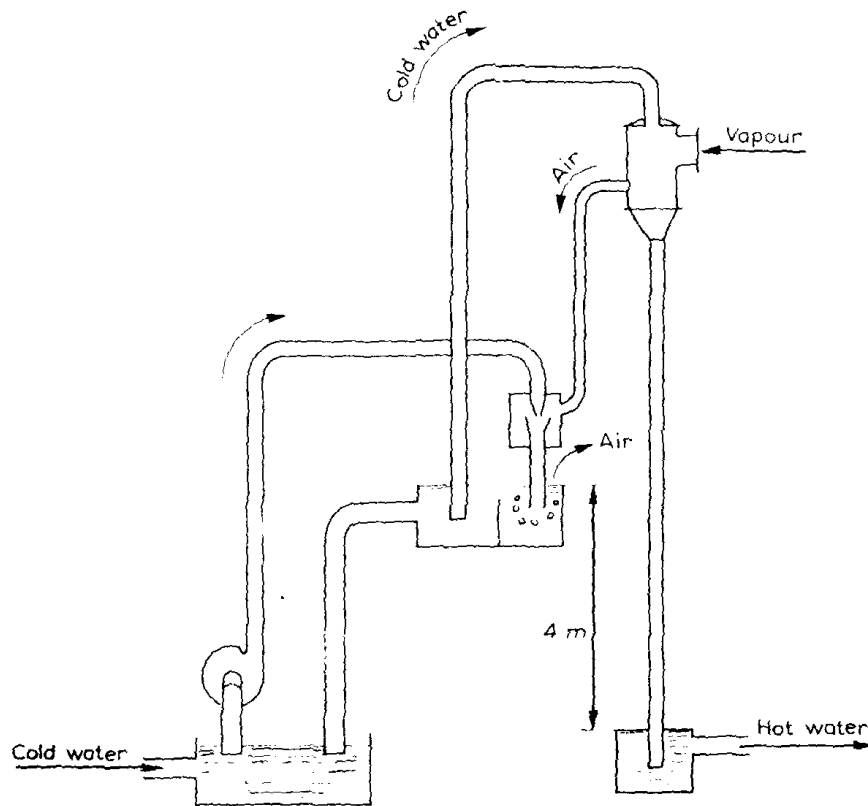


Fig. 40.25. Jet condenser with external extraction of air.

exactly the principle of the jet condenser, the only difference being that the air is extracted outside of the condenser.

The nozzle may be placed at such a height that the water discharged by it can be aspirated into the condenser. Since the quantity of air is very small, the water leaves the nozzle at a temperature practically equal to that at which it enters. The difference is not detectable on the thermometer.

Consequently a pump of rather low head will suffice (Fig. 40.25), and it may be arranged so that it is not necessary to pump the water leaving the water-actuated ejector condenser which is used to ensure condensation in the barometric column. For this, it is sufficient that the water level in the intermediate well below the ejector should be about 4 m (13 ft.) above the level in the well at the foot of the barometric column. The water in this intermediate well is thus aspirated into the condenser as soon as the vacuum approaches its normal value.

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41. Steam Production and Usage

The fibre in the cane is generally sufficient to enable the bagasse produced by the mills to supply all the steam necessary for power production and for manufacture, when utilised as fuel in the boiler furnaces.

With a normal fibre content (12 - 14%) and a well-balanced and well-designed factory, an excess of bagasse also remains (or of steam) which may be used for other purposes: irrigation pumping, manufacture of by-products, steam for the distillery, supply of electric energy to the regional network, etc.

We may comment in passing that the cane, which produces the sugar, supplies at the same time the fuel necessary for the factory which handles it; and also leaves, as by-products or waste-products:

(a) Molasses, from which is made rum, alcohol and sometimes motor fuel, for tractors and lorries.

(b) Muds, which form an excellent manure for the field, and from which wax is sometimes extracted.

(c) Ashes, produced by combustion of bagasse in the furnaces, which supplement the muds in a most convenient fashion, to supply a complete fertiliser.

Finally, it should not be forgotten that the cane itself provides the water necessary for steam production, which is very valuable for factories which do not have access to an unlimited supply of fresh water.

We shall study successively:

(a) Steam

(b) Bagasse

(c) The combustion of bagasse

(d) The boilers and their accessories

(e) The general conditions for steam production in the sugar factory.

STEAM

Tables

The principal properties of steam are given (in both British and metric units on account of their importance) in the following tables (to be found at the end of the chapter).

(a) **Table 41.1.** This gives the properties of dry saturated steam. A definite relationship exists between the pressure and the temperature of dry saturated water vapour: if one of these properties is known the other is immediately fixed.

However, the liquid phase ceases at the critical point, which corresponds to a temperature

of 374°C (705°F) and a pressure of 225 kg/cm² (3,200 p.s.i.); above this temperature, water can exist only in the gaseous form.

It will be noted that the total heat necessary to form a unit weight of steam by no means increases proportionately to the pressure. It increases at first, though slowly, passes through a maximum in the neighbourhood of 30 kg/cm² (450 p.s.i.), and then decreases:

For a pressure of

$p' = 0$	kg/cm ²	0 p.s.i.g.	$X = 639$	kcal/kg	1,150 B.T.U./lb.
$p' = 6$		100	$X = 660$		1,190
$p' = 30$		400	$X = 670$		1,205
$p' = 60$		800	$X = 665$		1,198

Now the possibilities of transformation and utilisation of the energy contained in this unit weight of steam increase very rapidly with pressure. Hence the interest of high pressures: it costs hardly any more in the way of heat content to produce steam at 30 (400) than at 6 kg/cm² (100 p.s.i.).

(b) **Table 41.2.** This table is applicable to superheated steam, and gives the mean specific heat of the steam, at different pressures, between the temperature corresponding to saturation and various temperatures of superheat. This table is useful for calculations relating to superheaters, and permits the calculation of the following table.

(c) **Table 41.3.** This table gives the total heat contained in unit weight of superheated steam, at various pressures and temperatures.

(d) **Table 41.4.** This gives the specific volume of superheated steam, at various pressures and temperatures. This table is useful for calculations relating to steam piping.

Formulae

The formulae (32.5), (32.6) and (32.7), while valid for low pressures, are only approximate above about 21 kg/cm² (300 p.s.i.), and should not be used beyond 30 kg/cm² (425 p.s.i.).

For high pressures, they may be replaced by the following formulae:

$$\lambda = 667 - \frac{(231 - t)^2}{1100} \quad \left(1,205.5 - \frac{(447.6 - t)^2}{1,980} \right) \quad (41.2)^*$$

$$r = 92.8 (374 - 1)^{0.315} \quad (138.8 (705 - t)^{0.315}) \quad (41.3)$$

X = total heat content of saturated steam at t° expressed in kcal/kg (B.T.U./lb.) for absolute pressures above 30 kg/cm² (425 p.s.i.)

r = latent heat of vaporisation of the steam at pressures above 30 kg/cm² (425 p.s.i.)

t = saturation temperature of the steam, in °C (°F).

* For eqn. (41.1) see p. 1042.

Dryness fraction

Steam produced in boilers not equipped with superheaters is generally not dry, since it carries with it droplets of water. The dryness fraction x of a wet steam is the quantity of dry steam contained in unit weight of that steam. Steam of a dryness fraction $x = 0.95$ contains 95% vapour and 5% water entrained as droplets.

Tromp² gives as mean values in the sugar factory, when there are no superheaters:

High-pressure steam: $x = 0.95-0.97$

Exhauststeam: $x = 0.80-0.90$

However, many designers assume 0.95 for the exhaust steam from engines.

A calorimeter for determining dryness fraction is fully specified by Tromp² and moreover is readily installed; in the absence of such an apparatus, we may adopt as a first approximation:

$$\begin{array}{ll} \text{Live steam:} & x = 0.95 \\ \text{Exhaust steam:} & x = 0.90 \end{array} \quad (41.4)$$

Entropy diagrams

The engineer who is studying plans for installations of boilers, turbines or steam engines cannot do without a steam diagram. This is a graph on which are set out the most important properties of steam. The most valuable diagrams are:

(a) *The Mollier diagram* (Fig. 41.1), which plots as abscissa the entropy of the steam and as ordinate its total heat X .

(b) *The entropy diagram* (Fig. 41.2), which plots as abscissa the entropy and as ordinate the temperature t .

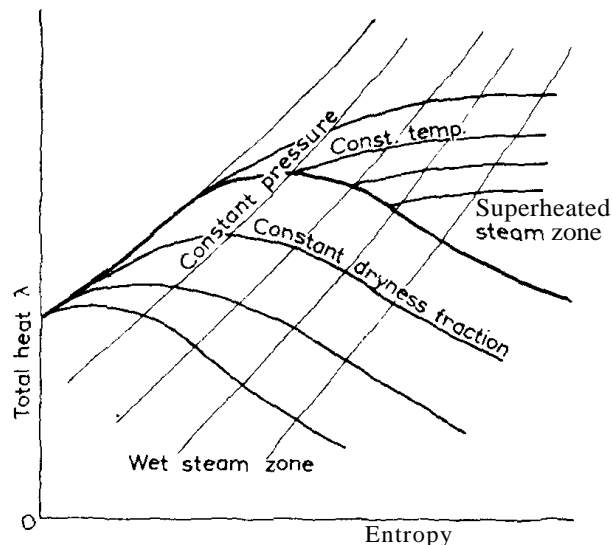


Fig. 41.1. Mollier diagram. Significance of graphs.

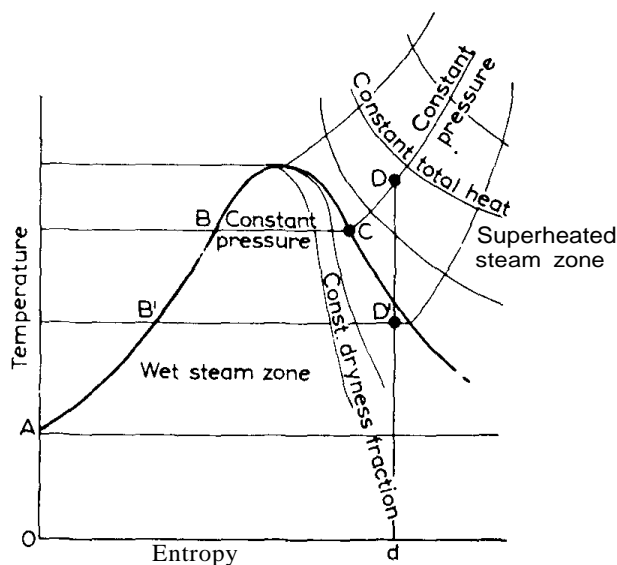


Fig. 41.2. Entropy diagram. Significance of graphs.

The entropy of steam is the quantity:

$$\Phi = \int_0^t \frac{dQ}{T} \quad \left(\Phi = \int_{32}^t \frac{dQ}{T} \right) \quad (41.5)$$

t = steam temperature

T = absolute temperature of the steam = $(273 + t) ^\circ\text{C}$ ($(459.4 + t) ^\circ\text{F}$)

Q = quantity of heat involved in the transformation under consideration.

This somewhat unfamiliar concept of entropy presents great advantages:

- (1) The diagram may be drawn at a suitable scale for giving the properties of steam under all conditions encountered in practice.
- (2) Isothermal transformations are represented on the entropy diagram by horizontal lines.
- (3) Adiabatic changes are represented by vertical lines.
- (4) Changes which involve no external work are represented by lines of equal total heat.

It is therefore easy to trace the properties of steam before and after a given change of conditions. We may consider:

(a) Changes at constant total heat:

Throttling in steam engines.

Expansion without external work (in a pressure reducer, for example).

The variation in internal energy of the steam, corresponding to the drop in pressure, is utilised to evaporate the water contained in the steam (wet steam) or to superheat it (dry steam).

(b) Isothermal changes:

Evaporation at constant pressure.

(c) **Adiabatic changes:**

Compression.

Expansion in a turbine or steam engine, with production of external work. This is the most important and the most interesting case in the present discussion.

The external work is obtained from the heat given up by the *steam*. The total heat of a unit mass of steam, the state of which is represented by the point *D* (Fig. 41.2) by reason of the choice of coordinates, is equal to the area *OABCDd*. In the same way, the total heat relative to the point *D'* will be equal to the area *OAB'D'd*. It is not necessary to measure these areas, however, since *X* is given by curves placed on the diagram and may be read off by interpolation.

Example. Assume a steam turbine receiving steam at 23 kg/cm^2 (327 p.s.i.a.) and 350°C (662°F). This condition is represented on the diagram by the point *D*, intersection of the isobar 23 kg/cm^2 with the horizontal 350° . The corresponding total heat is read off from the diagram: = 748 kcal/kg (1,347 B.T.U./lb.). The expansion in the turbine will be adiabatic, *i.e.* it will take place along a vertical line on the diagram. If the back pressure is $0.5 \text{ kg/cm}^2 \approx 1.533 \text{ kg/cm}^2$ abs. (21.8 p.s.i.a.) the final state of the steam will be represented by the point *D'*, at the intersection of the vertical through *D* and the horizontal of 1.533 kg/cm^2 (21.8 p.s.i.a.) or 111°C (232.5°F). It is readily seen on the diagram that this point *D'* corresponds to wet steam of dryness fraction $x = 0.935$ and of total heat $X = 605 \text{ kcal/kg}$ (1,090 B.T.U./lb.). The external work accomplished per kg (lb.) of steam is then:

$$T = (X - \lambda') 427 \text{ kgm/kg} \quad (778 \text{ ft.lb./lb.}) \quad (41.6)$$

or here:

$$T = (748 - 605) 427 = 61,000 \text{ kgm/kg} \quad (200,000 \text{ ft.lb./lb.})$$

$427 (778) =$ mechanical equivalent of the kcal (B.T.U.) = number of kgm (foot pounds) corresponding to 1 B.T.U.

In practice, it is obviously necessary to take into account the efficiency of the operation.

Moreover, the change is not exactly adiabatic; instead of following a vertical line *DD'* (Fig. 41.3), it follows a curve *DD''*: there is a slight degradation of energy leading to a slight superheat (or to an increase in dryness fraction).

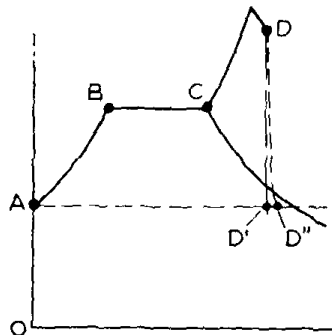


Fig. 41.3. Adiabatic expansion.

We shall have several occasions to return to the entropy diagram.

Figures 41.4 (metric units) and 4.15 (British units) give Mollier diagrams, which will allow consideration of problems concerned with the use of steam in the factory.

Equivalents

Table 41.5 will serve to recall the equivalent figures for heat and mechanical work.

TABLE 41.5

THERMO-MECHANICAL EQUIVALENTS

E , the work corresponding to 1 kcal	= 427	kg m
E , the work corresponding to 1 B.T.U.	= 778.3	ft.lb.
hence: 1 h.p.h. requires $\frac{75 \times 3,600}{427} = 632$ kcal ($\frac{550 \times 3,600}{778.3} = 2,544$ B T U.)		
1 kWh requires $\frac{632}{0.745} = 860$ kcal $\frac{2,544}{0.745} = 3,413$ B.T.U.		

BAGASSE

Final bagasse, or simply bagasse, is the solid fibrous material which leaves the delivery opening of the last mill of the tandem, after extraction of the juice.

It is the residue from the milling of cane.

Physical composition

In spite of the diversity of milling plants and machines employed, the physical composition of bagasse varies between rather narrow limits. Its most important property, from the point of view of steam production, is its *moisture content*. Poor work at the mills will give a bagasse of 50% moisture, while very good work will give a bagasse of 45% moisture. Certain factories in Hawaii and Formosa have reported moistures of the order of 38%, but these are exceptional figures. It is still difficult, even in a modern mill, to obtain figures as low as 44%.

The most frequent values are:

$$w = 45 - 50\% \quad (41.7)$$

w = moisture in bagasse

and generally we shall not involve any great error in adopting for practically all cases the standard value:

$$w = 48\% \quad (41.8)$$

In addition to water, the bagasse contains:

(a) Insoluble material, consisting mainly of cellulose, and comprising the fibre content of the bagasse

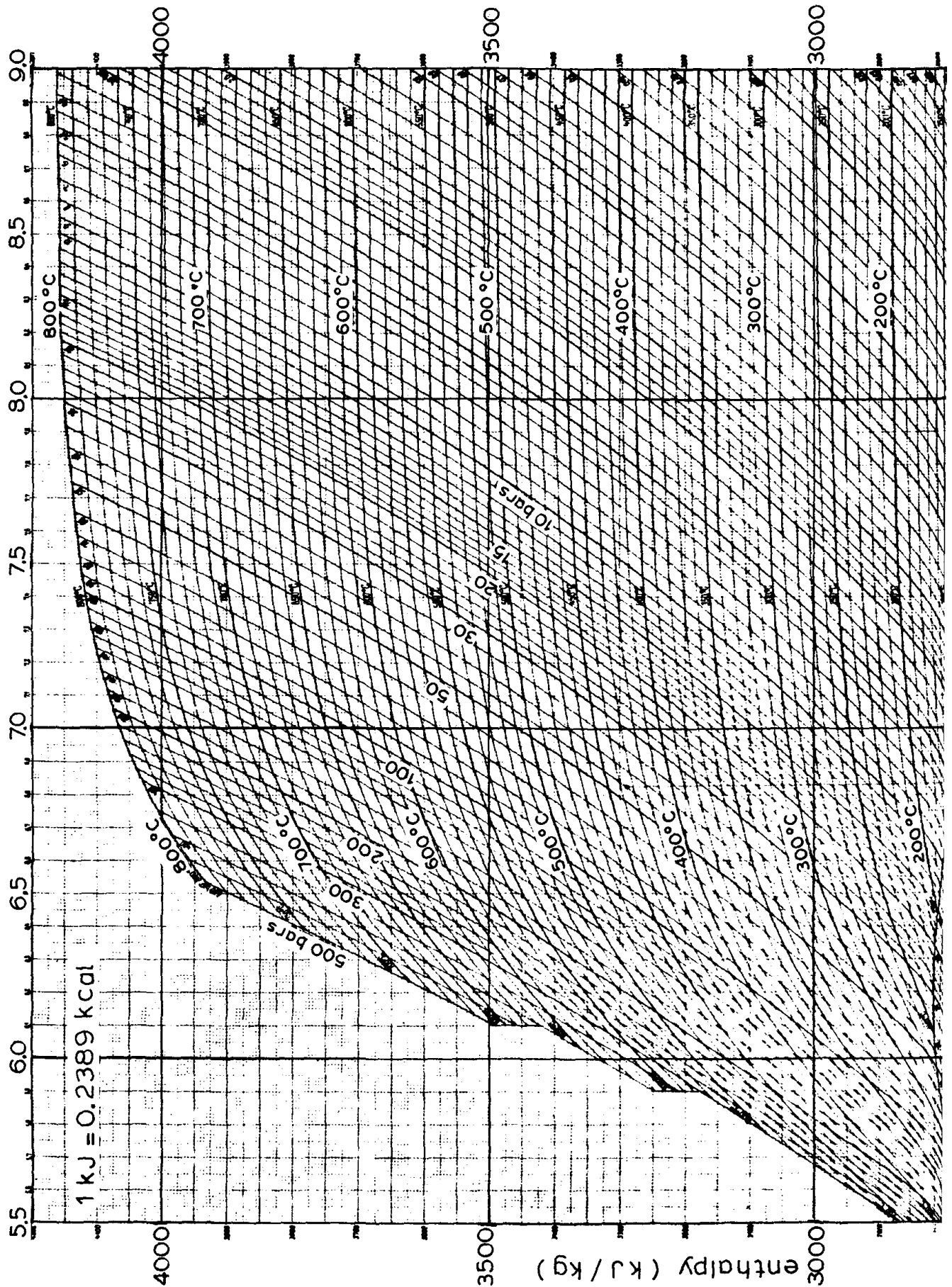
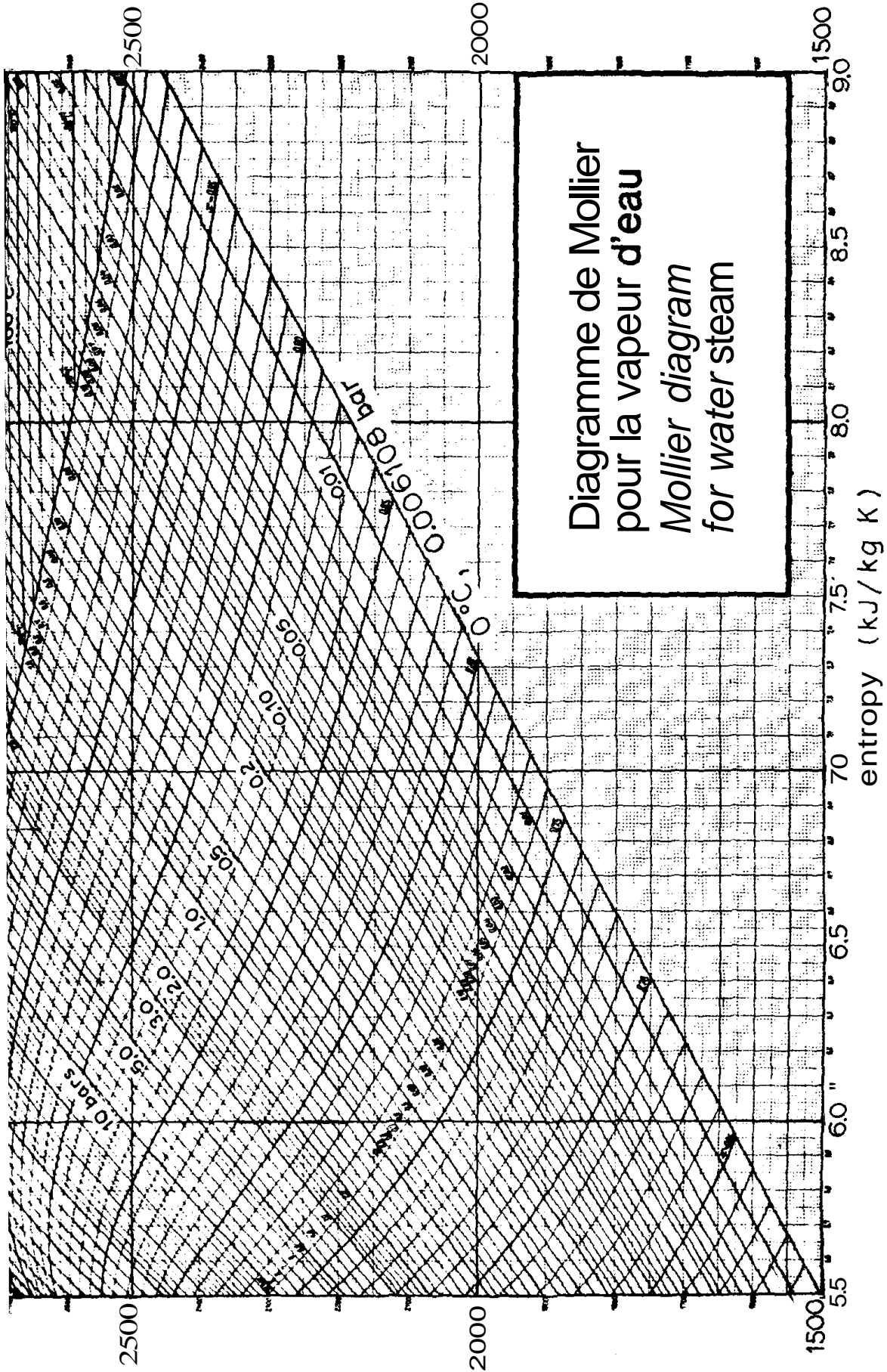


Fig. 41.4. Mollier diagram for calculation of projects (metric units).



A MOLLIER CHART FOR STEAM

Modified and greatly reduced from Keenan and Keyes's Thermodynamic Properties of Steam, published (1936) by John Wiley and Sons, Inc. Reproduced by permission of the publishers.

Abs. press., in. Hg.	Sat. temp., °F	Abs. press., lb. per sq. in.	Sat. temp., °F	Abs. press., lb. per sq. in.	Sat. temp., °F
0.30	34.56	1.0	101.74	120	341.25
(US)	40.23	2	126.08	140	353.02
0.30	44.96	3	141.48	160	363.53
0.35	49.06	4	152.97	180	373.06
0.40	52.64	5	162.24	200	381.79
0.45	55.87	6	170.06	230	389.86
0.50	58.80	7	176.85	240	397.57
0.55	61.48	8	182.96	260	404.42
0.60	63.95	9	188.28	280	411.05
0.65	66.26	10	193.21	300	417.33
0.70	68.40	12	201.96	400	444.59
0.75	70.43	14	209.25	500	467.01
0.80	72.32	1	212.00	600	486.21
0.85	74.13	16	216.32	700	503.10
0.90	75.84	18	222.41	800	518.23
0.95	77.47	20	227.96	1000	544.61
1.00	79.03	25	240.07	1200	567.22
1.10	81.95	30	250.13	1400	587.10
1.20	84.65	35	259.28	1600	604.90
1.30	87.17	40	267.25	1800	621.03
1.40	89.51	45	274.44	2000	635.82
1.50	91.72	50	281.01	2200	649.46
1.60	93.80	60	292.71	2400	666.12
1.70	95.77	70	302.92	2600	679.94
1.80	97.65	80	312.03	2800	684.99
1.90	99.43	90	320.27	3000	690.36
2.00	101.14	100	327.81	3206.2	705.40

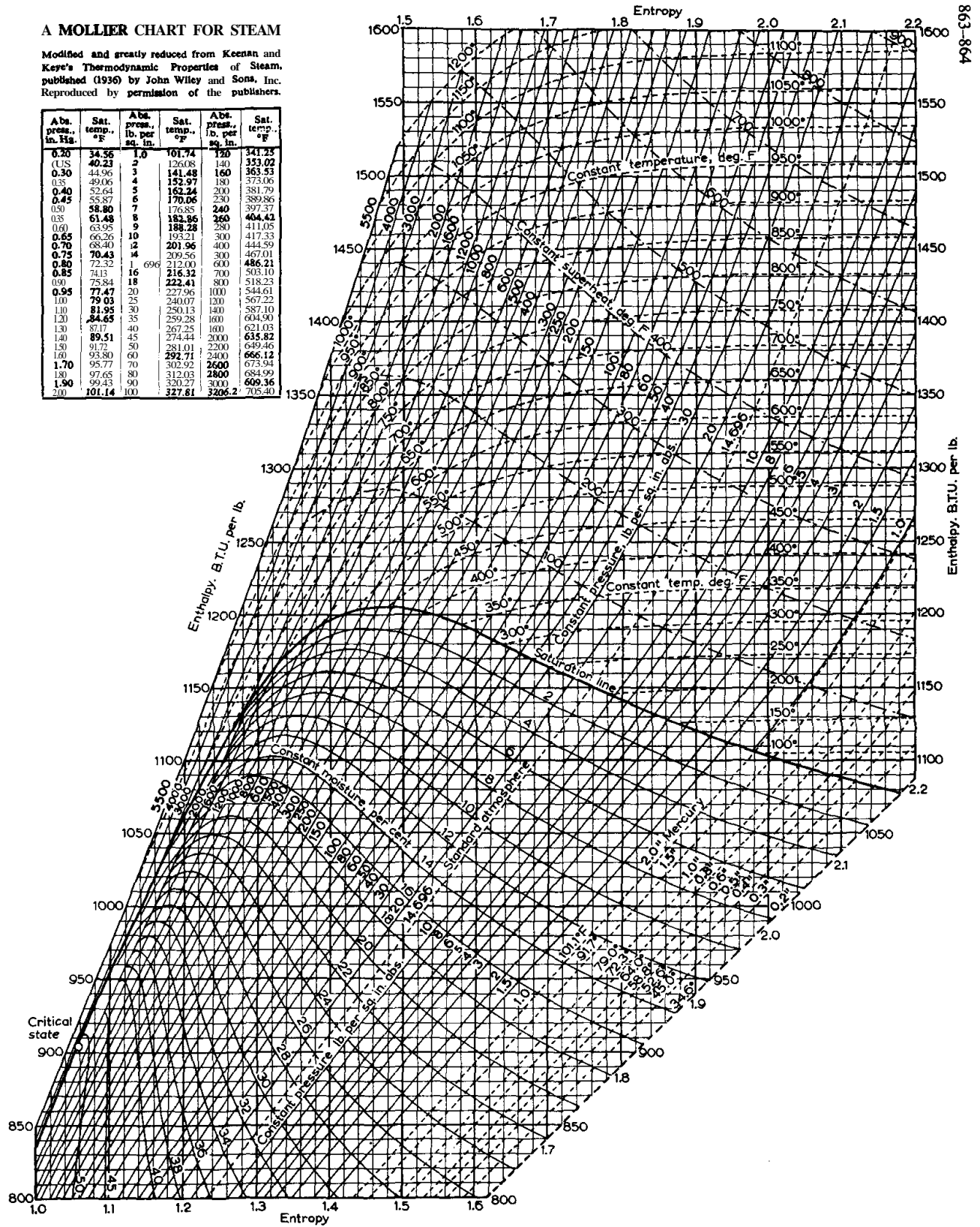


Fig. 41.5. Mollier diagram (British units).

(b) Substances in solution in the water (this water obviously originating in the juice and the imbibition), consisting of sugar and impurities.

These dissolved substances are present in small quantities, ranging from 2 to 4%. If we designate their proportion by weight as $D\%$, there remains of fibre:

$$/ = 100 - w - D = 46-53\%. \quad (41.9)$$

a fairly frequent value being:

$$/ = 50\% \quad (41.10)$$

$/$ = fibre % bagasse.

Quantity of bagasse

The extreme values of the mean fibre content of cane are close to: $/ = 10$ and 16% ; but it generally lies in the region $12 - 15\%$.

We obtain the quantity B of bagasse obtained from 100 parts of cane, by equating the weight of fibre entering the mills to that leaving:

$$100f = Bf'$$

hence:

$$B = 100 \frac{f}{/'} \quad (41.11)$$

It will be seen that the quantity of the bagasse varies between 24 and 30% by weight of cane, or approximately one-quarter.

We shall have available then about 275 kg (616 lb.) of bagasse per t.c.

Bulk density

Tromp³ gives as bulk density of bagasse:

160-240 g/dm³ (10–15 lb./cu.ft.) when stacked

80- 120 g/dm³ (5–7.5 lb./cu.ft.) in a loose condition.

This density depends mainly on its moisture content, thus bagasse is lighter when it contains less water. For a mean value $w = 48\%$, we may take:

$d = 200$ g/dm³ (12.5 lb./cu.ft.) for bagasse in a stack

$d = 120$ g/dm³ (7.5 lb./cu.ft.) for loose bagasse.

We may take $160 - 170$ g/dm³ (10–11 lb./cu.ft.) for loose bagasse loaded on a lorry to a height of 5 - 7 feet. Belt conveyors are calculated on a bulk density of 130 kg/m³ (8 lb./cu.ft.).

Storage of bagasse

The bulk density of bagasse makes it a very bulky material. Storage of excess bagasse from the factory presents a difficult problem for this reason.

Except in very dry localities, bagasse cannot be left out in the open, since it ferments, decays and loses a large proportion of its value even for use as fuel.

However, it may, if necessary, be stored in the open, on condition that it is placed in the form of a conical or pyramidal stack, with the top at a slope of at least 30° (Fig. 41.6), and that a roof is formed by means of cane leaves overlapping like roofing slates, running lengthwise in the direction of the slope, as in the thatched roofs of native cottages.

Generally, it will be of advantage to store the bagasse under a roof.

The angle of repose of bagasse is rather variable, but generally in the range of $45 - 50^\circ$. However, the edges of a stack may be trimmed to vertical walls, without risk of collapse; they hold very well.

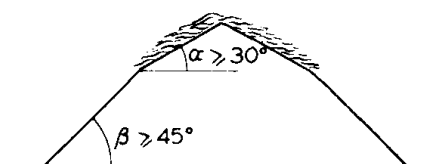


Fig. 41.6. Bagasse pile in open air.

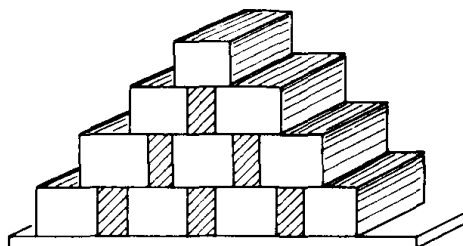


Fig. 41.7. "Draughtboard" stacking of baled bagasse.

To avoid the necessity for large and expensive buildings, stored bagasse should be compressed, in order to reduce its volume.

Penetration of rain in a stack of bagasse is given by:

$$h = \frac{10w}{d \left(0.056 - \frac{1}{k} \right)} \quad (41.12)$$

h = penetration, in cm

w = rainfall, in cm

d = specific weight of bagasse, in kg/m^3 ($d = 74$ for fresh bagasse, 110 for wet material)

k = consistency of the bagasse at the surface, in $\%$, before the rain ($k = 50$ if dry, 20 if wet)

Bagasse presses

There are two principal types of bagasse press:

(1) **Baling press.** Bales of $30 \times 30 \times 60$ cm ($12 \times 12 \times 24$ in.), or $45 \times 45 \times 60$ cm ($18 \times 18 \times 24$ in.) are formed, and are tied with two or three wires, to prevent them from re-expanding. A hydraulic press is employed for this purpose, similar to a press for hay.

The bales of fresh bagasse have a density of $400 - 600 \text{ kg/m}^3$ ($25 - 40 \text{ lb./cu.ft.}$), according to the pressure employed. They are then stacked in "draughtboard" fashion (Fig. 41.7) so

as to allow air to circulate between the bales. In this way they dry more rapidly. However, this practice is discouraged in certain countries, such as the Philippines, where it is considered that it allows fire to spread more rapidly. On the other hand, it decreases the risk of spontaneous combustion.

In two or three months, the bagasse loses part of its moisture, which decreases from 45–50% to about 25%, a value at which it becomes stable. The bagasse bales then weigh only about 300–400 kg/m³ (20–25 lb./cu.ft.).

Bagasse baled while moist, as it leaves the mills, can give rise to spontaneous combustion when stacked in a large pile.

Bagasse baled and stacked in "draughtboard" fashion and sheltered from the weather may be stored for a long time. When a long period of storage is foreseen, it is advantageous to sprinkle powdered boric acid on the stack proportionately as the stack is built up.

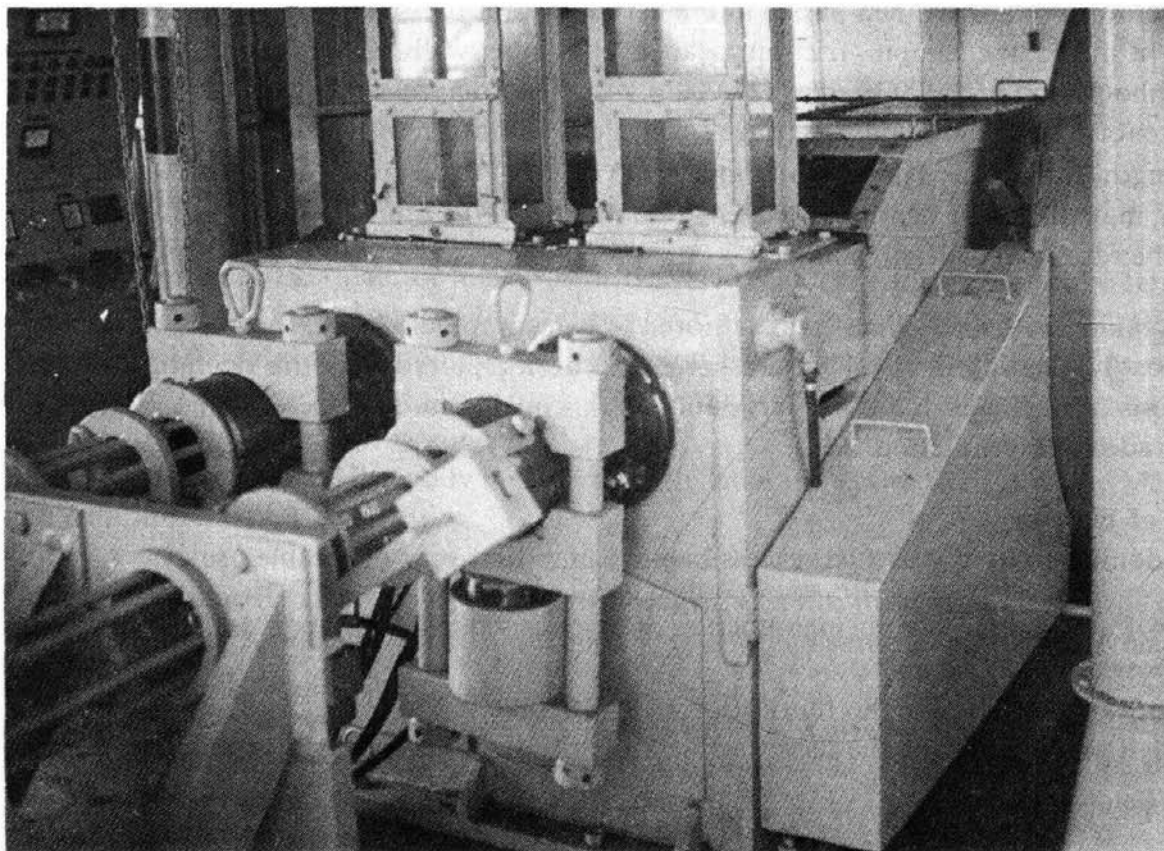


Fig. 41.8. Bagasse briquetting press (Pawert).

(2) Briquetting press. Bagasse in its original state cannot be made to remain compressed; whatever pressure be employed, it does not retain any cohesion. If, on the contrary, it is dried, it is possible to obtain briquettes:

(a) On condition that the moisture content has been reduced below 15%. Best results are obtained with bagasse of 7–8% water. If the drying is taken further the briquettes would re-absorb moisture from the air and would expand.

(b) Provided pressures of the order of 350 kg/cm² (5,000 p.s.i.) are used, or preferably 1,000 kg/cm² (15,000 p.s.i.)

Bagasse briquettes are made in two forms:

(a) Briquettes proper, small plates of 20 x 10 x 2 cm (8 x 4 x 1 in.), which can be stacked at a bulk density of 720–880 kg/m³ (45–55 lb./cu.ft.).

Tromp⁴ specifies a press of 70 h.p. for 5—10 tonnes of bagasse per hour.

Briquetting is facilitated and cohesive properties of the briquettes are improved by adding a binder, molasses for example, in the ratio of one part per four of bagasse.

In Cuba⁵, a pressure of 8.5 kg/cm² (120 p.s.i.) is considered sufficient, when a binder is incorporated.

(b) As "logs", cylindrical agglomerates of 6.9 or 13 cm (2.5, 3.5 or 5 in.) diameter. The best-known presses furnishing pieces of this type are the "Glomera" of Pawert in Basle. The bagasse is dried to about 10% moisture by hot flue gas, in vertical dry tubes, through which it is blown by fans. It then passes into a very powerful press (Fig. 41.8), where it is pressed into a tube; the simple friction against the tube walls generates a pressure in the neighbourhood of 1,400 kg/cm² (20,000 p.s.i.). The compressed bagasse leaves the cylinder in the form of a continuous "sausage" which can be broken into pieces 13—20 cm (5—8 in.) in length. It then has a density of about 1,100 kg/m³ (70 lb./cu.ft.) and, even when stored in bulk, may still have a density of 400–480 kg/m³ (25–30 lb./cu.ft.). Thus it forms a fuel which can be readily and cleanly handled.

Breaking up bales. When bagasse stored in bales is to be re-used, the bales are broken up by passing them through a light crusher. It has been reported⁶ that this operation may cause fibrosis of the lungs in the workers handling this operation. It is therefore necessary to provide adequate ventilation.

Use of bagasse

In addition to its use as fuel, excess bagasse may find very profitable use as:

(a) Raw material for the manufacture of fireproofed insulating boards, used for building purposes.

(b) Raw material for the fabrication of paper pulp.

(c) Raw material for the manufacture of various solvents utilised in industry.

TABLE 41.6

CHEMICAL COMPOSITION OF BAGASSE

	Deerr ⁷	Tromp ⁸	Kelly ⁹	M.R.* ¹⁰	Davies ¹¹	Gregory ¹²	
Carbon C	46.5	44	48.2	47.5	47.9	49	48.1
Hydrogen H	6.5	6	6	6.1	6.7	7.4	6.1
Oxygen O	46	48	43.1	44.4	45.4	41.8	43.3
Ash c	1	2	2.7	2	—	1.8	2.5
	100	100	100	100	100	100	100

* M.R. = mean of results of Kersten, Prinsen Geerligs, Bolck, v.d. Kreke and Spencer.

Chemical composition of bagasse

The chemical composition of dry bagasse varies slightly according to different authors (see Table 41.6).

The results given in the fourth column, originally reported as % fibre, have been converted to a basis of 2% ash.

Since these variations are not large, we shall use the following mean standard composition in our calculations:

$$\begin{array}{rcl}
 \text{C} & = & 47 \% \\
 \text{H} & = & 6.5\% \\
 \text{O} & = & 44 \% \\
 \epsilon & = & 2.5\% \\
 \hline
 & & 100 \%
 \end{array} \tag{41.13}$$

Calorific value of bagasse

The calorific value (C.V.) is the quantity of heat which will be released by combustion of unit weight of the fuel under consideration.

We distinguish two different calorific values:

(a) **The gross calorific value**, or higher calorific value (G.C.V.): this is the heat liberated by the combustion of one kg (lb.) of fuel, taken at 0°C (32°F) and under 760 mm (29.92 in.) of mercury, all the products of combustion being reduced to the same conditions. The water present in the fuel, as well as the water formed by combustion of the hydrogen entering into its composition, is consequently condensed.

The gross calorific value is readily measured in the laboratory with the aid of the Mahler bomb calorimeter.

(b) **The nett calorific value**, or lower calorific value (N.C.V.), which assumes on the contrary that the water formed by combustion, and the water of constitution of the fuel, remains in the vapour state.

The G.C.V. gives a good measure of the heat theoretically available from the fuel, but since, in industrial practice, it has not yet been found practicable to reduce the temperature of the combustion products below the dew point, the N.C.V. gives a more accurate indication of the heat practically obtainable.

It is the N.C.V. which therefore should be used in practice; but there is no means of determining it directly. It is therefore necessary to calculate this value.

In this calculation, there is a certain contradiction between the condition that the combustion gases should be reduced to 0°C (32°F) and 760 mm (19.91 in.), and the condition that the water should not be condensed. The convention is adopted of subtracting from the G.C.V. the weight of hydrogen in 1 kg (1 lb.) of fuel, multiplied by 600 kcal (1,080 B.T.U.).

This amounts to assuming that for the G.C.V., condensation takes place in the neighbourhood of 10°C (50°F), in the course of cooling to 0°C (32°F), since (eqn. 32.6):

$$r = 607 - (0.7 \times 10^\circ) = 600 \text{ kcal} \quad (1,093 - (0.7 \times 18^\circ) = 1,080 \text{ B.T.U.})$$

It is moreover easy to verify, from the latent heat and the specific heats of water and water vapour, that the temperature at which the condensation (or in the inverse sense, evaporation) is considered to take place, has no great influence on the total heat liberated. A temperature in *the* neighbourhood of 0°C (32°F) is therefore adopted while taking into account, in the calculation, only vapour and not liquid water; this simplifies matters.

The N.C.V. of a fuel is therefore given by the formula:

$$\text{N.C.V.} = \text{G.C.V.} - 600E \quad (\text{G.C.V.} - 1,080E) \quad (41.14)$$

E = weight of water vapour present in the gases produced by combustion of 1 kg (1 lb.) of fuel, expressed in kg (lb.).

Furthermore, combustion of hydrogen takes place according to the reaction:



The weight of water formed is thus equal to nine times the weight of hydrogen. We then have for a dry fuel:

$$E = 9H \quad (41.16)$$

H = weight of hydrogen contained in 1 kg (1 lb.) of fuel.

Hence:

$$\text{N.C.V.} = \text{G.V.C.} - 5,400H \quad (\text{G.C.V.} - 9,720H) \quad (41.17)$$

This equation applies only to a dry fuel. For a wet fuel, it is also necessary to take into account the original water present, which is then added to the water formed by combustion.

Gross calorific value of dry bagasse. In spite of considerable differences in appearance between different varieties of cane, the G.C.V. of *dry* bagasse is remarkably constant in all countries and for all varieties of cane. Table 41.7, for example, gives several values selected at random.

TABLE 41.7

O.C.V. OF DRY BAGASSE

Author	Country	G.C.V. of dry bagasse	
		kcal/kg	B.T.U./lb.
Behne ¹³	Queensland	4,542	8,177
Hedley ¹⁴	S. Africa	4,585	8,253
Ref. 15	Hawaii	4,622	8,320
Gregory ¹²	Cuba	4,691	8,444
Gregory ¹²	Puerto Rico	4,594	8,270
Ref. 15a	S. Africa	4,700	8,460
	Mean	4,622	8,320

It will scarcely involve an error of more than 2%, therefore, we adopt as a universal value for the G.C.V. of dry bagasse:

$$\text{G.C.V.} = 8,280 \text{ B.T.U./lb.} = 4,600 \text{ kcal/kg} \quad (41.18)$$

Nett calorific value of dry bagasse. We have seen (eqn. 41.13) that dry bagasse contains 6-7% hydrogen, and have taken the mean figure as 6.5%.

Equation (41.17) then gives:

$$\text{N.C.V.} = \text{G.C.V.} - (0.065 \times 9,720) = 8,280 - 630 = 7,650 \text{ B.T.U./lb.}$$

$$\boxed{\text{N.C.V.} = 7,650 \text{ B.T.U./lb.} = 4,250 \text{ kcal/kg}} \quad (41.19)$$

Calorific value of wet bagasse. We now know the calorific value of dry bagasse. How are we to deduce the value for the wet bagasse with which we have to deal in practice?

Theoretical calculation. We may first base our figures on the percentage composition of wet bagasse, which is given in Table 41.8.

TABLE 41.8

CALORIFIC VALUE OF CONSTITUENTS OF BAGASSE

Constituent	%	C. V.	
		kcal/kg	B.T.U./lb.
Fibre	/	4,600	8,280
Sugar	s	3,955 (= 4,000)	7,120
Impurities	t	4,100	7,380
Water	w	0	0

For the separate sugars: sucrose, 3,939 kcal/kg; glucose and fructose, 3,719 kcal/kg^{15b}.

Not only has water no calorific value, but on the other hand it absorbs heat in being vaporised during the combustion (N.C.V.).

$$\text{G.C.V.} = 4,600 \frac{f'}{100} + 3,955 \frac{s}{100} + 4,100 \frac{t}{100}$$

$$\text{N.C.V.} = 4,600 \frac{f'}{100} + 3,955 \frac{s}{100} + 4,100 \frac{t}{100} - 350 \frac{100 - w}{100} - 600 \frac{w}{100}$$

$$\left(\text{G.C.V.} = 8,280 \frac{f'}{100} + 7,120 \frac{s}{100} + 7,380 \frac{i}{100} \right)$$

$$\left(\text{N.C.V.} = 8,280 \frac{f'}{100} + 7,120 \frac{s}{100} + 7,380 \frac{i}{100} - 630 \frac{100 - w}{100} - 1,080 \frac{w}{100} \right)$$

or:

$$\left. \begin{aligned} \text{G.C.V.} &= 46f' + 39.555 + 41i \\ \text{N.C.V.} &= 46f' + 39.55s + 41i - 2.5w - 350 \end{aligned} \right\} \text{ kcal/kg}$$

or:

$$\left. \begin{aligned} \text{G.C.V.} &= 82.8f' + 71.25 + 7.38i \\ \text{N.C.V.} &= 82.8f' + 71.25 + 7.38i - 4.5w - 630 \end{aligned} \right\} \text{ B.T.U./lb.}$$

Since the C.V. of sugar and of the impurities differ only slightly, no great error will be involved by taking for these two quantities the mean values, which moreover vary only slightly. To simplify the calculation, they will be expressed in terms of sugar. We thus have:

$$f' = 100 - 5 - i - w$$

The purity of the residual juice is generally of the order of 45 - 50. If we adopt this value:

$$i = \frac{52.5}{47.5} s = 1.1 s$$

Hence:

<i>British units</i>	<i>Metric units</i>	
G.C.V. = 8,280 - 21.65 - 82.8 w	G.C.V. = 4,600 - 125 - 46 w	(41.20)
N.C.V. = 7,650 - 21.65 - 87.3 w	N.C.V. = 4,250 - 125 - 48.5 w	

G.C.V. = gross calorific value of the bagasse, in B.T.U./lb. (or kcal/kg)

N.C.V. = net calorific value of the bagasse, in B.T.U./lb. (or kcal/kg)

5 = sucrose % bagasse

w = moisture % bagasse.

Analogous formulae, though slightly different, have been utilised.

In particular in India¹⁶:

$$\text{G.C.V.} = 4,600 - 85 - 53 w \quad (8,280 - 14.45 - 95.4 w) \quad (41.21)$$

In Java the formula of von Pritzelwitz van der Horst⁷ was used:

$$\begin{aligned} \text{G.C.V.} &= 4,550 - 105 - 45 w && (8,190 - 18s - 81 w) \\ \text{N.C.V.} &= 4,250 - 105 - 48 w && (7,650 - 185 - 86.4 w) \end{aligned} \quad (41.22)$$

In Mauritius¹⁸:

$$\text{N.C.V.} = 4,150 - 7.55 - 47.5 w \quad (7,470 - 13.55 - 85.5 w) \quad (41.23)$$

Hessey¹⁹ in Queensland gave:

$$\begin{aligned} \text{G.C.V.} &= 4,636 - 12.35 - 46.36 w && (8,345 - 225 - 83.45 w) \\ \text{N.C.V.} &= 4,324 - 12.35 - 49.04 w && (7,783 - 225 - 88.27 w) \end{aligned} \quad (41.24)$$

the figures being determined experimentally and verified fairly well in practice.

South Africa gives:

$$\text{N.C.V.} = 18,309 - 31.14 B - 207.63 w - 196.09c \text{ kJ/kg}$$

which, in kcal/kg, becomes:

$$\text{N.C.V.} = 4.375 - 7.44 B - 49.62 w - 46.86c \text{ kcal/kg}$$

B = Brix % bagasse

c = ash % bagasse.

If we express s and w not in per cent but per unit of bagasse, we would have:

$$\begin{aligned} \text{G.C.V.} &= 4,600 (1 - w) - 1,200s && (8,280 (1 - w) - 2,160s) \\ \text{N.C.V.} &= 4,250 - 4,850 w - 1,200s && (7,650 - 8,730 w - 2,160s) \end{aligned} \quad (41.25)$$

We may comment that, in this formula, we have already taken into account the following heat losses, from the point of view of steam production in the factory:

(a) Latent heat of vaporisation of the water formed by combustion of the hydrogen contained in the bagasse, which is lost in the flue gases with this water vapour if the latter is not condensed.

(b) Latent heat of vaporisation of the water content of bagasse, which in the same way is lost with the flue gases.

We have then to take into account only the following losses:

(a) Sensible heat lost in the flue gases

(b) Losses by radiation

(c) Losses in unburnt solids

(d) Loss by incomplete combustion of carbon giving CO instead of CO₂.

COMBUSTION OF BAGASSE

As this section is rather complex, it would be over-loaded if calculations were given in both systems of units. They will be given in metric units, with the comments:

- (a) That they are given in British units in the second edition, in identical fashion
- (b) All values given as kg/kg are the same in lb./lb.
- (c) Volumes only are changed, and $1 \text{ m}^3/\text{kg} = 16.02 \text{ cu.ft./lb.}$
- (d) Numbers of formulae and Tables which are identical in both systems are underlined.
- (e) Both systems are retained where it does not interfere with the clarity of the discussion.

Composition of air

The composition of dry air is given in Table 41.9.

TABLE 41.9

COMPOSITION OF DRY AIR

	<i>% By weight</i>	<i>% By volume</i>
Oxygen	23.15	20.84
Nitrogen and inerts	76.85	79.16

Reactions of combustion

The combustible elements in bagasse are carbon and hydrogen. In burning they give:



By weight

$$12 \text{ g} + 32 \text{ g} = 44 \text{ g} \qquad 2 \text{ g} + 16 \text{ g} = 18 \text{ g} \qquad (41.26)$$

Or:

$$1 + 2.67 = 3.67 \qquad 1 + 8 = 9$$

Properties of gaseous products of combustion

Table 41.10 gives the principal properties which we shall use in the study of the combustion, for the principal components of the flue gases.

(A) Combustion of dry bagasse without excess air

The composition (41.13) of bagasse and the proportions given by the reactions (41.26) allow us to calculate the quantity of oxygen necessary for combustion.

TABLE 41.10

PROPERTIES OF GASEOUS PRODUCTS OF COMBUSTION

	<i>Mol wt</i> (kg/m ³)	<i>Constant R</i> (kg/m ² and m ³ /kg)	<i>(p.s.i.</i> <i>and</i> <i>cu.ft./lb.)</i>	<i>Density</i> <i>at 0°C and 760 mm</i> (kg/m ³)	<i>(lb./cu.ft.)</i>	<i>Specific vol.</i> <i>at 0°C and 760 mm</i> (m ³ /kg)	<i>(cu.ft./lb.)</i>
CO ₂	44	19.27	0.2434	1.977	0.1234	0.506	8.10
H ₂ O	18	47.06	0.5944	0.804	0.0502	1.244	19.92
N ₂ (atmospheric)	28	30.26	0.3822	1.256	0.0784	0.796	12.75
O ₂	32	26.50	0.3347	1.429	0.0892	0.700	11.21
CO	28	30.29	0.3826	1.250	0.0780	0.800	12.81
Air	29	29.27	0.3697	1.293	0.0807	0.773	12.39

(a) **Oxygen.** In order to burn 1 kg of dry bagasse, we require.

$$\text{C } 0.470 \text{ kg} \times 2.67 = 1.253 \text{ kg O}_2 = 0.877 \text{ m}^3 \text{ of oxygen}$$

$$\text{H}_2 \text{ } 0.065 \text{ kg} \times 8 = 0.520 \text{ kg O}_2 = 0.364 \text{ m}^3 \text{ of oxygen}$$

$$\text{or a total of } 1.773 \text{ kg O}_2 = 1.241 \text{ m}^3 \text{ of oxygen}$$

$$\text{But the bagasse already contains } 0.440 \text{ kg O}_2 = 0.308 \text{ m}^3 \text{ of oxygen}$$

$$\text{Hence the air must supply } 1.333 \text{ kg O}_2 = 0.933 \text{ m}^3 \text{ of oxygen}$$

(b) **Nitrogen.** This oxygen brings with it (*cf.* Table 41.9):

$$1.333 \times \frac{76.85}{23.15} = 4.425 \text{ kg N}_2 = 3.522 \text{ m}^3 \text{ nitrogen}$$

Hence the total weight of air required is: 5.758 kg air = 4.455 m³, all these volumes of oxygen, nitrogen or air being reckoned at 0°C and 760 mm.

Since the composition of bagasse is not strictly constant, it is futile to retain decimals which have no real significance. We shall say then:

$$\text{Total weight of air necessary} = 5.76 \text{ kg} \tag{41.27}$$

$$\text{Total volume of air necessary} = 4.45 \text{ m}^3 \text{ (at } 0^\circ\text{C and 760 mm)}$$

and the quantity of water formed is:

$$0.065 + 0.520 = 0.585 \text{ kg water} = 0.728 \text{ m}^3 \text{ water vapour} \tag{41.28}$$

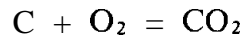
(B) Combustion of wet bagasse with excess air

We can now pass on to the general case.

It is not possible in practice to burn a fuel in industrial conditions by supplying only the quantity of air theoretically necessary: combustion would be poor and incomplete. In order to obtain complete combustion, without unburnt, and so that all the carbon burns to the form of CO_2 , it is necessary to supply a certain excess of air. A large proportion of the available heat would be lost if part of the carbon is allowed to burn only to the form of CO . The reaction:



liberates only 2.415 kcal/kg carbon, whereas the normal combustion:



releases 8,000 kcal/kg

We shall write:

Moisture of bagasse relative to unity	w
Ratio of weight of air used to weight theoretically necessary	m

and we shall designate by:

P_a	= weight	V_a	= volume of air used per kg of bagasse
P_g	= weight	V_g	= volume of the gaseous products of combustion
P_s	= weight	V_s	= volume of the gases (assumed dry)

all these volumes being reduced to 0°C and 760 mm.

In combustion calculations, to specify that the volumes are reckoned at 0°C and 760 mm, *it is customary to insert the letter N preceding the figure for volume: e.g. 5 Nm³ means 5 m³ measured at 0°C and 760 mm atmospheric pressure.*

(a) P_a . We have:

$$P = 5.76 (1 - w) m \quad (41.29)$$

(b) P_g . And in consequence:

$$P_g = 5.76 (1 - w) m + 1 \quad (41.30)$$

since the products of combustion consists of: (1) the unit weight of fuel the combustion of which we are studying; (2) the air for combustion which serves to burn this fuel.

(c) P_{gs} . We shall obtain this quantity by deducting from P_g the water formed by combustion of the hydrogen and the water of composition of the bagasse (*cf.* 41.28):

$$P_{gs} = P - 0.585 (1 - w) - w \quad (41.31)$$

or

$$P_{gs} = (1 - w) (5.76m + 0.415) \quad (41.32)$$

(d) V_u . We have similarly:

$$V_a = 4.45 (1 - w)m \text{ Nm}^3/\text{kg} \quad (41.33)$$

(e) V_g . To calculate V_g , we must note:

(1) That we have more oxygen than that strictly necessary to burn all the carbon and hydrogen in the bagasse

(2) That:

volume of CO_2 = volume of oxygen from which it is formed, and

volume of H_2O = volume of oxygen from which it is formed, multiplied by 2.

The combustion thus gives, per kg of dry combustible (see above), the quantities given in Table 41.11, for a quantity $(1 - w)$ of dry substance.

It is also necessary to add the volume of water vapour originating in the moisture contained in the bagasse. We have finally:

$$V_g = 4.45 (1 - w)m + 0.672 (1 - w) + 1.244 w \quad (\text{NmVkg}) \quad (41.34)$$

or:

$$V_g = 4.45 (1 - w)m + 0.572 w + 0.672 \quad (\text{NmVkg}) \quad (41.35)$$

TABLE 41.11

VOLUME OF GASEOUS PRODUCTS OF COMBUSTION

V_g = volume of air introduced	Air
+ volume of oxygen originating in the bagasse	+ 0.308
- volume of O_2 used to form water	- 0.364
- volume of O_2 used to form CO_2	- 0.877
+ volume of water formed	+ 0.728
+ volume of CO_2 formed	+ 0.877
<hr/>	
giving as total: volume of air introduced	+ 0.672

(f) V_{gs} . To obtain this quantity, it is sufficient to subtract from expression (31.34) the water of composition, $1.244 w$, and the water formed by combustion, $0.728 (1 - w)$. There remains:

$$V_{gs} = 4.45 (1 - w)m + 0.672 (1 - w) - 0.728 (1 - w)$$

or:

$$V_{gs} = 4.45 (1 - w)m - 0.056 (1 - w) \quad (\text{NmVkg}) \quad (41.36)$$

Comment. All the volumes given above have been calculated at 0°C and 760 mm of mercury. To obtain the volumes at any temperature t , it will be necessary to apply Mariotte's law: $pV = RT$ (where $T = 273 + t$). Since R is a coefficient and p is constant (atmospheric pressure), we have:

$$v_t = v_0 \frac{273 + t}{273} \quad \left(v_0 \frac{459.4 + t}{459.4 + 32} \right) \quad (41.37)$$

v_t = volume at temperature t

v_0 = volume at 0°C.

Composition of flue gases

We know the total weight of gases:

$$P_g = 5.76 (1 - w)m + \dots \quad (41.30)$$

The weight of the individual gases is given by:

(a) Nitrogen:

$$N_2 = 1.333 \times \frac{76.85}{23.15} (1 - w)m \quad \text{or} \quad N_2 = 4.43 (1 - w)m$$

(b) Oxygen:

O ₂ derived from the air:	1.333 (1 - w)m
+ O ₂ derived from the bagasse:	0.440 (1 - w)
- O ₂ used to form water:	-0.520 (1 - w)
- O ₂ used to form CO ₂ :	- 1.253 (1 - w)

or: $O_2 = 1.33 (1 - w) (m - 1)$

(c) Water:

Water formed:	0.585 (1 - w)
+ water of constitution:	w

$$H_2O = 0.585 (1 - w) + w$$

(d) Carbon dioxide:

$$CO_2 = 0.47 \times 3.67 (1 - w) \quad CO_2 = 1.72 (1 - w)$$

Substituting the values of m and w , and dividing by P_g , we may readily calculate the proportion by weight of each of these constituents in the flue gases.

Example. If $w = 0.48$ and $m = 1.5$:

$$P_g = (5.76 \times 0.52 \times 1.5) + 1 = 5.49 \text{ kg}$$

and:

N_2	4.42×0.52	$\times 1.5$	$= 3.455 \text{ kg}$	or	63.1%
O_2	1.33×0.52	$\times 0.5$	$= 0.346 \text{ kg}$	or	6.3%
H_2O	(0.585×0.52)	$+ 0.48$	$= 0.784 \text{ kg}$	or	14.3%
CO_2	1.72×0.52		$= 0.894 \text{ kg}$	or	16.3%
			5.479 kg		100%

The slight difference between the total and P_g is due to the ash content, as the sum of the components C + H + O of the bagasse which we have assumed (41.13) does not amount to 1 kg, while we have taken 1 kg in eqn. (41.39). To be absolutely accurate, it would have been necessary to take $1 - \epsilon$.

CO₂ content of flue gas

We have just seen the composition of the combustion gases by weight. Their composition by volume is equally interesting, since their CO₂ content gives information on the excess air used.

The quantity of CO₂ formed by combustion of 1 kg of dry bagasse is constant, since the carbon content of the bagasse is assumed to be constant (C = 47% approximately). If only the theoretical quantity of air were used, the CO₂ content of the flue gas would be at a maximum; if there is excess air, the quantity of CO₂ will remain constant in a volume of air which becomes greater as the excess air becomes higher. The proportion of CO₂ will therefore decrease as the excess air increases.

Simple types of apparatus are used which give a determination of the percentage of CO₂ in the flue gases. We seek the relation between this percentage and the quantity of excess air. We shall assume that we are dealing with an apparatus giving the CO₂ as percentage of the dry gases, *i.e.* that it condenses or fixes the water vapour contained in the gases before analysing them (if we use an apparatus giving the percentage of CO₂ in the wet gas, this percentage will be appreciably lower).

The total volume of the dry gases V_{gs} is given by eqn. (41.36). The volume of CO₂ which these gases contain is obtained immediately by means of the weight of CO₂ found above:

$$\text{Volume of } CO_2 \text{ in flue gas} = 1.72 (1 - w) \times 0.506 = 0.87 (1 - w)$$

Since the coefficient of expansion is the same for all gases, the proportion calculated from volumes at 0°C will remain the same at any temperature of the measurement.

The CO₂ content of the flue gas by volume is therefore:

$$\gamma = \frac{0.87 (1 - w)}{V} = \frac{0.87 (1 - w)}{4.45 (1 - w)m - 0.056 (1 - w)} \quad (41.38)$$

Hence we may derive m :

$$m = \frac{0.196}{y} + 0.0126 \quad (41.39)$$

m = ratio of weight of air used to weight of air theoretically necessary

y = CO_2 content per unit volume of dry gases.

Table 41.12 gives several values of y for the corresponding values of m .

TABLE 41.12

RELATION BETWEEN CO_2 CONTENT OF FLUE GASES (y) AND EXCESS AIR (m)

y	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
m	3.27	2.81	2.46	2.18	1.97	1.79	1.64	1.52	1.41	1.32	1.23	1.16

Optimum proportion of CO_2 . The theoretical maximum CO_2 content of the flue gases is 19.8%, according to eqn. (41.39). If it is desired to obtain complete combustion, without appreciable formation of CO, it is necessary to work with a minimum of excess air.

Eigenhuis²⁰ considers, following his experience in Java and in Queensland, that it is possible to maintain an average of 15% CO_2 without solid or gaseous unburnt. Later tests with modern boiler furnaces in Queensland have reported up to 16% CO_2 without unburnt.

Shillington²¹ considers that good combustion can be obtained only with a CO_2 content maintained between 10 and 14%. Above 14%, the proportion of CO would become much too high.

In south Africa, Hayes²² considers that heat losses due to excess air are not serious so long as the CO_2 content is maintained above 12%; but below that figure the loss in efficiency becomes considerable. Conversely, above 14%, he confirms having found the presence of CO in the gas. He considers then that good combustion corresponds to a CO_2 content lying between 12 and 14%.

In Java²³, the mean excess air was 63%.

In Jamaica, Davies²⁴ considers that the optimum excess air in practice corresponds to 50%.

If in addition to these authoritative opinions we consider results of tests carried out in various sugar countries, and take particular note of modern furnaces and boilers, where the air supply is better regulated, we may state that the best boiler efficiencies are obtained for:

$$\begin{array}{l} \text{a } \text{CO}_2 \text{ content of } 12 - 16\% \\ \text{an excess air of } 25 - 60\% \end{array} \quad (41.40)$$

Loss in efficiency due to CO. It is considered in South Africa²² that each per cent of CO in the gases of combustion corresponds to a loss of heat of 4.5%. In Cuba²⁵, it is indicated with more precision: 4.36% of the calorific value of the bagasse.

Calculation of combustion temperature

The combustion temperature T prevailing in the bagasse furnace is readily calculated from the fact that the heat developed in the combustion is recovered in the gases passing from the furnace to the boiler.

Since the calorific value and all the heat quantities are expressed with reference to a basic temperature of 0°C , it is necessary to take into account the heat already stored in the fuel and in the combustion air, at an ambient temperature t_0 .

We have then, for 1 kg of combustible:

$$\int_0^{t_0} 1 c_c dt + \int_0^{t_0} P_a c_a dt + N_i = E \int_{J_0}^T P c dt \quad (41.41)$$

t_0 = ambient temperature at which the air and the fuel arrive in the furnace

T = combustion temperature sought

P_a = weight of air used/kg fuel

P = weight of each of the gaseous products of combustion/kg combustible

c_c = specific heat of combustible

c_a = specific heat of the air

c = specific heat of each of the gaseous products

N_i = lower calorific value of the combustible.

The nett calorific value is employed, since in practice the water vapour contained in the gases is not condensed.

The expression $\Sigma \int_0^T P C dt$ represents the sum of the heat capacities of the different gases constituting the flue gas.

It is necessary to use the integral since the specific heat is a function of temperature. We may avoid integrals by using Table 41.13, which gives the mean specific heat of the various substances with which we have to deal, between 0°C and any temperature t or T . With the aid of this table we can readily read off the mean specific heat between any two temperatures t and T , at atmospheric pressure.

Equation (41.41) then becomes:

$$(1 c_c + P_a c_a) t_0 + N_i = T [\Sigma P c]_0^T \quad (41.42)$$

But:

$$1 + P_a = \Sigma P = P_g$$

and, since these specific heats of air and the gases do not differ greatly, we may take:

$$1 c_c + P_a c_a = \Sigma P c \quad (41.43)$$

Finally, taking losses into account:

$$T = t_0 + \frac{\alpha \beta_0 N_i}{[L P c]_0^T} \quad (41.44)$$

TABLE 41.13
COMBUSTION OASES (Habif²⁶)

Temp. °C	Specific heat at T°				Mean sp. heat between 0 and T°				Total heat to heat from 0 to T° at const. press. (kcal/kg)*			
	CO ₂	H ₂ O	N ₂ , CO	O ₂	CO ₂	H ₂ O	N ₂ , CO	O ₂	CO ₂	H ₂ O	N ₂ , CO	O ₂
0	0.199	0.468	0.246	0.214	0.199	0.468	0.246	0.214	0	0	0	0
50	0.207	0.483	0.248	0.216	0.203	0.476	0.247	0.215	10	23	12	10
100	0.215	0.499	0.250	0.218	0.207	0.484	0.248	0.216	20	48	24	21
150	0.224	0.515	0.252	0.220	0.211	0.492	0.249	0.217	31	73	37	32
200	0.232	0.530	0.254	0.222	0.215	0.499	0.250	0.218	43	100	50	43
250	0.240	0.546	0.256	0.223	0.219	0.507	0.251	0.219	55	126	62	54
300	0.248	0.562	0.258	0.225	0.224	0.515	0.252	0.220	67	154	75	66
350	0.256	0.577	0.260	0.227	0.228	0.523	0.253	0.221	79	183	88	77
400	0.264	0.593	0.262	0.229	0.232	0.530	0.254	0.222	92	212	101	88
450	0.272	0.608	0.264	0.231	0.236	0.538	0.255	0.223	106	242	114	100
500	0.281	0.624	0.266	0.232	0.240	0.546	0.256	0.223	120	273	128	111
550	0.289	0.639	0.268	0.234	0.244	0.554	0.257	0.224	134	304	141	123
600	0.297	0.655	0.270	0.236	0.248	0.562	0.258	0.225	149	337	154	135
650	0.305	0.670	0.272	0.238	0.252	0.569	0.259	0.226	164	370	168	147
700	0.313	0.686	0.274	0.240	0.256	0.577	0.260	0.227	179	404	182	159
750	0.321	0.701	0.276	0.241	0.260	0.585	0.261	0.228	195	439	195	171
800	0.330	0.717	0.278	0.243	0.264	0.593	0.262	0.229	211	475	209	183
850	0.338	0.732	0.280	0.245	0.268	0.600	0.263	0.230	228	510	223	195
900	0.346	0.748	0.282	0.247	0.272	0.608	0.264	0.231	245	547	237	208
950	0.354	0.764	0.284	0.248	0.277	0.616	0.265	0.232	263	585	251	220
1,000	0.362	0.779	0.286	0.250	0.281	0.624	0.266	0.232	281	624	266	232
1,050	0.370	0.795	0.288	0.252	0.285	0.631	0.267	0.233	299	663	280	245
1,100	0.378	0.810	0.290	0.254	0.289	0.639	0.268	0.234	318	703	294	258
1,150	0.387	0.826	0.292	0.256	0.293	0.647	0.269	0.235	337	744	309	270
1,200	0.395	0.841	0.294	0.258	0.297	0.655	0.270	0.236	356	786	324	283
1,250	0.403	0.857	0.296	0.259	0.301	0.663	0.271	0.237	376	828	338	296
1,300	0.411	0.872	0.298	0.261	0.305	0.670	0.272	0.238	397	872	353	309
1,350	0.419	0.888	0.300	0.263	0.309	0.678	0.273	0.239	417	916	368	322
1,400	0.427	0.903	0.302	0.265	0.313	0.686	0.274	0.240	439	960	383	336
1,450	0.435	0.919	0.304	0.267	0.317	0.694	0.275	0.241	460	1,006	398	349
1,500	0.444	0.935	0.306	0.268	0.321	0.701	0.276	0.241	482	1,052	414	362
1,550	0.452	0.950	0.308	0.270	0.325	0.709	0.277	0.242	505	1,099	429	376
1,600	0.460	0.966	0.310	0.272	0.330	0.717	0.278	0.243	528	1,147	444	389
1,650	0.468	0.981	0.312	0.274	0.334	0.725	0.279	0.244	551	1,196	460	403
1,700	0.476	0.997	0.314	0.276	0.338	0.732	0.280	0.245	574	1,246	476	417
1,750	0.484	1.012	0.316	0.277	0.342	0.740	0.281	0.246	598	1,296	491	431

* To convert to B.T.U./lb., multiply by 1.8.

a = coefficient taking into account unburnt solids

β_0 = coefficient taking into account losses by radiation in the furnace.

We may take for bagasse furnaces:

$a = 0.98-0.99$, according to draught, for step-gate, hearth or spreader-stoker furnaces
 $\alpha = 0.94-0.95$ for Ward furnaces
 $\beta_0 = 0.98 - 0.99$.

We may comment that:

- (1) The combustion temperature increases as t_0 increases
- (2) It decreases as ΣPc increases. In other words excess air has a marked influence on the temperature of combustion.

In the same way, the moisture of the bagasse will similarly lower this temperature greatly, on account of the additional water vapour present, and all the more so since the specific heat of water vapour is nearly double that of the other gases. Further, it must still not be forgotten (*cf.* eqn. 41.20) that the moisture also reduced TV, and consequently has a double effect on the temperature of combustion.

Example. We shall take again the example on p. 929 and calculate EPc for the composition of flue gases found. We shall assume:

$$\left. \begin{array}{l} t_0 = 30^\circ\text{C} \\ \alpha = 0.98 \\ \beta_0 = 0.99 \\ \text{TV} = 1,900 \text{ kcal} \end{array} \right\} \alpha\beta_0 N_i = 1,843 \text{ kcal/kg}$$

We obtain EPc by determining, for each of the products of combustion, the corresponding term $(T - t_0) PC$. The simplest method is to operate by interpolation, which is very quickly done when we already have some idea of the temperature sought (Table 41.14).

TABLE 41.14

	$T = 1,050^\circ\text{C}$	$T = 1,100^\circ\text{C}$
N ₂	3.455 kg (280 - 7) = 943 kcal	3.455 kg (294 - 7) = 992 kcal
O ₂	0.346 (245 - 6) = 83	0.346 (258 - 6) = 87
H ₂ O	0.784 (663 - 14) = 509	0.784 (703 - 14) = 540
CO ₂	0.894 (299 - 6) = 262	0.894 (318 - 6) = 279
Ash	0.025 x 0.2 x 1,050 = 5	0.025 x 0.2 x 1,100 = 6
	$[\Sigma Pc]_{t_0}^T (T - t_0) = 1,802$	$[\Sigma Pc]_0^T (T - t_0) = 1,904$

Equation (41.44) may be written:

$$(T - t_0) [\Sigma Pc]_{t_0}^T = \alpha\beta_0 N_i \quad (41.45)$$

The first term of this equation is the heat required to raise the flue gases from t_0 to T . We require to know for what value of T the heat required is equal to the heat produced, which forms the second term in the equation. If it is considered for example that the temperature

T required lies between 1,050 and 1,100°C, the calculation will be made for these two temperatures with subsequent interpolation. As the ash content is very low, its specific heat may be assumed as 0.2 (Table 41.14). Interpolating for 1,843 kcal, we find:

$$T - t_0 = 1,070^\circ\text{C} (1,958^\circ\text{F}) \quad T = 1,070 + 30 = 1,100^\circ\text{C} (2,012^\circ\text{F})$$

It will be seen that a change of one point in excess air (e.g., 1.50 replaced by 1.49) involves an increase in temperature of 4–5°C.

Mean specific heat. We may dispense with Table 41.13 by utilising the formulae given in Table 41.15.

TABLE 41.15

MEAN SPECIFIC HEAT OF COMBUSTION GASES BETWEEN 0°C (or 30°C) AND $T^\circ\text{C}$

Nitrogen	$0.246 + 0.000020 T$
Oxygen	$0.214 + 0.000018 T$
Water vapour	$0.468 + 0.000155 T$
CO ₂	$0.199 + 0.000082 T$

It will then be necessary to multiply by T the mean specific heat found in order to obtain the heat necessary to raise the temperature of the gas under consideration from 0°C (or 30°C) to T° .

No appreciable error is involved if the mean specific heat between 0°C and T° is used in place of that between 30°C and T° .

Since the composition of the flue gases varies only between relatively narrow limits, we may speak of the mean specific heat of the mixed gases. We may thus take approximately:

For mixed combustion gases from bagasse:

$$\text{Mean sp. heat} = \begin{pmatrix} 0.27 + 0.00006 T (T \text{ in } ^\circ\text{C}) \\ 0.27 + 0.00003 T (T \text{ in } ^\circ\text{F}) \end{pmatrix} \quad (41.46)$$

or, roughly:

$$\text{M.S.H.} = 0.3 \quad (41.47)$$

Measurement of furnace temperature

The furnace temperature is measured with the aid of pyrometers or Seger cones. It may also be estimated by eye with a certain degree of accuracy, particularly if the observer has trained himself to compare his estimates with pyrometer readings.

Temperatures obtained in practice

The most inefficient old furnaces give 800–1,000°C (1,500–1,800°F). The highest temperatures recorded are 1,290°C (2,350°F) for a few minutes, and 1,250°C (2,275°F) con-

tinuously. The commonest temperatures are approximately $1,100^{\circ}\text{C}$ ($2,000^{\circ}\text{F}$). Table 41.17 summarises approximately the influence of excess air and bagasse moisture on furnace temperature. The table assumes an ambient temperature of 30°C (86°F) and a product $\alpha\beta_0$ of 0.96.

TABLE 41.16

COLOUR SCALE OF TEMPERATURE		
	$^{\circ}\text{C}$	<i>op</i>
Dazzling white	1,500	2,700
Welding heat	1,400	2,500
White	1,300	2,400
Straw yellow	1,200	2,200
Pale yellow	1,150	2,100
Golden yellow	1,100	2,000
Orange	1,050	1,900
Light cherry red	1,000	1,800
Cherry red	900	1,650
Incipient cherry red	800	1,500
Dark red	650	1,200

The colour of the furnaces is more brilliant as the temperature increases (see Table 41.16).

TABLE 41.17

<i>Moisture in bagasse</i>	COMBUSTION TEMPERATURE IN BAGASSE FURNACES $^{\circ}\text{C}$ ($^{\circ}\text{F}$)					
	<i>Excess air</i>					
	<i>30% (m = 1.3)</i>		<i>50% (m = 1.5)</i>		<i>75% (m = 7.75;</i>	
	<i>($^{\circ}\text{C}$)</i>	<i>($^{\circ}\text{F}$)</i>	<i>($^{\circ}\text{C}$)</i>	<i>($^{\circ}\text{F}$)</i>	<i>($^{\circ}\text{C}$)</i>	<i>($^{\circ}\text{F}$)</i>
45% (<i>w</i> = 0.45)	1,185	2,165	1,100	2,000	1,000	1,830
50% (<i>w</i> = 0.50)	1,120	2,050	1,040	1,900	950	1,740

Heat losses in flue gases

Our formula (41.25), giving the nett calorific value of the bagasse, already takes into account the loss of heat of the water vapour passing with the gases to the chimney.

The most important of the losses which remain to be accounted for is that corresponding to the sensible heat lost in these flue gases, and we shall seek to evaluate this loss.

We know the composition of the flue gases, and the specific heat of its component gases. We thus have immediately the corresponding heat loss.

The mean specific heat of the flue gases between 0°C and the flue gas temperature varies only slightly because this temperature itself is limited. In a modern installation, with economiser or air heater, it is easy to obtain flue gas temperatures below 200°C , for example 175°C ; but, although it is possible to reach 130°C , there is little interest in going below 150°C , which may be considered as the lower economic limit.

Conversely, it would be only a very old or inadequate installation which would allow the gases to leave at more than 300°C.

We shall therefore not involve any appreciable error if we take the mean specific heat between 0°C and the flue gas temperature, as equal to the specific heat at 100°C.

(We may comment that our reference temperature should be 0°C, and not the ambient air temperature, since the calorific value is calculated as from 0°C, and hence it is necessary to refer the whole heat balance to this same basic temperature.)

Calculation. Starting from the weights found for the components of the flue gases (*cf.* p. 928), the sensible heat q carried by each of these gases will be, per pound of bagasse burnt:

$$\begin{aligned} \text{N}_2 \quad q_1 &= 4.43 (1 - w)m \times 0.250t \\ \text{O}_2 \quad q_2 &= 1.33 (1 - w) (m - 1) \times 0.218t \\ \text{H}_2\text{O} \quad q_3 &= [0.585 (1 - w) + w] 0.499t \\ \text{CO}_2 \quad q_4 &= 1.72 (1 - w) \times 0.215t \end{aligned}$$

Adding, simplifying, and approximating to avoid retaining figures without practical significance, we obtain the total sensible heat lost (reckoned as from 0°C):

$$\begin{aligned} & m. \text{ units} \\ q &= [(1 - w) (1.4m - 0.13) + 0.5]t \end{aligned} \tag{41.48}$$

$$\begin{aligned} & Br. \text{ units} \\ q &= [(1 - w) (1.4m - 0.13) + 0.5]t - 32 \end{aligned}$$

q = sensible heat lost in flue gases, in kcal/kg (B.T.U./lb.) of bagasse

$?$ = temperature of the flue gases, in °C (°F)

w = moisture per unit bagasse

m = ratio of weight of air used for combustion to weight theoretically necessary.

Example. For: $m = 1.4$, $w = 0.48$, $? = 200^\circ\text{C}$, we shall have:

$$q = [0.52 (1.4 \times 1.4 - 0.13) + 0.5] 200 = 290 \text{ kcal/kg of bagasse} \quad (534 \text{ B.T.U./lb.})$$

or approximately 15% of N.C.V. of the bagasse 1,900 kcal/kg (3,400 B.T.U./lb.)

It will be seen that with the values chosen for m and w in this example, which are good typical values, we have to a close approximation:

$$\begin{aligned} & m. \text{ units} & & Br. \text{ units} \\ q &= 1.5? & & q = 1.5 (? - 32) \end{aligned} \tag{41.49}$$

Quantity of steam obtainable

We may now calculate the quantity of steam which we can obtain from unit weight of bagasse. The losses of heat in the furnace and at the boiler consist of the following:

- (a) Latent heat of the water formed by combustion of hydrogen in the bagasse
- (b) Latent heat of the water contained in the bagasse
- (c) Sensible heat of the flue gas leaving the boiler
- (d) Losses in unburnt solids
- (e) Losses by radiation from the furnace and especially from the boiler
- (f) Losses due to bad combustion of carbon giving CO instead of CO₂.

Now the use of the nett calorific value (formula 41.25) has already taken into account losses (a) and (b).

The loss (c) is given by eqn. (41.48).

The three other losses are taken into account by means of coefficients applied to the total quantity of heat which is still available after the first three losses:

a = coefficient taking into account losses by unburnt solids

β = coefficient taking into account losses due to radiation

η = coefficient taking into account losses due to incomplete combustion.

The quantity of heat remaining to be transferred to the steam is therefore given by the expression:

$$\begin{aligned}
 & \text{m. units} \\
 M_v &= (4,250 - 1,200s - 4,850w - q) \alpha\beta\eta \\
 & \text{Br. units}
 \end{aligned}
 \tag{41.50}$$

$$M_v = (7,650 - 2,160s - 8,730w - q) \alpha\beta\eta$$

M_v = heat transferred to the steam per kg (lb.) of bagasse burnt, in kcal (B.T.U.)

w = moisture of bagasse

s = sucrose per unit bagasse

q = sensible heat of flue gases (eqn. 41.48).

a (solid unburnt) is of the order of 0.98 for ordinary furnaces. It rarely descends below 0.97, unless a high draught is used, carrying away to the chimney relatively large pieces of bagasse. For Ward furnaces, Fives Cail – Babcock estimates the losses at 5.5 (normal operation) to 6% (forced rating), or: $a = 0.940 - 0.945$. For spreader-stoker furnaces, a figure of 2.5% may be used, or $a = 0.975$.

β (radiation and convection) varies from 0.97 to 0.995 according to the more or less efficient lagging of the boiler. Fives Cail – Babcock reckon: $\beta = 0.99$ (normal rating) to 0.995 (forced rating). For a well-lagged boiler, a value of 0.995 may be used.

η (imperfect combustion) may vary from 0.99 down to 0.80. This coefficient will be better with:

(1) Lower bagasse moisture

(2) Lower excess air

(3) Higher furnace temperature (this condition moreover is a consequence of the two preceding ones).

This coefficient falls rapidly when the moisture in bagasse exceeds 50%, or similarly when the furnace temperature falls below 900°C (1,650°F). In a good modern furnace, it easily exceeds 0.96. For a well-conducted combustion, we may use a figure of 0.98 – 0.99.

Overall efficiency. The ratio:

$$e = \frac{M_v}{N} = \frac{\text{Heat units transferred to the steam}}{\text{G.C.V. of the bagasse}} \quad (41.51)$$

is called the overall efficiency of the boiler.

This efficiency generally varies between 50 and 65%. Tromp²⁷ gives the figures shown in Table 41.18.

TABLE 41.18

BOILER EFFICIENCIES (TROMP²⁷)

Type of boiler	Type of grate	Overall efficiency
Elephant boiler	Step-grate	50%
Water-tube	Horseshoe	60%
Water-tube	Ward furnace	66%

Tromp reported that this value of 66% was the highest which he had encountered. It corresponds to a value of the coefficient η of about 0.99.

At the time of the Second World War, it was considered in Cuba²⁵ that the overall efficiency never exceeded 61.3% and attained that value only in the best installations. Modern spreader-stoker furnaces give an efficiency of 83 - 88% on NCV when operated on bagasse, and 92 - 94% on fuel oil.

In Queensland, tests reported by Behne²⁸ gave the figures shown in Table 41.19.

TABLE 41.19

BOILER EFFICIENCIES (QUEENSLAND²⁸)

Type of boiler	kg steam/kg bagasse	Overall efficiency
Thompson	2.3	53.1
B. & W.	2.4	56.2
Semi-tubular	1.8	42.2

TABLE 41.20

BOILER EFFICIENCIES (QUEENSLAND 1961)

	Overall efficiency (%)	
	On G.C.V.	On N.C.V.
Typical mean value in Australia	50	60
Modern installations	63	78
Modern well-operated installation	65	80
Best test figures	68	83

More recent tests in Queensland, reported by Jenkins²⁹, give figures of 2.9 kg steam/kg bagasse and overall efficiency of 68% on G.C.V. for a boiler with a large combustion chamber and large air heater.

Jenkins³⁰ summarised performance in 1961 as in Table 41.20.

Weight of steam per unit weight of bagasse. Table 41.21 gives figures for the total heat required to produce unit weight of steam, for different steam conditions obtaining in the factory.

TABLE 41.21

HEAT REQUIRED TO PRODUCE UNIT WEIGHT OF STEAM
(Temperature of feed water 90°C)

<i>Steam pressure</i>		<i>Steam temperature</i>		<i>Heat required</i>	
<i>kg/cm²</i>	<i>p.s.i.</i>	<i>°C</i>	<i>°F</i>	<i>kcal/kg</i>	<i>B.T.U./lb.</i>
6	85	Saturated		569	1,025
8	114	Saturated		572	1,030
10	142	Saturated		574	1,033
10	142	225	437	599	1,078
15	213	300	572	635	1,143
20	284	325	617	645	1,163
25	355	350	662	656	1,181
30	425	400	752	681	1,226
45	640	425	793	690	1,242

We have assumed in all cases a feed-water temperature of 90°C (194°F), which is a common temperature, within a few degrees, at the feed-water tank, before the water is pumped to the boiler or to the economiser; but operation at 105°C (221 °F) is now common with boiler pressures above 18 kg/cm² (250 p.s.i.).

The total quantity of heat from the bagasse which is effectively utilised and is recovered in the steam depends on four principal factors:

$$w m \eta t \text{ (temperature of flue gases)}$$

Adopting the following figures as average or readily obtainable values:

$$\begin{aligned} w &= 0.48 \\ m &= 1.4 \\ a &= 0.975 \\ \beta &= 0.975 \\ \eta &= 0.96 \end{aligned}$$

we shall have for M_v and for the weight of vapour obtainable per unit weight of bagasse the corresponding values given in Table 41.22.

TABLE 41.22

HEAT TRANSMITTED TO STEAM PER POUND OF **BAGASSE**, AND **WEIGHT** OF STEAM PRODUCED PER UNIT WEIGHT OF BAGASSE, FOR DIFFERENT TEMPERATURES t OF FLUE GASES
($w = 0.48$; $m = 1.4$; $a = 0.975$; $0 = 0.975$; $17 = 0.96$)

t ($^{\circ}\text{C}$)		150	180	200	250	300
t ($^{\circ}\text{F}$)		302	356	392	482	572
M_v (B.T.U./lb.)		2,772	2,700	2,653	2,534	2,416
<i>Steam produced at:</i>						
6 kg/cm ² satd.	85 p.s.i. satd.	2.71	2.64	2.59	2.47	2.36
8 kg/cm ² satd.	114 p.s.i. satd.	2.69	2.62	2.58	2.46	2.35
10 kg/cm ² satd.	142 p.s.i. satd.	2.68	2.61	2.57	2.45	2.34
10 kg/cm ² 250 $^{\circ}\text{C}$	142 p.s.i. 482 $^{\circ}\text{F}$	2.52	2.45	2.41	2.30	2.19
15 kg/cm ² 300 $^{\circ}\text{C}$	213 p.s.i. 572 $^{\circ}\text{F}$	2.43	2.36	2.32	2.22	2.11
20 kg/cm ² 350 $^{\circ}\text{C}$	284 p.s.i. 662 $^{\circ}\text{F}$	2.34	2.28	2.24	2.14	2.04
25 kg/cm ² 375 $^{\circ}\text{C}$	355 p.s.i. 707 $^{\circ}\text{F}$	2.30	2.24	2.20	2.10	2.00
30 kg/cm ² 400 $^{\circ}\text{C}$	427 p.s.i. 752 $^{\circ}\text{F}$	2.26	2.20	2.16	2.07	1.97
45 kg/cm ² 425 $^{\circ}\text{C}$	640 p.s.i. 797 $^{\circ}\text{F}$	2.23	2.17	2.14	2.04	1.94

These figures for steam production naturally vary according to the conditions, and particularly with the four factors indicated. In most cases, however, the weight of vapour produced per unit weight of bagasse lies within the range 2—2.7 and is often in the neighbourhood of 2.25.

According to the fibre content of the cane and the conditions of operation of the boiler station, the steam production available will thus be 45—75% of cane, and often 60—65%.

Calorific value of bagasse pith. Bordenave^{30a} gives as G.C.V. of bagasse at 12% moisture: 4056 kcal/kg, and for moist pith: 3778 kcal/kg. He gives the value for pith as 93% of that of the whole bagasse.

In Taiwan, Chou^{30b} reckons for dry pith: G.C.V. = 4400 kcal/kg and for pith at 6% hydrogen: N.C.V. $4400 - (0.06 \times 5400) = 4076$ kcal/kg (dry pith); N.C.V. = $4076 - 46.8w$ (wet pith). For dry bagasse in storage the moisture of the pith would be 15—22%.

FUELS OTHER THAN BAGASSE

Owing to insufficient fibre content in cane, excess moisture in the bagasse, a low efficiency at the boiler station, or wastage of steam, certain factories have not sufficient bagasse to produce all the steam which they require. They have therefore to use some supplementary fuel: firewood, coal, fuel oil, etc.

Wood

The N.C.V. of cellulose is fairly constant at about 4,250 kcal/kg (7,650 B.T.U./lb.). However, the water content of different woods varies greatly, according to the species (and even the variety) and the time which has elapsed since it was cut.

According to Izart³¹, a very dry wood contains 20% water, a dry wood 30% and a green wood 40—50%.

Firewoods generally used have between 30 and 40% water. Equation (41.20) with $5 = 0$, may be applied to firewood without great error.

Certain woods are very superior to others for fuel purposes. The tree *Casuarina equisetifolia* would have, at a given age, 5–10% less water than most other woods.

We may take approximately:

Ordinary wood moderately dry: N.C.V. = 2,500 kcal/kg (4,500 B.T.U./lb.)
 Casuarina moderately dry: N.C.V. = 3,250 kcal/kg (5,850 B.T.U./lb.)

In South Africa³² a value of 2,000 kcal/kg (3,600 B.T.U./lb.) only is taken for firewood. In Australia, Jenkins³³ gives 3,333 kcal/kg (6,000 B.T.U./lb.).

Coal and fuel oil

The calorific value of these two fuels is:

Coal: N.C.V. = 6,000–8,000 kcal/kg (11,000–14,000 B.T.U./lb.)
 Fuel oil: N.C.V. = 9,500–10,700 kcal/kg (17,000–19,000 B.T.U./lb.)

the difference between G.C.V. and N.C.V. being moreover rather small, except for coal having a high hydrogen content.

Fuel oil presents the advantage that it can be burnt in bagasse furnaces, by means of special burners. The width of the furnace in the direction of the flame, and the quality of the refractories should accordingly be designed with this in view.

Figures quoted are:

In South Africa³²: 10,300 kcal/kg (18,500 B.T.U./lb.) for fuel oil, and
 6,600 (12,000) for Natal coal
 In Australia³³: 10,550 kcal/kg (19,000 B.T.U./lb.) for fuel oil, and
 6,100 (11,000) for coal

Molasses

The calorific value of molasses, per unit weight of dry substance, is:

(N.C.V.) = 3,400–3,800 kcal/kg (6,000–7,000 B.T.U./lb.), according to Deerr³⁴

(G.C.V.) = 3,800 kcal/kg (6,840 B.T.U./lb.) (Prinsen Geerligs³⁵)

(G.C.V.) = 4,250 kcal/kg (7,650 B.T.U./lb.) (Naus Bey³⁶).

The latter value would correspond to an N.C.V. of 3,900 kcal/kg (7,020 B.T.U./lb.) We may reckon approximately, for an average molasses as fired:

$$\text{N.C.V.} = (375 - 500) \text{ kcal/kg} \quad (67B - 900 \text{ B.T.U./lb.}) \quad (41.52)$$

where B = brix of the molasses.

Provided it is heated and diluted (the latter practice, however, being most unfortunate from the point of view of fuel value), molasses may be burnt in burners similar to those used for fuel oil. It has the disadvantage of giving such quantities of ashes that their removal

sometimes poses a problem difficult to solve. Taking into account the efficiency, which is by no means the same as with bagasse, we may reckon approximately that one tonne of molasses is the equivalent of 150 kg of fuel oil, 260 kg of coal or 6.35 m³ of natural gas.

FURNACES

Bagasse feed

Older bagasse furnaces were provided with a rectangular hopper, a simple opening through the arch of the furnace. Through this wide open passage, air was drawn in with the bagasse. It encountered practically no resistance, unlike the combustion air, which was forced to pass through the grate and traverse the bed of bagasse. This air, passing through the top opening, entered the furnace directly, contributed practically nothing to the combustion, and served only to increase the excess air *greatly*.

To avoid this drawback, bagasse furnaces are fitted with a hopper and a rotary bagasse feeder (Fig. 41.9).

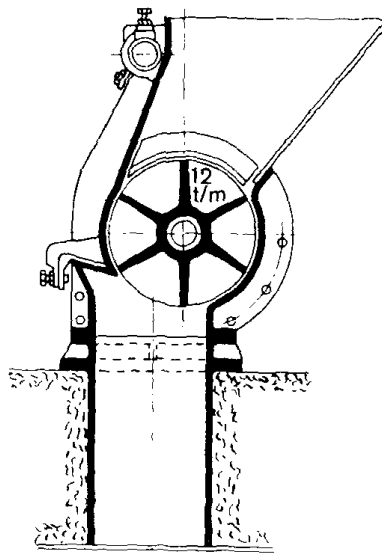


Fig. 41.9. Rotary feeder for grate- or Cook-type bagasse furnace.

Types of furnace

There are four *principal* types of bagasse furnace:

- (a) The step-grate furnace
- (b) The Cook or horseshoe furnace
- (c) The Ward furnace
- (d) The spreader-stoker furnace.

(a) Step-grate furnace (Fig. 41.10).

The grate consists of small plates of cast iron, arranged in steps.

Its inclination to the horizontal should be 52°, the value recommended by the Java experiment station, and by Tromp³⁷, and adopted by most manufacturing firms.

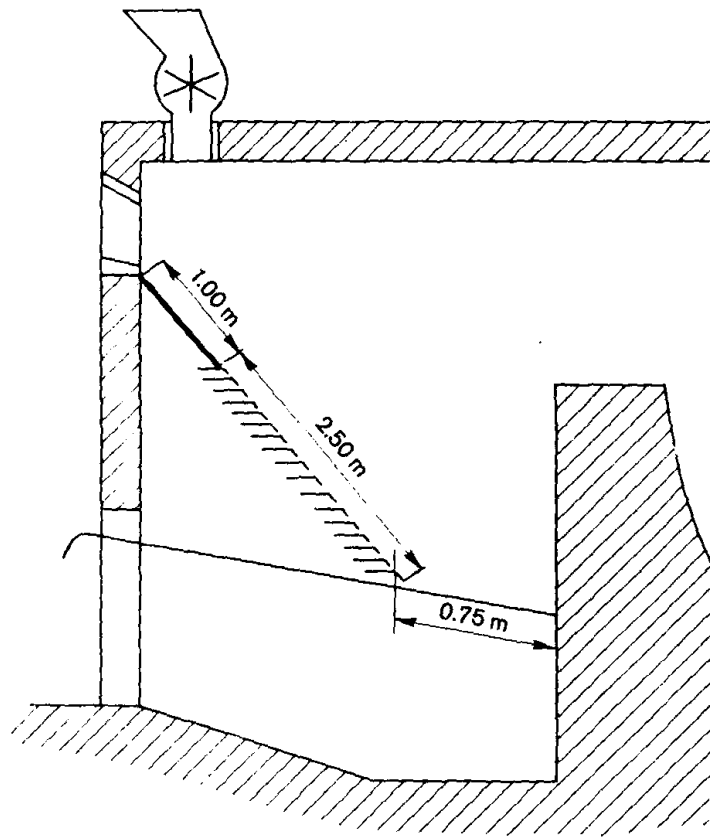


Fig. 41.10. Step grate.

The grate consists of three parts:

- (1) The upper part, or dead plate, without steps or openings for passage of air, and on which the bagasse is dried before passing on to the grate proper.
- (2) The grate proper, corresponding to the steps or grate bars. Certain manufacturers progressively increase the space between bars, so as to proportion the quantity of air passing through the bagasse to the degree of combustion required. Generally the bagasse ignites at the first bars of the grate, and burns from the first bars along the whole length of the grille proper.
- (3) The portion of slight slope, or ash grate, at the lower end of the grate, on which combustion of the bagasse is completed, leaving ashes, which fall between the bars of the grate, into the ashpit.

Certain manufacturers make this ash grate horizontal, but it is of advantage to give it a slight slope. Tromp³⁷ recommends 20°. French firms generally adopt 15°.

Figure 41.10 shows a sectional elevation of the step-grate furnace, with typical dimensions (in metric units); these of course will vary somewhat.

(b) Cook furnace (Fig. 41.11). This type of furnace is also described as the horseshoe furnace. It is more recent, and was more widely adopted at the expense of the step-grate.

The bagasse falls directly from the feed chute into the furnace, which takes the form of a tank in the shape of a horseshoe, and forms a conical heap (Fig. 41.12). By means of tuyeres placed all around the horseshoe, and more numerous near the bottom, air, which may

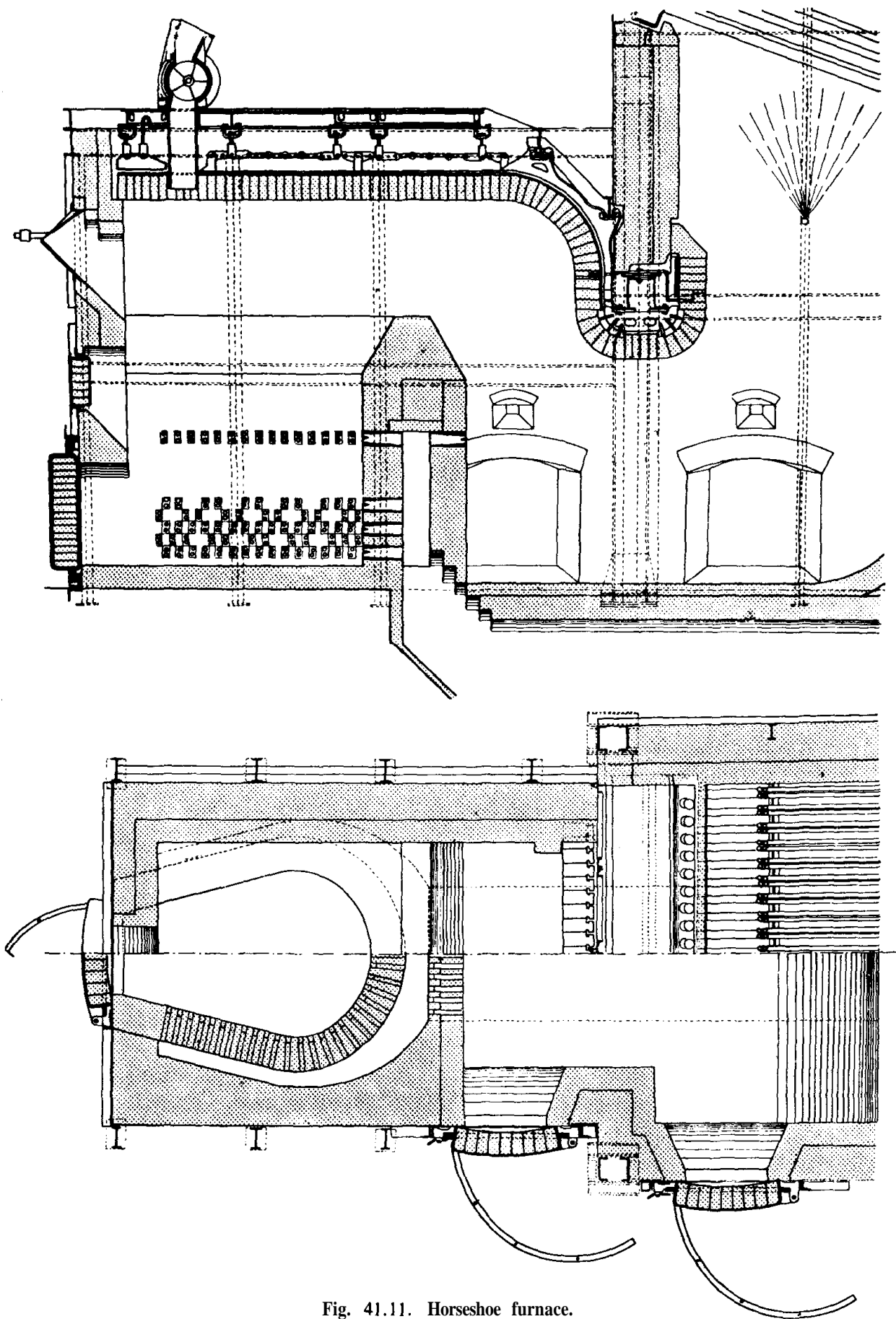


Fig. 41.11. Horseshoe furnace.

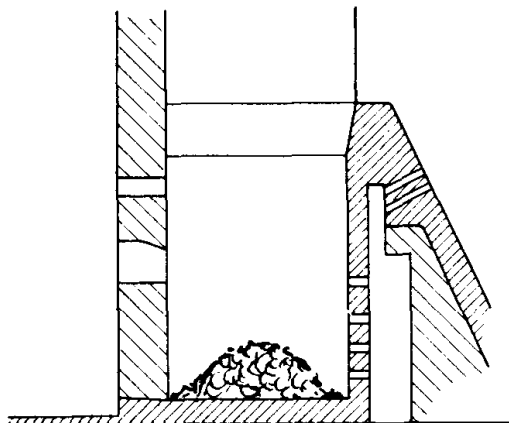


Fig. 41.12. Conical pile of bagasse in furnace.

be cold but is preferably preheated, is blown into the pile of bagasse, which burns away and collapses on itself. The ashes accumulate at the bottom of the furnace, the height of which should be designed accordingly.

The horseshoe furnace requires slightly more height than does the step-grate furnace, but has the advantage of dispensing with the grate bars and bearer bars. It permits of higher combustion rates and gives better results from the point of view of efficiency.

(c) **Ward furnace.** This type (Fig. 41.13), of American origin, closely resembles the

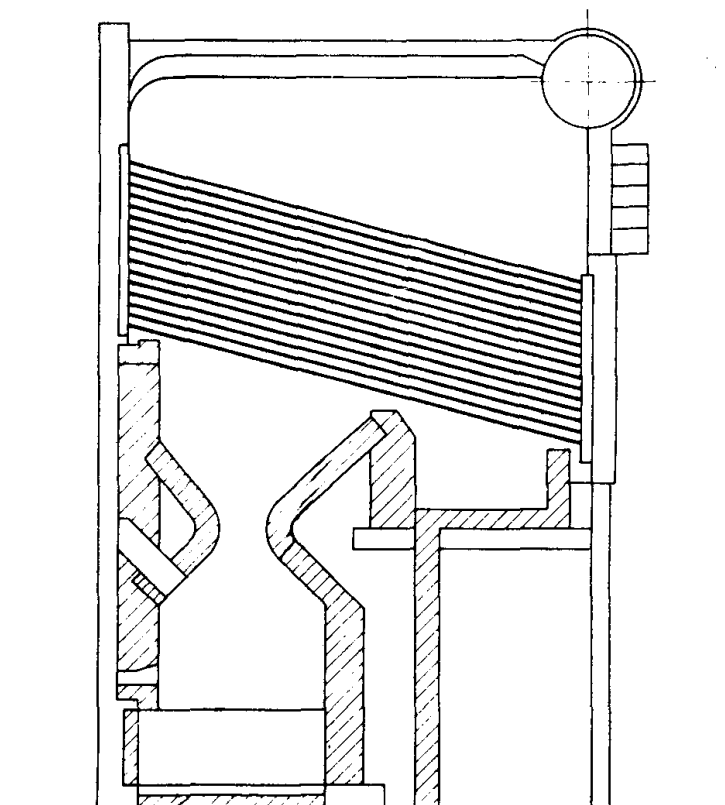


Fig. 41.13. Ward furnace. Straight-tube boiler (Babcock & Wilcox).

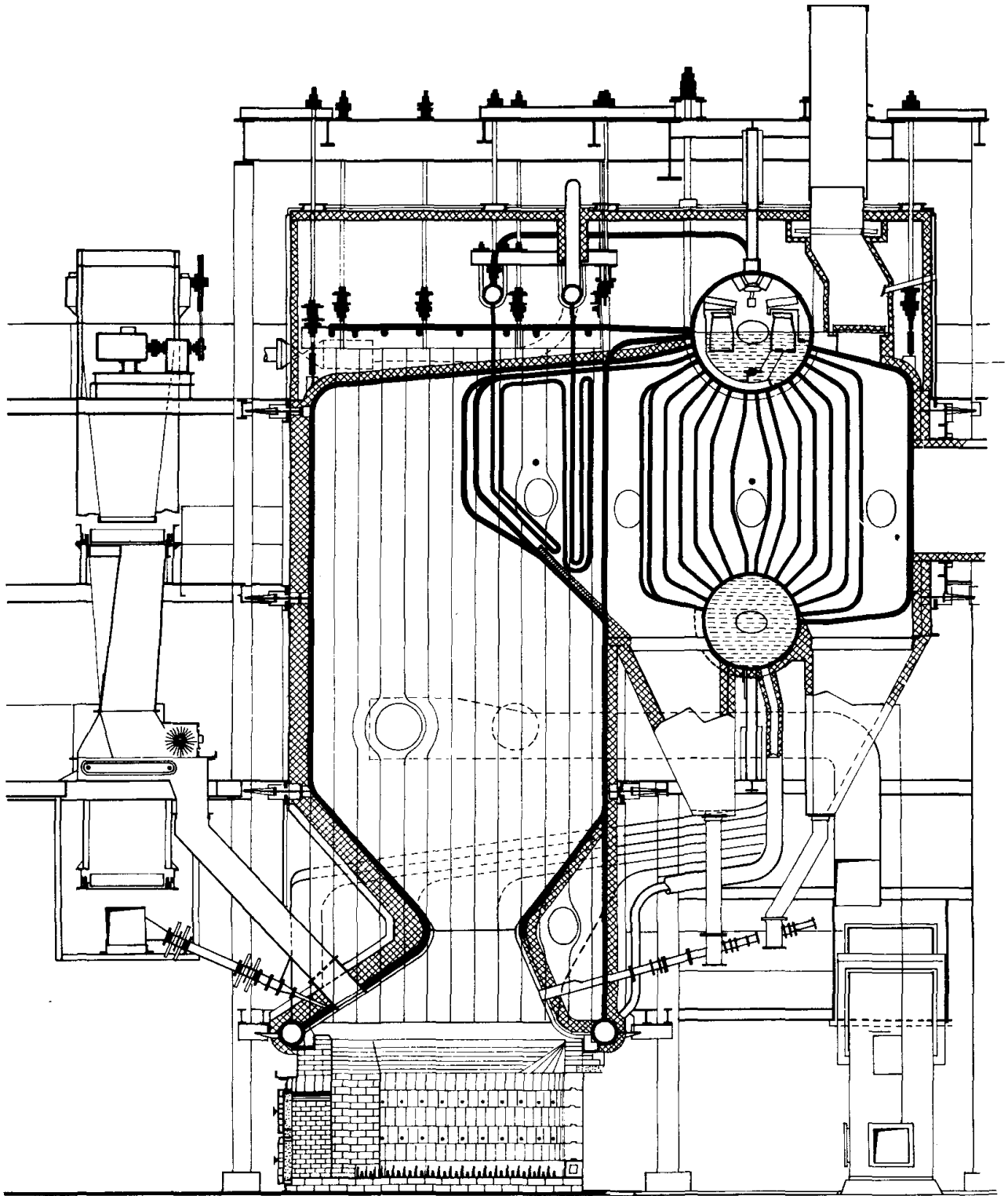


Fig. 41.14. Ward furnace. Bent-tube boiler (Babcock Atlantique).

horseshoe furnace. The hearth is very similar; it sometimes has a cast-iron plate with provision for circulation of a small proportion of the air, surmounted by an oblong furnace carrying three rows of tuyeres. The bagasse burns in a conical heap in the middle of the hearth.

The original feature of this furnace lies in its upper portion, and its advantage is its simplicity. The disadvantage of bagasse furnaces in general lies in the necessity to install a separate furnace in such a way that the heat of the furnace is concentrated on the burning bagasse, while the hot gases are made to follow a circuitous path before reaching the relatively cold heating surface of the boiler, which is placed out of "sight" of the furnace. With the Ward design, the furnace is placed squarely below the heating surface; and so that combustion will not suffer in consequence, a small inclined arch is interposed over the greater part of the vertical space immediately above the furnace, so as to reflect heat on the burning bagasse while screening the furnace from the cold tubes.

In boilers with straight tubes of Babcock or Steinmuller types, two such arches sloping in opposite directions are provided (Fig. 41.13), with a space between them about 50 cm wide across the width of the furnace. A small part of the heating surface is thus in sight of the fuel bed.

With bent-tube boilers, of the Stirling type, the first pass for the gases is placed ahead of the boiler itself (Fig. 41.14); since the heating surface is not in sight of the furnace, a single small arch is sufficient. The space left between it and the opposite wall is then of the order of 100 cm.

This restriction in the gas path, especially in the former case, obviously causes a high gas velocity at this point, and an appreciable draught loss. In spite of the simplification of the gas circuit, the Ward furnace requires a higher draught than other types of furnace, and suffers a greater unburnt gas loss, of the order of 5–6%.

Like the Cook furnace, it is particularly well suited for use with an air-heater.

A height of about 4 m is allowed between the hearth and the arch.

In the same way, at least 4.3 m will be allowed between the top of the furnace and the tubes nearest the fire (for straight-tube boilers, as Fig. 41.13). This recommendation applies mainly to Ward furnaces but also to those of horseshoe (Cook) type^{37a}.

When it appeared, the Ward furnace involved a slight increase in the height required, but presented a reduction in space and length of the furnace, and a saving in the amount of refractories. At present (1982) the Ward furnace and the Cook furnace have been completely replaced by the development of the spreader-stoker furnace; their efficiency is limited by the excess air requirements, they still require an excessive quantity of refractories, their upkeep is expensive and they are not suitable for automatic operation.

(d) Spreader-stoker furnace. This is the most recent type of furnace. It has no enclosing wall as has the Cook or Ward furnace, and consists simply of the space situated between the boiler tubes and a special flat grate.

The latter may be: (a) fixed; (b) of hand-rocking type for removal of ashes (Fig. 41.15); (c) mechanical, with continuous discharge of ashes. This third type is to be recommended for capacities of 35 tonnes of steam per hour or higher³⁸. At present (1982) this can well be recommended for above 20 t/h.

It is the method of feeding the bagasse which constitutes the most original feature of the spreader-stoker furnace. It is effected as in a Ward furnace, by means of a steeply sloping chute; but at the moment this chute discharges the bagasse into the furnace, air under pressure, supplied by a pipe located just behind the chute, is blown in through a longitudinal

slot running the whole length of the bottom end of the chute, and throws the bagasse violently into the furnace. The bagasse is thus dried and burnt as it falls into the furnace, and the largest pieces complete their combustion on the grate. The proportion of bagasse burned in suspension ranges from 80 to 90% according to the size of the particles.

The quantity of unburnt solids remaining in the ash pit is thus greater than that evacuated by the chimney. It is estimated that a total unburnt loss of 2.5% is normal ($a = 0.975$).

Whatever type of grate is used, the air spaces through the grate are generally made 3 - 5% of the grate area, instead of 25 - 40% for step grates. A certain part of the combustion air is supplied by the air under pressure which serves to throw the bagasse into the furnace.

It is considered that the spreader-stoker furnace permits of reducing the normal excess air to 30% (instead of 40 - 50%) and consequently of improving the efficiency substantially.

Moreover, this type of furnace permits a combustion rate very superior to that of other types. For example, an evaporation rate of 35 - 40 kg/h/m² (7 - 8 lb./h/sq.ft.) of heating surface may be obtained compared with 25 - 30 (5 - 6) for furnaces of Cook or Ward type. Of course, the dimensions of the combustion chamber must be planned to suit the type of boiler chosen; it will be advisable to keep to combustion rates of about 220,000 kcal/m³/h (25,000 B.T.U./cu.ft./h), and not to go above 260,000 (40,000), values reckoned on G.C.V.

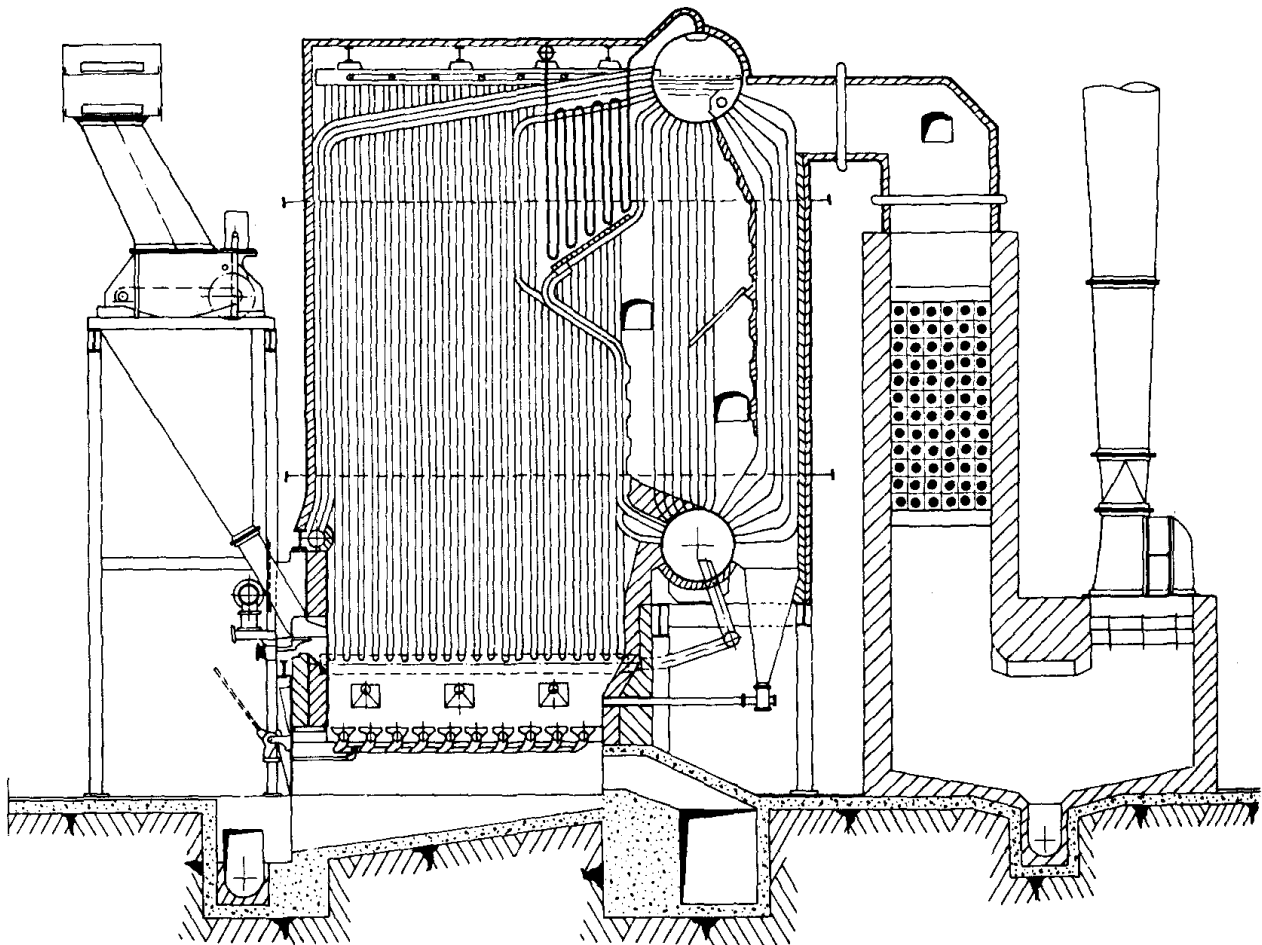


Fig. 41.15. Spreader-stoker furnace with rocking grate type BR1 (Fives Cail - Babcock).

Finally, the spreader-stoker makes ash removal easy, is easy to clean, and, having no arches or separate furnace, is economical in brickwork.

Each fuel distributor can ensure a good distribution of bagasse on a maximum width of approximately 1.2 – 1.5 m (4-5 ft.). In the series of sizes, when a change is made from a unit with n fuel distributors to one with $(n + 1)$ distributors, the cost per tonne of steam per hour is appreciably increased. It is therefore advisable to choose the largest model available for a given number of fuel distributors.

The total weight of air supplied to the furnace is approximately five times the weight of bagasse (eqn. 41.30). It is distributed in the following proportions:

Air supplied below the grate:	86 %
Air for fuel distributors:	6.5%
Air for providing turbulence above the grate:	7.5%

Combustion chamber

The furnace is the space included between the grate (or the hearth, for a hearth type of furnace), the side walls of the furnace, the arch, the front wall and the bridge wall.

The combustion chamber includes the furnace plus the free space traversed by the gases between leaving the furnace and reaching the boiler tubes. The ashpits are not included, if such are provided.

Combustion chamber volume. The volume of the combustion chamber should be proportioned to the volume of gases necessary for combustion. This volume is therefore generally fixed in relation to the quantity of heat liberated per hour by the fuel used (N_f).

Since a certain ratio exists between the heating surface of the boiler and the quantity of steam which it can produce (*cf.* p. 955), the combustion chamber volume may also be related to the heating surface of the boiler.

Deerr³⁹ recommends 30-90 dm³ of combustion chamber volume per m² (10-30 cu.ft./100 sq.ft.) of heating surface, and this, according to his figures, would correspond to 100,000 – 300,000 kcal/h/m³ (10,000 – 30,000 B.T.U./h/cu.ft.).

Tromp⁴⁰ suggests 20 - 100 dm³/m² (7 - 33 cu.ft./100 sq.ft.) of heating surface, the highest figures corresponding to the highest rate of combustion. However, as a function of the evaporation rates which he indicates, this would correspond to about 300,000 – 700,000 kcal/h/m³ (30,000 – 70,000 B.T.U./h/cu.ft.).

In Cuba, figures of 40 - 50 dm³/m² (13 – 16 cu.ft./100 sq.ft.) of heating surface are used; in Hawaii, about 75 (25 cu.ft.). These values are not applicable to furnaces of the Ward type.

In Louisiana⁴¹ values of 180,000 kcal/m³/h (20,000 B.T.U./cu.ft./h) for bagasse are taken, 220,000 for fuel oil, and a maximum, even with water-walled furnaces, of 270,000 kcal/m³/h.

For spreader-stoker furnaces of Riley type, Miller⁴² gives a maximum of 350,000 kcal/mVh (40,000 B.T.U./cu.ft./h), but recommends keeping in the region of 220,000 kcal/mVh as an optimum value, in order to improve the efficiency and minimize the formation of fly ash.

French manufacturing firms design for 170,000 – 250,000 kcal/h/m³ (19,000 – 28,000

B.T.U./h/cu.ft.) of combustion chamber volume. Relating this to the heating surface of the boiler, and expressing combustion volumes in volume relative to heating surface area, this corresponds to about $140 - 160 \text{ dm}^3/\text{m}^2$ ($46 - 52 \text{ cu.ft./100 sq.ft.}$) of boiler tube surface, increased where appropriate by the area of water walls.

The best results, to the author's knowledge, are obtained with the latter values, but the combustion chamber volume is a relatively elastic figure which may vary between fairly wide limits.

The volume V of the combustion chamber is then given by:

$$V = \frac{BN_i}{200,000} \quad \left(\frac{BN_i}{28,500} \right) \quad (41.53)$$

V = combustion chamber volume, in m^3 (cu.ft.)

B = weight of bagasse burned, in kg/h (lb./h)

N_i = N.C.V. of the bagasse, in kcal/kg (B.T.U./lb.).

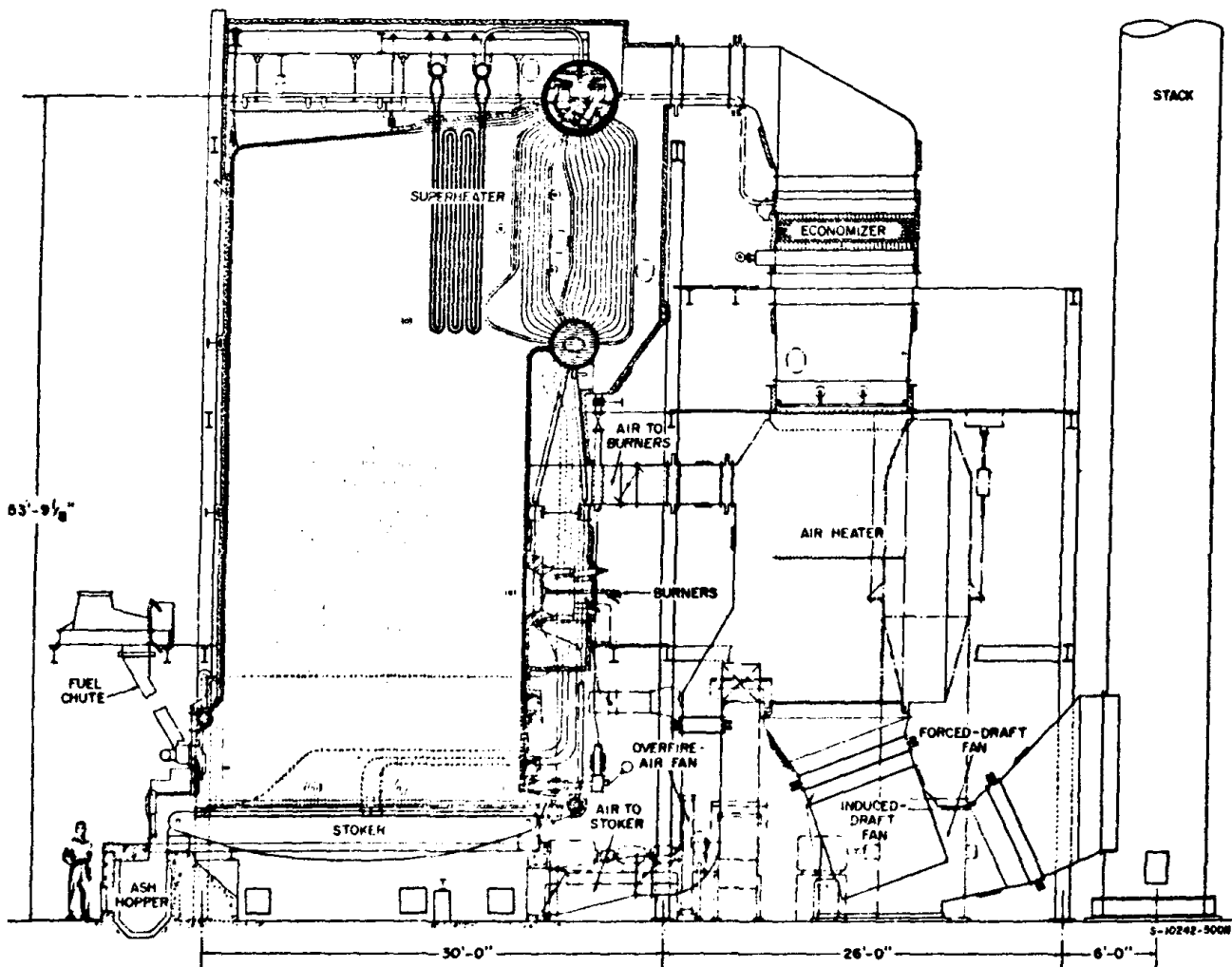


Fig. 41.16. Spreader-stoker furnace with travelling grate (Babcock & Wilcox).

Dimensions of the furnace

All the furnace dimensions are fixed by the necessity to observe the following conditions:

(a) **Length of flame.** The length of passage for the burning gases, between the grate and boiler tubes, should be at least 5 m, and preferably 7-8 m. It should not exceed 10 m.

Below 7 m and particularly below 5 m, the gases would not be completely burnt on reaching the cold-water tubes, and the sudden cooling caused by their passage between the tubes would to a great extent arrest the combustion, thus increasing the proportion of CO, and decreasing the efficiency. Furthermore, below 5 m, the ash entrained with the gases will not be completely burnt, and will thus tend to adhere to the tubes, thus becoming harmful and dangerous (see Fig. 41.17).

On the other hand, if the length of path for the gases is unnecessarily increased, there will be increased losses by radiation and by air leakage, as well as an increase in the space required and in the cost of the installation.

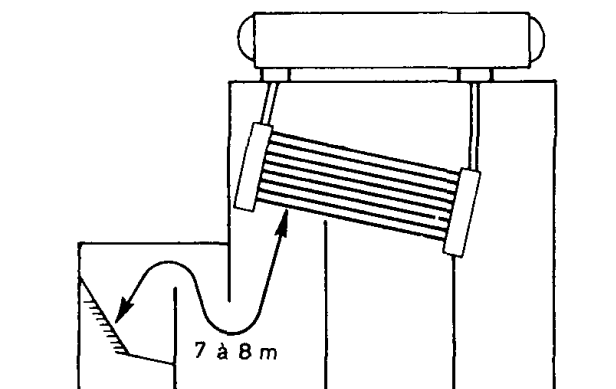


Fig. 41.17. Length of flame.

(b) **Width of the boiler.** The various types of water-tube boilers generally have a given heating surface per unit width of furnace. For example:

Cail – Steinhilber boilers with headers: 110 m²/m (360 sq.ft./ft.) width

Fives – Stirling boilers with 3 drums: 135 m²/m (443 sq.ft./ft.) width

With recent installations, where the capacity is expressed in tonnes of steam per hour rather than in heating surface area, we have:

Fives Cail – Babcock boilers with BC1 type furnace: 5 t/h/m (1.5 t/h/ft.) of width

Fives Cail – Babcock boilers with spreader-stokers,

type BR1: 6 t/h/m (1.8 t/h/ft.) of width

Fives Cail – Babcock boilers with spreader-stokers,

type BR2: 7 t/h/m (2.3 t/h/ft.) of width

BR3: 9 t/h/m (2.7 t/h/ft.) of width

BR22D: 12.5 t/h/m (3.7 t/h/ft.) of width

To avoid the drawbacks of a complicated shape, the combustion chamber should have the same interior width as the boiler, and the total width of the furnace or furnaces should be at least equal to the latter.

(c) Volume of combustion chamber. Finally, the total volume of the combustion chamber should observe the conditions given on p. 949.

It will be seen that the length, width and volume of the combustion chamber must conform to certain conditions, leaving little liberty to the designer of the furnace. The least imperative condition and the most elastic of the three is, however, that of the volume, which can, without great inconvenience, differ appreciably from the values given.

Secondary air. In all modern furnaces, there is provided, in addition to the normal air or "primary air", entering directly through the grate or by the tuyeres of the hearth furnace, a complementary air supply for "secondary air", behind the bridge wall, and consequently after the furnace proper. This air is introduced by a small duct built into the bridge wall (Figs. 41.11 and 41.12).

The introduction of this supplementary air for combustion has the object of ensuring complete combustion by changing to CO_2 the CO which may remain after combustion in the furnace.

Secondary air is generally made 5 – 15% of the total air supplied, averaging 10%. There is no advantage in exceeding this amount; if combustion is good in the furnace, *i.e.* if the combustion temperature is high, it forms very little CO , and there would be risk of causing a useless increase in excess air.

Suspended arch furnaces

The roofs of older boiler furnaces were made as true arches, formed from curved bricks. Then flat suspended arches were adopted, of the Fama type, which cost somewhat more, but lasted much longer and were much more economical in upkeep. It is considered in Louisiana⁴³ that above 3.5 m (12 ft.) in height, it is similarly advantageous to use walls of suspended bricks, the weight of which does not increase the pressure on the bricks at the base of the furnace.

Today, furnaces are designed practically without horizontal portions and thus without arches, and these are the most economical.

THE GRATE

Grate area

In step grates, combustion takes place on the steps. The dead plate which precedes them serves to remove the excess moisture from the bagasse before it comes to the grate. The ash grate which follows the step grate serves to complete the combustion and to utilise the heat transmitted by radiation from the incandescent bagasse before the ashes are dropped into the ashpit. The grate area proper is that obtained by multiplying the width of the step grate by its length measured along the slope, between the first and the last step.

In horseshoe or Ward furnaces, the grate area is that of the hearth plate inside the horseshoe or the walls surrounding the furnace.

In spreader-stoker furnaces, the grate area is obviously simply the flat surface which receives the bagasse, of the rocking grate, or the upper portion of the travelling grate.

Weight of bagasse burnt per unit grate area per hour

The rate of working of a furnace grate is expressed by the quantity of bagasse burnt on this grate per unit area of the grate. We shall designate this rate by:

$$\beta = \frac{\text{weight of bagasse burnt per hour on the grate}}{\text{area of the grate}} \quad (41.54)$$

This "combustion rate" is interesting, since, to obtain sufficient combustion, it is necessary to proportion the grate area to the quantity of bagasse to be burnt per hour. If the draught were constant, the optimum value of the weight of bagasse burnt per unit area of the grate per hour would be easily fixed. However, this value increases rapidly with the rate of operation adopted, whereas the efficiency of the combustion varies little with the rate. In other words, the graph of combustion efficiency as a function of rate of working is very flat, and we may obtain a very good combustion:

- (a) At low draught and low combustion rate
- (b) At high draught and high combustion rate.

For this reason, great differences are found in the figures given by different authors on optimal combustion rates.

Habif⁴⁴ quotes a rate corresponding to 425 kg/m²/h (87 lb./sq.ft./h), while Deerr³⁹ gives a figure of 500 (100), and Tromp⁴⁵ gives 1,200 - 1,460 (250 - 300); Shillington⁴⁶ gives a figure of 1,460 (300) in the same units.

We shall accept as optimal rates the figures in Table 41.23.

TABLE 41.23

OPTIMAL COMBUSTION RATES PER UNIT ORATE AREA FOR BAGASSE

	<i>kg/m²/h</i>	<i>lb. /sq.ft./h</i>
Low	500 – 700	100 – 140
Moderate	700 – 800	140 – 160
High	800 – 1,000	160 – 200

Contrary to what might be imagined, these combustion rates scarcely vary with the type of furnace. At continuous high ratings, spreader-stoker furnaces sometimes reach 1,100 – 1,200 kg/m²/h (225 – 245 lb./sq.ft./h).

Rocking grate and travelling grate. The rocking grate is becoming widely used. It has been adopted by Babcock and Wilcox, Thompson, Foster Wheeler, Takuma and Yoshimine. Fives Cail – Babcock suggest it for capacities up to 160 t/h of steam. It is virtually static and for

this reason gives less trouble, its upkeep is practically nil whereas the travelling grate requires checking and attention at each weekly shut-down; it allows less bagasse loss. The total cost of a boiler with a rocking grate is about 7% lower than that of a boiler with a travelling grate.

The travelling grate is required only in the case of fuel of high ash content, say above 8%. This is, for example, the case with bagasse pellets in Hawaii. It is then advisable to control its speed so as to obtain an ash layer as low as possible, but sufficient to protect the grate from direct radiation from the furnace.

The travelling grate generally does not accept combustion air at a temperature above 200°C.

Evaporation rate

The "evaporation rate" or "rating" of a boiler is the weight of dry steam which it produces per unit heating surface per hour.

We shall designate it by the Greek letter T .

Standard evaporation rate. We have already seen (Table 41.21) that the quantity of heat units required for the production of unit weight of steam varies according to the temperature of the feed water for the boilers, and according to the pressure and also the superheat of the steam.

To render evaporation rates comparable with one another, the British and American practice is to relate the quantity of steam produced to that which would be produced if the feed water were at 100°C (212°F) and if evaporation were carried out at atmospheric pressure (from and at 100°C (212°F)). In other words, the quantity of heat transferred to one kg (lb.) of the steam produced is taken, divided by 539 (970) (latent heat at 100°C (212°F)) and the weight of steam generated per unit area of heating surface is then multiplied by the coefficient so obtained.

The standard conditions in French practice, on the other hand, are: feed water at 0°C, steam at 100°C. In other words, French engineers replace the divisor by the figure 639 (1,150). This value has the advantage of approaching more closely to the order of magnitude of the quantity of heat supplied per unit weight of steam in modern industrial practice (*cf.* Table 41.21).

Example. Consider a boiler receiving feed water at 90°C and producing 36 kg steam/m² heating surface/h at 30 kg/cm² (427 p.s.i.), superheated to 400°C.

Each kg of steam has thus received 681 kcal (*cf.* Table 41.21) The standard evaporation rate will then be:

$$(a) \text{ English rating: } T\dot{O} = 36 \times \frac{681}{539} = 45.5 \text{ kg/m}^2/\text{h} \quad (8.85 \text{ lb./sq.ft./h})$$

$$(b) \text{ French rating: } T\dot{O} = 36 \times \frac{681}{639} = 38.4 \text{ kg/m}^2/\text{h} \quad (7.46 \text{ lb./sq.ft./h})$$

We may comment that the English rating, to be rendered comparable with the French rating, should be multiplied by:

$$\frac{539}{639} \quad 0.8435 \quad (41.55)$$

and figures according to the French rating, to convert to English rating, should be multiplied by the reciprocal of this or 1.186.

If the steam produced is wet, it will obviously be necessary to take into account the lower quantity of heat which has been supplied to each kg of steam.

In the following discussion, the evaporation rates indicated will be either standard English ratings (from and at 100°C (212°F), TO) or actual practical ratings obtained (τ).

Average evaporation rates. Older semi-tubular boilers gave an evaporation rate of about 12 kg/m²/h (3.0 lb./sq.ft./h).

All water-tube boilers are capable of a normal rate of 18 (4.4), but those with two circulations cannot be forced to the same ratings as those with three circulations. We may reckon for a continuous operation when pushed, the figures of Table 41.24.

TABLE 41.24

EVAPORATION RATES kg/m ² /h (lb./sq.ft./h)			
<i>Boilers</i>	<i>Accessory heating surfaces</i>	<i>Furnace</i>	<i>Evaporation rate τ_0</i>
Straight tubes 2 circulations	Economiser or air-heater	Step-grate	22 (5.3)
Straight tubes 3 circulations	Economiser or air-heater	Horseshoe or Ward	32 (7.8)
Vertical bent tubes	Economiser and/or air-heater	Spreader-stoker	36 (8.7)

With an economiser and air-heater of generous dimensions, it is possible to achieve a rating of 40 kg/m²/h (10 lb./sq.ft./h).

In all these discussions, when speaking of the heating surface of boilers, we take into account the heating surface of the boiler tubes and of the water walls, but not those of the superheater, economiser and air-heater, which contribute to the total evaporation, but to a smaller extent.

When the evaporation rate is raised by modifying the draught, it may be estimated approximately that the evaporation rates will vary in proportion to the square root of the draught expressed in cm (in.) of water.

Ratio of heating surface to grate area

We shall designate this ratio by

$$\sigma = \frac{\text{heating surface } S \text{ of the boiler}}{\text{area } s \text{ of the grate}} \quad (41.56)$$

Let:

B = weight of bagasse burnt per hour

Q = weight of steam generated per hour

We shall have:

$$BM_v = 970 Q \quad (41.57)$$

M_v = fraction of the C.V. of the bagasse which is recovered in the steam, given by eqn. (41.30), in kcal/kg (B.T.U./lb.).

But:

$$B = Os$$

and:

$$Q = \tau'_0 S$$

Whence:

$$\sigma = \frac{S}{s} = \frac{M_v \beta}{639 \tau'_0} \left(\frac{M_v \beta}{970 \tau'_0} \right) \quad (41.58)$$

If we take mean values: $\beta = 750$ kg/m²/h (150 lb./sq.ft./h), $M_v = 1,475$ kcal/kg (2,650 B.T.U./lb.), $T\dot{O} = 30$ kg/m²/h (7–7.5 lb./sq.ft./h), we see that a is of the order of 50–75. For a spreader-stoker furnace, it is in the neighbourhood of 50.

However, there is no need to attribute to a an exaggerated importance. The important and interesting values are those of $T\dot{O}$ and β . To produce the quantity of steam which the factory requires, a certain boiler heating surface will be necessary, corresponding to a suitable value of the evaporation rate $T\dot{O}$. To obtain this quantity of steam, with the normal efficiency of the boilers, it is necessary to burn a certain weight of bagasse, and consequently to install the desired grate area, after having chosen an appropriate combustion-rate figure β . It follows that we shall have a certain boiler heating surface S and a certain grate area s . However, if the heating surface S is altered for any reason, without altering the weight of bagasse burnt, there will be no need to alter the grate area s for this reason, in the same ratio, with the object of maintaining a certain value of a . Hence, if the area S has the correct value, it is necessary on the contrary to retain it.

In other words there is no definite interdependence or proportionality between s and S : these two areas should each fulfil certain independent conditions.

PERFORMANCE OF DIFFERENT FURNACES

We shall now discuss particular points of the principal types of furnace.

Horseshoe furnace

Bagasse burnt per unit grate area. In taking as area of the furnace the plane surface at the bottom, comprising the interior of the horseshoe, this furnace permits a combustion rate appreciably higher than the step grate. Tromp⁴⁵ reports 2,200–3,200 kg bagass/m²/h (450–650 lb./sq.ft./h). French manufacturing firms prefer to base their designs on 1,300–1,800 (270–360), or an average of 1,500 (300), and consider rates of 2,000–2,200 (400–450) as maxima. The combustion rate of the furnace is a function of its perimeter and not of its area. This is one reason for the large variation in the weight of bagasse burnt per unit area^{37a}.

As will be seen from the two groups of limits indicated, this rating may be modified, by altering the draught or the arrangement of tuyeres, to a greater extent than with an ordinary grate. It is thus a more flexible design of furnace.

Efficiency. This is also a furnace in which the highest temperatures and efficiencies have been recorded.

It offers moreover a further advantage, which is by no means negligible, in that it gives ashes which are very rich in potash, and nicely pulverised.

Height. Since it has no ashpit, the horseshoe furnace requires a greater height. A total height of 4.5–5 m (15–17 ft.), between hearth and arch, is not excessive. This height should not be less than 3 m (10 ft.).

The furnace proper, that is the portion enclosed in the horseshoe, should have a height of about 2.5 m (8 ft.), or 2 m (6.5 ft.) as a minimum.

Draught. Ample draught should be provided so that, when high ratings are required, the air jets from the tuyeres may penetrate deeply into the burning pile of bagasse. Otherwise, some bagasse would remain unburnt, or at least combustion would be incomplete, with formation of CO, since the air has no other means of reaching the fuel, in this type of furnace.

Air velocity in the tuyeres. The total cross-section of the tuyeres will be calculated so that the mean velocity of the air will lie between 15 and 20 m/s (50–65 ft./s). A value of less than 15 (50) should not be used, otherwise the tuyeres will become fouled and blocked. The tuyeres should be mainly located at the bottom of the furnace: 85–90% of them in the first 30–38 cm (12–15 in.) from the hearth, with the remainder in one row 0.6–1 m (2–3 ft.) higher^{46a}.

Ward furnace

Bagasse burnt per unit grate area. A value of 1,000–1,200 kg/m²/h (200–250 lb./sq.ft./h) is taken as a normal figure. In exceptional cases 1,500 kg (300 lb.) may be attained but, at this rating, a large amount of unburnt bagasse particles is carried through to the chimney.

Spreader-stoker furnace

Bagasse burnt per unit grate area. A figure of 2,500–3,000 kg/m²/h (500–600 lb./sq.ft./h)

h) may be attained, but most manufacturers design for 800 (160); it is recommended that a figure of 1,200 (175) should not be exceeded.

HEATING SURFACE

The heating surface comprises that of the boiler tubes and the tubes of water walls, when these are provided.

It should be noted that these two sections of the heating surface are in no way comparable as regards the proportion of heat absorbed; if we express this heat absorption as weight of steam produced, the boiler tubes would have a capacity of about 25 kg/m²/h (6 lb./sq.ft./h); whereas the tubes of the water walls, placed vertically and in single layer, exposed to the full radiation of the furnace, would attain a figure of 65 (16) (relating their heat absorption to their total commercial surface area). If there are no water-wall tubes, the full heat developed by the furnace would arrive at the superheater and the boiler tubes, and the evaporation of the latter would obviously be greater.

The heating surface required in a sugar factory per t.c.h. depends on three factors:

(a) The steam consumption of the factory per t.c. This may vary from a maximum of 750 kg (1,680 lb.) in a poorly equipped factory which is manufacturing white sugar, to a minimum of 360 kg (800 lb.) in a modern factory making raw sugar, and using pressure evaporation or thermo-compressors. Generally, steam consumption varies between 450 and 550 kg/t.c. (1,000 – 1,250 lb./t.c.).

(b) The type of boiler used, the normal evaporation rate of which may be high or low.

(c) The greater or lesser intensity of boiler operation.

Since it is advisable to allow some reserve capacity, it will be preferable to provide a heating surface corresponding to a normal rating.

If:

A = crushing rate of the factory, in t.c.h.

Q = steam requirements of the factory, in kg/t.c. (lb./t.c.)

T' = evaporation rate of the boilers, in kg/mVh (lb./sq.ft./h)

S = heating surface of the boilers, in m² (sq.ft.).

we have:

$$AQ = \tau' S \quad (41.59)$$

hence:

$$S_s = \frac{S}{A} = \frac{Q}{\tau'} \quad (41.60)$$

S_s = specific heating surface, in m²/t.c.h. (sq.ft./t.c.h.)

If we take Q - 450 kg/t.c. and $\tau' = 30$ kg/mVh we will have:

$$S_s = 15 \text{ mVt.c.h.} \quad (165 \text{ sq.ft./t.c.h.})$$

It is advisable to keep in reserve some older boilers, to avoid possible stoppage by some mishap with one of those in service. Also, many factories work at a value of S_s of 20 $m^2/t.c.h.$ (215 $sq.ft./t.c.h.$), to avoid lighting up the reserve boiler.

Radiation and convection

With spreader-stoker boilers, when the tubes of the water walls occupy about 40% of the area of the lateral walls (*e.g.* vertical tubes of 3 in. diam. at 7.5 in. pitch, in the case of Fives Cail - Babcock boilers), it is considered that 27 - 28% of the total heat developed in the furnace is absorbed by direct radiation.

Distribution of heat absorption

In a modern boiler, such as is illustrated in Fig. 41.15, the usable heat is distributed as follows:

	<i>Relative tube surface (%)</i>	<i>Proportion of heat absorbed (%)</i>
Water walls of furnace:	12	34
Superheater:	6	15
Boiler tubes:	37	38
Economiser:	45	13
	—	—
	100	100

Distribution of heating surface

In modern boilers, the total heating surface has no great significance, on account of the difference in rates of heat transfer existing between the different components of the boiler unit. For this reason the figure quoted is the evaporation per hour of which the unit is capable, rather than the heating surface.

By way of example, the following figures give the heating surfaces for a Fives Cail - Babcock boiler type BR2 43/68 of 60 t/h steam capacity, tested at 49 atm, producing steam at 42 atm at the superheater outlet, temperature of superheat 420°C (788°F), feed water temperature 105°C (221°F), flue gas temperature 200°C (392°F), efficiency 83%.

Water-wall tubes:	359 m^2 = 3,864 $sq.ft.$ = 12% = 6 m^2/t steam/h
Plain tube assembly:	1,340 m^2 = 14,424 $sq.ft.$ = 44% = 22.3 mVt steam/h
Superheater tubes:	140 m^2 = 1,507 $sq.ft.$ = 5% = 2.3 mVt steam/h
Economiser:	1,189 m^2 = 12,798 $sq.ft.$ = 39% = 20 mVt steam/h
	—
	3,028 m^2 = 32,593 $sq.ft.$ = 100% = 50 mVt steam/h

The combustion chamber is 8.21 m in width, 4.51 m in depth and 8.50 m in mean height, giving 277 m^3 which is 4.6 mVt of steam. Grate dimensions are 8.21 m in width and 4.26 m in depth = 35 m^2 grate area.

	<i>d (mm)</i>	<i>thickness (mm)</i>	<i>pitch (mm)</i>
Water-wall tubes:	76	4	185
Front tubes	76	4	
Middle tubes:	63.5	3.2	
Rear tubes:	63.5	3.2	
Superheater:	38	3.2	

The lateral water-wall tubes are welded into upper and lower headers. The front water-wall tubes are welded into the lower header and expanded into the upper drum. The tubes on the bridge are expanded into the upper and lower drums, and the tubes connecting the two drums are expanded into both drums.

The superheater is located at the delivery from the combustion chamber immediately after the direct radiation tubes. The boiler has six Riley-France spreader-stokers.

The grate area is calculated on the basis of 1170 kg/m²/h; this should correspond to 2,700,000 kcal/m²/h reckoned on G.C.V. of 2310 kcal/kg (standard proposed by Detroit Stoker Co.).

SUPERHEAT AND SUPERHEATERS

Use of superheated steam

Use of superheated steam is indispensable where power supply to the factory is by a steam turbine. Table 41.25 gives steam conditions often used.

TABLE 41.25

CHARACTERISTICS OF STEAM FOR USE IN TURBINES

18 kg/cm ² gauge:	325 – 350°C	250 p.s.i.g.:	620 – 660°F
25 kg/cm ² gauge:	350 – 375°C	350 p.s.i.g.:	660 – 700°F
30 kg/cm ² gauge:	375 – 400°C	425 p.s.i.g.:	700 – 750°F
45 kg/cm ² gauge:	400 – 440°C	640 p.s.i.g.:	750 – 820°F

Limit of superheat. With carbon steels, temperatures above 425°C (800°F) in the superheater tubes cannot be used. With special steels, temperatures of 500°C (930°F) are reached and exceeded, but the latter limit is of little interest in the cane sugar factory.

Superheaters

Superheaters are heat exchangers placed in the path of the hot gases. They are generally located in any suitable free space in the neighbourhood of the boiler tubes (Figs. 41.14 and 41.15).

They receive the saturated or slightly wet steam coming from the boiler drum and deliver it in a superheated state to the general steam main of the factory.

They are generally formed from tubes of small diameter, all of the same shape with several bends, interposed between two headers.

In France, the heating surface of superheaters is calculated from the external diameter of the tubes. This is important as, due to the small diameter and the thickness of the tubes, there is a substantial difference between the internal and external surface areas.

Calculations for superheaters

We have two principal equations:

$$M = \alpha PC (T_1 - T_2) = p (1 - x)r + pc (T - t) \quad (41.61)$$

M = quantity of heat transmitted by the superheater, in kcal/h (B.T.U./h)

α = coefficient ≤ 1 , generally 0.90

P = weight of gas passing over the superheater, in kg/h (lb./h)

C = specific heat of these gases (Fig. 41.46)

T_1 = temperature of gases at entry to the superheater, in °C (°F)

T_2 = temperature of gases leaving the superheater, in °C (°F)

p = weight of steam to be superheated, in kg./h (lb./h)

x = dryness fraction of the saturated steam (0.80 — 0.98 in general)

r = latent heat of vaporisation, at the boiler pressure, in kcal/kg (B.T.U./lb.) (cf. Table 41.1)

c = mean specific heat of the superheated steam, between t and T° (cf. Table 41.2)

t = temperature of the saturated steam, at the boiler pressure

T = temperature of superheat desired or obtained.

$$M = kS \left(\frac{T_1 + T_2}{2} - \frac{T + t}{2} \right) \quad (41.62)$$

k = coefficient of heat transfer, in kcal/m²/h/°C (B.T.U./sq.ft./h/°F), which varies from 50 to 65 (10 — 13) according to the temperature and velocity of the hot gases. Generally, a value of 55 - 60 (1 - 12) may be used.

S = heating surface of the superheater, in m² (sq.ft.)

Eliminating M between eqns. (41.61) and (41.62), we have:

$$S = \frac{p (1 - x)r + pc (T - t)}{k \left(\frac{T_1 + T_2}{2} - \frac{T + t}{2} \right)} \quad (41.63)$$

Dryness fraction. It may be remarked that the influence of droplets of water entrained in the steam, $p (1 - x)r$ often constitutes the greater part of the heat to be supplied to the steam.

The dryness fraction of the steam for superheating is generally 0.96 — 0.98. In the cane sugar factory, figures are sometimes quoted of 18% moisture in steam ($x = 0.82$). It will be advisable therefore to measure the dryness fraction or to make a generous allowance for it.

Location of superheaters. In boilers with straight inclined tubes, the superheaters are plac-

ed in the gas circuit, after the first pass across the tubes and before the second, above the tubes. The gas temperature at this point is then of the order of 600 - 700°C (1,100 - 1,300°F). In boilers with vertical bent tubes and two drums, it is often placed before the tubes. It thus receives the hot gases, of temperature 800 - 1,000°C (1,500 - 1,800°F); the gases then carry the full heat content apart from that radiated to the water walls of the furnace,

Design of a superheater. We shall apply the preceding formulae to a concrete example.

Rated capacity of boiler:	50 t/h	
Working pressure:	30 kg/cm ² (427 p.s.i.)	($t = 235^{\circ}\text{C}$ (454°F))
Moisture in bagasse:	48%	($w = 0.48$)
Sugar in bagasse:	2%	($s = 0.02$)
Excess air:	40%	($m = 1.4$)
Feed-water temperature:	90°C (194°F)	
Superheat temperature required:	400°C (752°F)	(7)
Dryness fraction of saturated steam:	98%	($x = 0.98$)
Gas temperature at the point where superheater will be placed:	900°C (1,650°F)	(T_1)
Gas temperature leaving the boiler:	225°C (437°F)	(T_2)
Coefficients of heat loss:		
unburnt:	$a = 0.975$	
radiation:	$0 = 0.975$	
incomplete combustion:	$\eta = 0.97$	

It is required to calculate the superheater to be placed in this boiler.

Total heat to be supplied per kg of steam (Table 41.3)

Total heat in steam at 30 kg/cm ² and 400°C:	771 kcal/kg	1,388 B.T.U./lb.
Total heat in feed water:	90	162
Heat to be supplied per kg of water:	681	1,226

Sensible heat lost in flue gases (eqn. 41.48)

$$q = l(w - w) (1.4m - 0.13) + 0.5]t =$$

$$[(0.52 \times 1.83) + 0.5] 225 = 327 \text{ kcal/kg (588 B.T.U./lb.)}$$

Heat available per kg of bagasse (eqn. 41.50)

$$M_v = (4,250 - 1,200s - 4,850w - q)\alpha\beta\eta =$$

$$(4,250 - 24 - 2,328 - 327) 0.975 \times 0.975 \times 0.97 = 1,450 \text{ kcal/kg (2,610 B.T.U./lb.)}$$

Weight of steam supplied per kg of bagasse burnt

$$\frac{1,450}{681} = 2.13 \text{ kg}$$

Total weight of bagasse to be burnt

$$B = \frac{50,000}{2.13} = 23,475 \text{ kg/h (52,580 lb./h)}$$

Weight of gas. We have (eqn. 41.30):

$$P_g = 5.76 (1 - w)m + 1 = 5.193 \text{ kg/kg}$$

Or total:

$$P = 5.193 \times 23,475 = 121,900 \text{ kg/h (273,050 lb./h)}$$

Temperature of gases leaving the superheater. From eqn. (41.61), we have:

$$T_2 = T_1 - \frac{n}{\alpha PC} [(1 - x)r + c (T - t)] \tag{41.64}$$

$$T_2 = 900 - \frac{50,000}{0.90 \times 121,900 \times 0.32} [(0.02 \times 428) + 0.567 (400 - 235)]$$

$$T_2 = 900 - 145 = 755^\circ\text{C (1,385}^\circ\text{F)}$$

We have taken C from formula (41.46) and c from Table 41.13.

We may also, for c, apply the formula:

$$c = 0.468 + 0.000311t \tag{41.65}$$

c = specific heat of water vapour at temperature t.

Calculation of S. The heating surface of the superheater installed is thus (eqn. 41.63):

$$S = \frac{50,000 (0.02 \times 428 + 0.567 (400 - 235))}{55 \left(\frac{900 + 755}{2} - \frac{400 + 235}{2} \right)} = \dots \text{ m}^2 \text{ (2,205 sq.ft.)}$$

(calculated on external dia. of tubes).

Area of steam passage. The superheater should offer to the steam sufficient cross-sectional area of passage to give a velocity of steam, calculated at the mean temperature $(T + t)/2$, of about 20 m/s (10–40 m/s) (30–130 ft./s).

In order to obtain this velocity, which it is desirable to realise, it is often necessary to subdivide the superheater into a certain number of sections traversed by the steam in series.

Value of T_1 . If the temperature of the gases at entry to the superheater is not known, it may be measured, or an approximate value may be obtained by a calculation analogous to that for juice heaters (where the gases play the role of the heating steam in the heater, and the water in the boiler that of the juice), based on the fraction of the heating surface of the boiler already swept by the gases before reaching the position where the superheater is to be installed.

The value of T_1 should lie between 500 and 700°C (950–1300°F) for straight-tube boilers, or between 800 and 1,000°C (1,500–1,800°F) for boilers with vertical bent tubes and water walls; *i.e.* the position chosen for the superheater should be one where the temperature of the combustion gases lies between those limits.

Empirical rule. If higher temperatures of superheat are adopted for higher steam pressures, as is suggested by the values indicated in Table 41.25 and which is a logical practice, it is satisfactory, for straight-tube boilers, to dispense with the calculations for the superheater and to apply the following empirical rule: the heating surface of the superheater should be one-third of the boiler heating surface:

$$s = \frac{S}{3} \quad (41.66)$$

s = heating surface of the superheater

S = heating surface of the boiler.

Superheat temperature furnished by a superheater

Problem. A superheater is installed and it is desired to know what temperature it is capable of giving to the steam in certain conditions of operation.

We shall retain the nomenclature used previously. Here S is given, and the unknowns are T_2 and T .

We eliminate M between eqns. (41.61) and (41.62), replacing T_2 by its value, eqn. (41.64), and solve for T . We have:

$$T = \frac{\frac{kS}{2} (2T_1 - t) + p [ct - r(1 - x)] \left(\frac{Sk}{2\alpha PC} + 1 \right)}{\frac{kS}{2} + pc \left(\frac{kS}{2\alpha PC} + 1 \right)} \quad (41.67)$$

which gives the temperature sought.

De-superheater. In order to maintain a uniform temperature of superheat, which does not vary abruptly according to momentary overloads in the factory, or due to unexpected stops, the steam leaving the superheater is passed into a de-superheater surface traversing the lower boiler drum, which evens out the temperature due to the fact that the temperature of the water in the drum is practically constant. This procedure obviously does not alter the heat balance of the unit.

This operation of de-superheating is often practised, not only at the superheater outlet, but between the primary and secondary superheaters, these two sections being distributed, in series and in equal sections, across the width of the boiler. This arrangement avoids an excessive drop in superheat.

Maintenance of superheaters

On account of their complicated form, it is not possible to clean the tubes of the superheater.

Now, the droplets of water entrained with the wet steam are completely evaporated in the superheater. It is therefore in the superheater that any dissolved materials will be found which may have been introduced into the boiler with the feed water.

It is necessary therefore to take great care that no water is allowed to enter a boiler fitted with a superheater, which may contain sugar or other material in solution.

If the superheater becomes fouled with a deposit of carbon or a sort of tarry deposit of burnt sugar due to this cause, the cross-section of the tubes will be reduced, with consequent restriction of steam flow, while at the same time its heat transfer coefficient will be reduced.

There is then no alternative but to replace the tubes, and to take all precautions to avoid a recurrence of the incident.

ECONOMISERS

In the sugar factory, the boiler feed water is generally at a temperature of about 90°C (194°F). Now, the saturation temperature at which the water should be fed into the boiler to be transformed into steam varies, according to the pressure used, between 164°C at 6 kg/cm² (327°F/85 p.s.i.) and 236°C at 30 kg/cm² (457°F/440 p.s.i.). There is thus a large margin of temperature to be made up in the boiler. This margin means that a substantial proportion of the total heat has to be supplied to the water before evaporation proper commences. Assuming the feed water is at 90°C (194°F), Table 41.26 gives for three typical cases, the fraction of the total heat represented by this sensible heat.

Now the combustion gases leave the boiler at a temperature which is still relatively high, and generally above the saturation temperature. This sensible heat content of the gases would be lost in the stack. Hence the idea of utilising their sensible heat content to raise the temperature of the feed water, and so decrease the quantity of heat to be supplied to the steam in the boiler.

This is the principle of the economiser. It is a heat exchanger placed in the path of the flue gases leaving the boiler, and through which the feed water is circulated between the feed pump and the boiler. It generally takes the form of tubes, in most cases with fins, through which the water circulates in series (Fig. 41.18). They are arranged in groups, the water passing from one tube to the following by means of a 180° bend.

TABLE 41.26

FRACTION OF THE TOTAL HEAT REPRESENTED BY THE SENSIBLE HEAT TO BE SUPPLIED TO THE WATER
($t_o = 90^\circ\text{C}$ (194°F))

Steam press.		Superheat		Saturation temp.		Total heat		Sensible heat		Sensible heat % total heat
(kg/cm^2)	(p.s.i.g.)	($^\circ\text{C}$)	($^\circ\text{F}$)	($^\circ\text{C}$)	($^\circ\text{F}$)	(kcal/kg)	(B.T.U./lb.)	(kcal/kg)	(B.T.U./lb.)	
6	85	Saturated		164	327	569	1,026	76	136	13
10	142	Saturated		183	362	574	1,033	96	172	17
25	356	375	707	225	437	670	1,206	141	254	21
30	427	400	752	235	455	681	1,226	154	277	23
45	640	450	842	256	493	704	1,299	176	317	25

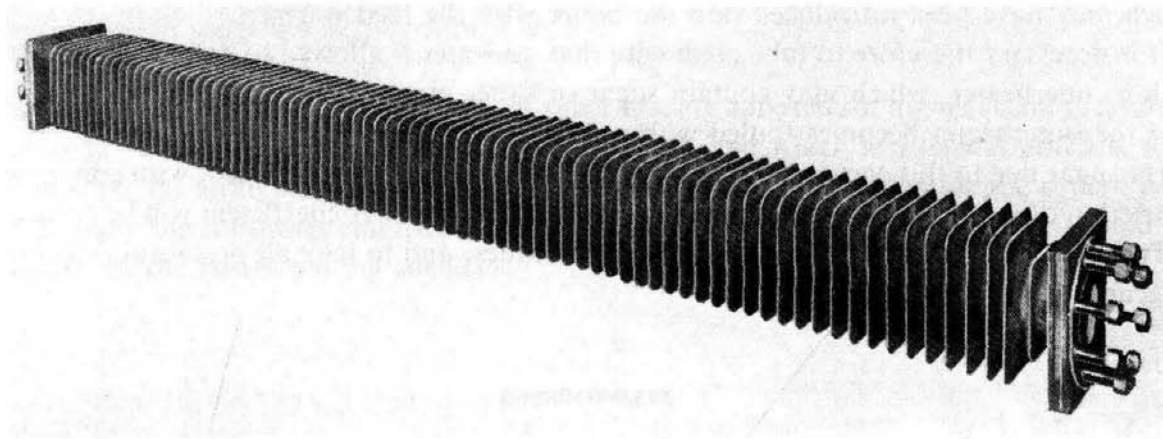


Fig. 41.18. Finned tube for economiser.

Two principal arrangements are adopted for economisers:

(a) **General economiser.** When a single economiser is installed for the whole boiler station, it is placed in the main flue (Fig. 41.19). A by-pass is sometimes provided for the gases so that they may pass through the economiser in normal operation or may go direct to the stack, to permit of cleaning the economiser.

(b) **Individual economisers.** It is now generally preferred to provide each boiler with its separate economiser. It is then placed in the last pass of the combustion gases leaving the boiler (Figs. 41.15, 41.16 and 41.20).

Corrosion. To avoid corrosion⁴⁷, the temperature of the metal should not fall below 125°C (257°F). If, instead of bagasse, a fuel containing sulphur is used, this limit should be increased (for example, to 143°C for a fuel containing 3% sulphur).

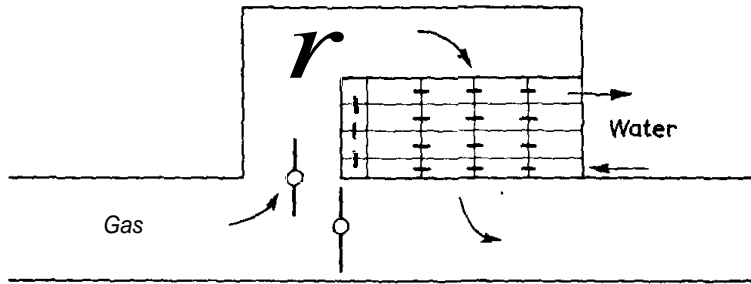


Fig. 41.19. Group economiser installed in main flue.

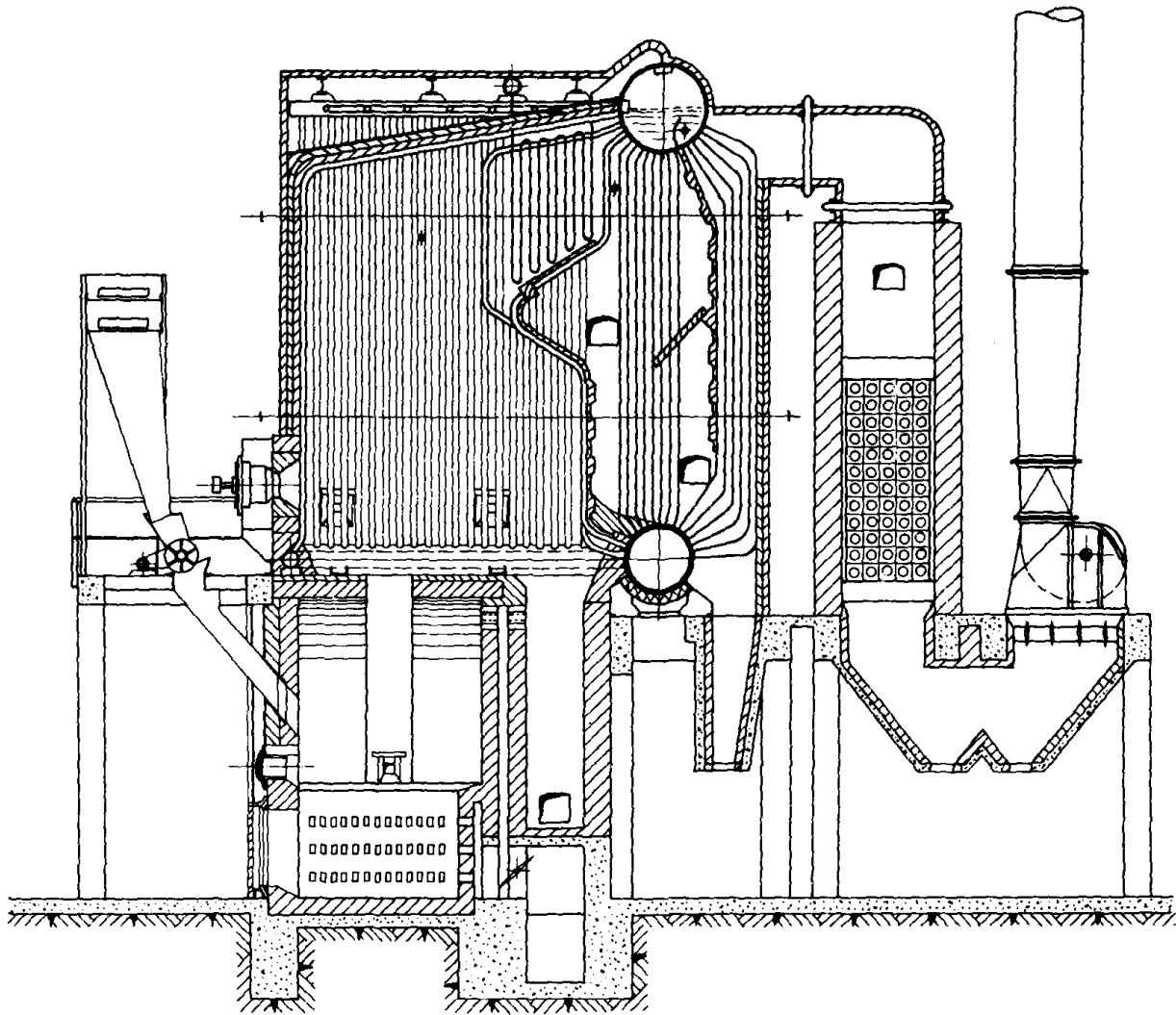


Fig. 41.20. Babcock Atlantique boiler type BC1 with Ward furnace and economiser.

Calculations for economiser

We have two main equations:

$$M = \alpha PC (T_0 - T) = pc(t - t_0) \quad (41.68)$$

$$T - t_0 = (T_0 - t)e^{-k(1-r)S/\alpha PC} \quad (41.69)$$

M = quantity of heat transmitted by the economiser, in kcal/h (B.T.U./h)

a = coefficient ≤ 1 , generally between 0.92 and 0.96; frequently 0.95

P = weight of gas passing through the economiser, in kg/h (lb./h)

C = specific heat of the gases (eqn. 41.46)

T_0 = temperature of gases entering the economiser, in °C (°F)

T = temperature of gases leaving the economiser, in °C (°F)

p = weight of water to be heated, in kg/h (lb./h)

c = specific heat of the water = 1

t_0 = temperature of water entering the economiser, in °C (°F)

t = temperature of water leaving the economiser, in °C (°F)

k = heat transfer coefficient for the economiser, in kcal/m²/h/°C (B.T.U./sq.ft./h/°F), generally between 10 and 35 (2 - 7), according to the degree of cleanness of the exterior of the tubes, the velocity of the water in the tubes and of the gases around the tubes, and the temperature of the gases. As a mean value, we may take 20 - 30 (4 - 6).

S = heating surface of the economiser, in m² (sq.ft.)

r = ratio:

$$r = \frac{\alpha PC}{pc} = \frac{t - t_0}{T_0 - T} \quad (41.70)$$

Equation (41.69) may again be written:

$$S = \frac{\alpha PC}{k(1-r)} \cdot \ln \frac{T_0 - t}{T - t_0} \quad (41.71)$$

We may recall that: $\ln x = 2.3 \log x$

Eliminating αPC between eqns. (41.68) and (41.71), we have also:

$$M = kS \cdot \frac{(T_0 - t) - (T - t_0)}{\ln \frac{T_0 - t}{T - t_0}} \quad (41.72)$$

First problem. Checking an existing installation. Determination of the heat transfer coefficient k .

Data: PC ; pc ; T_0 ; T ; t_0 ; t ; S .

Unknown: k .

From eqn. (41.71), we have:

$$k = \frac{\alpha PC}{S(1-r)} \cdot \ln \frac{T_0 - t}{T - t_0} \quad (41.73)$$

Second problem. Design of an economiser. Determination of heating surface.

Data: PC ; pc ; T_0 ; t_0 and the desired temperature t .

Unknowns: T ; S .

From eqn. (41.68), we have:

$$r = T_0 - \frac{pc(t - t_0)}{\alpha PC} = T_0 - \frac{t - t_0}{r} \quad (41.74)$$

Hence S by eqn. (41.71), where we may take: $k = 15 - 25 \text{ kcal/m}^2/\text{h}/^\circ\text{C}$ ($3 - 5 \text{ B.T.U./sq.ft./h.}/^\circ\text{F}$).

Numerical example. It is required to add an economiser to a boiler working under the same conditions as that in the project for a superheater on p. 962. The weight of gases will be calculated in the same way, and we proceed from the value P found. We assume then that we have:

Steam production of the boiler:	50 t/h	
Temperature of gases leaving the boiler tubes:	$T_0 = 350^\circ\text{C}$	662°F
Temperature of feed water:	$t_0 = 90^\circ\text{C}$	194°F
Feed water temperature required (leaving economiser):	$t = 165^\circ\text{C}$	330°F
Weight of combustion gases (calculated p. 963):	$P = 121,900 \text{ kg/h}$	$273,050 \text{ lb./h}$

Calculation of T . We have (eqn. 41.70):

$$r = \frac{\alpha PC}{pc} = \frac{0.95 \times 121,900 \times 0.285}{50,000} = 0.66$$

where C is calculated by eqn. (41.46); and:

$$T = 350 - \frac{165 - 90}{0.66} = 236^\circ\text{C} \text{ (456}^\circ\text{F)}$$

Calculation of S . We have then (eqn. 41.71):

$$S = \frac{0.95 \times 121,900 \times 0.285}{25(1 - 0.66)} \times 2.3 \log \frac{350 - 165}{236 - 90} = 918 \text{ m}^2 \text{ (10,286 sq.ft.)}$$

Variation of $(t - t_0)$ as function of S . It is of interest to ascertain in what proportion t varies, that is to say $(t - t_0)$, when S is varied.

If we adopt the following conditions assumed as standard:

Moisture in bagasse:	$w =$	0.48	
Excess air:	$m =$	1.4	
Steam consumption per tonne cane:		450 kg/t.c.	1,000 lb./t.c.
Weight of bagasse per tonne cane:		252 kg/t.c.	560 lb./t.c.
Temperature of gases:	$T_0 =$	320°C	608°F
Temperature of water in feed tank:	$t_0 =$	90°C	194°F

and if we calculate the heating surface for the economiser corresponding to a crushing rate of 100 t.c.h., we obtain the graph of Fig. 41.21.

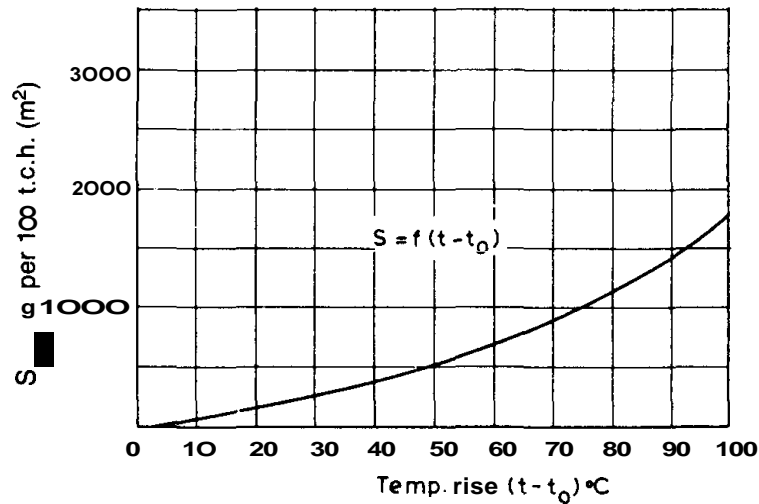


Fig. 41.21. Temperature rise of water as a function of economiser heating surface.

This is obviously an exponential curve, and demonstrates that there is no advantage in seeking too great a gain in temperature; this would require an enormous heating surface, which would hardly be payable. With 1,800 m^2 (20,000 sq.ft.), a temperature rise of 100°C (180°F) is obtained; with half this heating surface (900 m^2 ; 10,000 sq.ft.) we obtain 70°C or 70% of this temperature rise. It requires approximately 10 m^2 to gain each of the first 10°C, whereas it requires 35 m^2 for a temperature rise of 1°C between 90 and 100°C.

Safety margin

On the other hand, economisers are generally made of cast iron, to decrease the corrosive action of flue gases and ashes. It is necessary then to avoid evaporation of water in the economiser, for which it is not designed. It is necessary therefore to keep a safety margin between the temperature t of the water leaving the economiser and the boiling point corresponding to the working pressure of the boiler.

Von Pritzelwitz⁴⁸ recommends a margin of 40°C (72°F), Eigenhuis 20°C (36°F). We consider that it is wise not to go below 30°C (54°F).

Certificate

It is considered that an economiser should be certified for a pressure 20% higher than that of the boiler⁴⁹. This margin is not imposed by law; on the other hand, the interior diameter of the tubes is limited by law to a maximum of 10 cm (4 in.)

Ratio r

This is the ratio $(t - t_0)/(T_0 - T)$ between the rise in temperature of the water and the drop in temperature of the gases.

It varies between 0.6 and 0.8; that is, the water temperature increases by 0.7° when the gases drop by 1° .

Maximum economy

The maximum economy which can be obtained with an economiser is represented by the fraction of the total heat of the steam corresponding to the sensible heat, such as is given for example in Table 41.26 reduced by the heat quantity corresponding to the safety margin which must be allowed.

TABLE 41.27

MAXIMUM ECONOMY REALISABLE BY ECONOMISER
Safety margin 30°C (54°F). Feed water at 90°C (194°F)

Steam pressure (kg/cm^2) (p.s.i.g.)	Superheat		Saturation temp.		T_{max}		Total heat		Sensible heat		Recov- erable (%)	
	($^\circ\text{C}$)	($^\circ\text{F}$)	($^\circ\text{C}$)	($^\circ\text{F}$)	($^\circ\text{C}$)	($^\circ\text{F}$)	(kcal/kg)	(B.T.U./ lb.)	(kcal/kg)	(B.T.U./ lb.)		
10	142	Saturated	183	362	153	308	574	1033	65	118	11	
25	356	375	707	225	437	195	383	670	1205	111	200	17
30	427	400	750	235	454	205	400	681	1225	121	219	18
45	640	450	840	257	495	227	441	704	1263	148	266	21

It therefore depends on the steam pressure. With superheat as in Table 41.26 and keeping a safety margin of 12°C (54°F), it would be approximately as given in Table 41.27.

Velocities

Economisers are generally designed for the following velocities:

- (a) Water: $0.5 - 1.0$ m/s ($1.5 - 3$ ft./s), preferably $0.5 - 0.6$ m/s ($1.5 - 2$ ft./s)
- (b) Gases: $4 - 7$ m/s ($13 - 23$ ft./s), preferably $5 - 6$ m/s ($16 - 20$ ft./s).

These values are those which offer the best compromise between the optimum conditions for heat transfer and permissible pressure drops.

Variation of heat transfer with gas velocity. The coefficient of heat transfer k of finned tubes of cast iron varies with the interior diameter of the tube and the mean velocity of the gases (Fig. 41.22).

The two principal types of economiser used in the sugar factory are:

- (a) The "Coméconometeur" (French) and "Air Industrie" with steel tubes 51 mm dia. with cast iron fins 122×122 mm, $5 - 7$ mm thick.

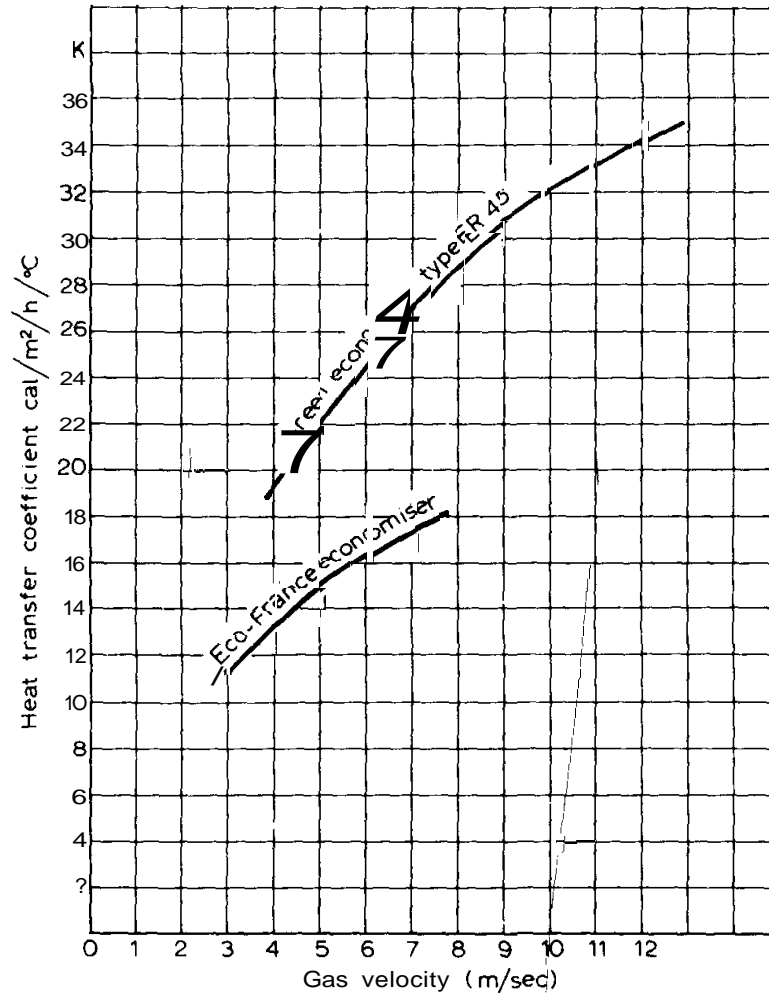


Fig. 41.22. Variation of heat transfer coefficient with flue gas velocity.

(b) The Green economiser, type ER 45, with fins in the shape of elongated hexagons; tubes are of 7 cm (2½ in.) interior dia., in staggered arrangement; heating surface is 1.4 m² per m (4.5 sq.ft./ft.) length of finned tube. The bends are of smaller diameter than of the tubes. This economiser takes up much more space but has a high coefficient of heat transfer.

When an economiser has bends of smaller diameter than the tubes, water remains in the lower portion of the tube when the economiser is emptied, and so causes rapid rusting. It is thus preferable to leave it full during the slack season.

AIR-HEATERS

In addition to the economiser, there is another type of equipment permitting of the partial recovery of the sensible heat of the combustion gases, which are still hot as they pass to the chimney. Instead of absorbing this heat in boiler feed water, it is absorbed by the air which is to be used for combustion in the furnace. The equipment is then called an "air-heater" or a "pre-heater".

Types of air-heater

There are three principal types of air-heater:

- (a) Tubular air-heaters
- (b) Plate type air-heaters (Fig. 41.23)
- (c) Regenerative heaters.

The first two types are ordinary heat exchangers. The difference between them is simply that the heat exchange surface consists in one case of tubes, and in the other case of plates of sheet metal.

The third type is based on a different principle: flue gases and air are passed alternately through flues containing brick chequer-work. The brickwork is heated by the passage of the gases, and gives up the heat thus stored when the gases are replaced by air. Alternatively, a suitable mass for absorbing the heat may be made to pass from the gas stream to the air stream, and conversely.

These recuperative heaters have a rather low efficiency, but present the advantage of not being subject to corrosion, and of not requiring any cleaning. They are not employed in sugar factories to our knowledge.

The term “pre-heater” is generally reserved for air-heaters of the first two types, with heat transfer surfaces of metal.

Such heaters cannot be used with flue gas temperatures above about 500°C (930°F). At higher temperatures, the tubes or the plates, even if of cast iron, would rapidly become unserviceable.

Tubular air-heaters are made with tubes of ordinary or chrome – copper steel, of 7 cm (2½ in.) o.d. The gases pass through the tubes and the air around them.

The surface of the tubes or the plates is often increased by the use of fins. As there is almost always corrosion at the cold end of the heater, tubular heaters have the advantage that it is much easier to replace a tube than a plate, since the plates are all welded together. As the heat transfer coefficients are comparable, plate-type heaters have been almost entirely replaced by tubular heaters.

As the fluids on both sides of the heat-exchange surface are gases, the heating surface is

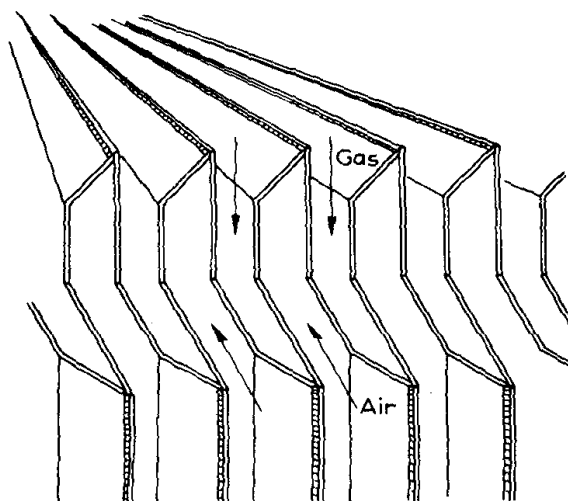


Fig. 41.23. Plate type air-heater.

calculated on the mean diameter of the tubes in the case of tubular heaters, *i.e.* from the mean of internal and external diameters.

Calculations for air-heaters

We have here exactly the same equations as for economisers (41.68–41.72), where p , c , t and t_0 apply to the air instead of to the water.

Ratio r . In an air-heater:

- (1) The weight of flue gases is greater than the weight of air (since $P_g = P_a + 1$)
- (2) The specific heat of the gases is higher than that of the air ($C > c$).

It follows that, in spite of the coefficient a , the ratio:

$$r = \frac{aPC}{pc} = \frac{t - t_0}{T_0 - T} \quad (41.75)$$

is generally greater than 1. It varies between 1.2 and 1.6, generally between 1.3 and 1.4. That is, when the flue gases drop by 1°, the air rises in temperature by 1.3 or 1.4°.

We may comment that if:

$$r = \frac{t - t_0}{T_0 - T} > 1$$

it follows similarly that:

$$\frac{T - t_0}{T_0 - t} > 1$$

so that the expression for S (eqn. 41.71) always has a plus sign. We may in fact write:

$$S = \frac{aPC}{k(1-r)} \cdot \ln \frac{T_0 - t}{T - t_0} = \frac{aPC}{k(r-1)} \cdot \ln \frac{T - t_0}{T_0 - t} \quad (41.76)$$

since:

$$\ln \frac{a}{b} = - \ln \frac{b}{a}$$

It will be necessary to make the same changes of sign in eqns. (41.72) and (41.73).

Value of a . For air-heaters of metal with effective circulation, a value of 0.92–0.95 may be adopted for the coefficient a .

Value of k . The coefficient of heat transfer in the air-heater is of the same order of magnitude as in an economiser.

Tromp⁵⁰ gives: $k = 10 \text{ kcal/m}^2/\text{°C/h}$ ($2 \text{ B.T.U./sq.ft./°F/h}$).

Clayton⁵¹ gives, as a function of gas velocity:

Velocity of gases:	5	10	m/s	$\left(\begin{array}{cc} 16 & 33 \\ 1.84 & 3.28 \end{array} \right)$
it:	9	16	kcal/m ² /°C/h	

"Sciences and Technology"^{51a} give the following values for the coefficient k , as function of gas and air velocities:

Gas velocity:	4	6	8	10	m/s
	13	20	26	33	ft/s
Air velocity:	6	8	10	12	m/s
	20	26	33	40	ft/s
Coefficient k :	8	10.5	11.5	12.5	kcal/m ² /°C/h
	1.6	2.2	2.4	2.6	B.T.U./sq.ft./°F/h
Pressure drop gas side:	4	7	13	20	mm water
	0.2	0.3	0.5	0.8	in. water
Pressure drop air side:	8	13	20	30	mm water
	0.3	0.5	0.8	1.2	in. water

In general we may use:

$$k = 15 - 20 \text{ kcal/m}^2/\text{°C/h} \text{ (3 - 4 B.T.U./sq.ft./°F/h), often a figure of 18 (3.7)}$$

Velocities. The velocities generally used are as follows:

(a) Air: 3 - 8 m/s (10 - 26 ft./s), in general 5 - 7 m/s (16 - 23 ft./s), calculated for the cold air. A larger cross-section will be taken for the hot air so as to give the same velocity, or one slightly higher.

(b) Gases: 5 - 12 m/s (16-40 ft./s), generally 10 - 12 (33 - 40).

Design of air-heater. It is required to determine the heating surface of a metal air-heater with systematic circulation.

Data: PC ; pc ; T_0 ; t_0 and the desired temperature t .

Unknowns: T ; S .

Numerical example. We shall calculate the heating surface in the following case. An air-heater is to be installed in a boiler working under the same conditions as that for the design of the superheater and of the economiser, but not furnished with an economiser. The weight of gases being calculated always in the same way, we commence the calculations here with the value of B found (p. 963). We assume then that we have:

Steam production of the boiler:	50 t/h	
Temperature of gases leaving the boiler tubes:	$T_0 = 350^\circ\text{C}$	662°F
Temperature of ambient air:	$t_0 = 30^\circ\text{C}$	86°F
Temperature of hot air required:	$t = 200^\circ\text{C}$	392°F
Weight of combustion gases:	$P = 121,900 \text{ kg/h}$	$273,050 \text{ lb./h}$

We require to calculate the heat exchange surface.

Solution. We may assume that the data are those applying before installation of the air heater. It is logical to calculate the air heater on the assumption that it will give a certain improvement in operating conditions:

(a) Reduction of excess air from 40 to 36%: $m = 1.36$.

(b) Increase in temperature of gases leaving the boiler from 350 to 360°C (662 – 680°F).

These two improvements reduce the weight of bagasse to: $B = 23,400 \text{ kg/h}$ (52,400 lb./h) and the weight of gases to: $P = 118,700 \text{ kg/h}$ (265,900 lb./h).

Calculation of p. The weight of air necessary, in the same way, is (eqn. 41.29):

$$P_a = 5.76 (1 - w)m = 5.76 \times 0.52 \times 1.36 = 4,073 \text{ kg/kg (lb./lb.) of bagasse}$$

or a total:

$$p = 4,073 \times 23,400 = 95,300 \text{ kg/h (213,425 lb./h)}$$

Calculation of T. We have:

$$r = \frac{0.93 \times 118,700 \times 0.287}{95,300 \times 0.24} = 1.385$$

We have calculated C by eqn. (41.46). For c , the value varies only slightly with temperature, as will be seen readily from the columns for N_2 and O_2 in Table 41.13, and no appreciable error will be involved, in calculations for the heater, if we take in all cases $c = 0.24$.

$$T = 360 - \frac{200 - 30}{1.385} = 237^\circ\text{C (459}^\circ\text{F)}$$

Calculation of S. Taking the heat transfer coefficient as $18 \text{ kcal/m}^2/^\circ\text{C/h}$ (3.7 B.T.U./sq.ft./°F/h) we have:

$$S = \frac{0.93 \times 118,700 \times 0.287}{18 (1.385 - 1)} \times 3 \log \frac{237 - 30}{300 - 200} = 1,176 \text{ m}^2 (12,870 \text{ sq.ft.)}$$

Variation of $(t - t_0)$ as a function of S

We may now *calculate*, as for the economiser, the variation of the rise in temperature of the air $(t - t_0)$ as function of the heating surface S employed.

Adopting the following conditions, which have been purposely chosen to correspond to those adopted for the economiser:

Moisture in bagasse:	$w =$	0.48	
Excess air:	$m =$	1.36	
Steam consumption per ton cane:		450 kg	1,000 lb.
Temperature of flue gases leaving boiler tubes:	$T_0 =$	330°C	626°F
Temperature of ambient air:	$t_0 =$	30°C	86°F

and plotting as ordinates the air-heater surface corresponding to a crushing rate of 100 t.c.h., we obtain the graph of Fig. 41.24.

It exhibits exactly the same form as that found for the economiser, and prompts the same comments.

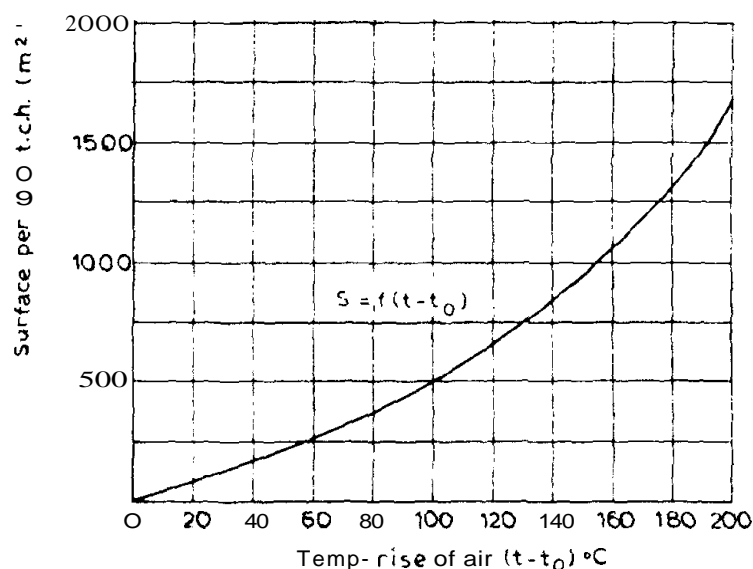


Fig. 41.24. Temperature rise of air as a function of air-heater surface.

Limit of temperature obtainable

Up to what point is it possible to pre-heat the combustion air?

Tromp⁵² indicates 200°C (400°F) as a temperature obtained in practice. Clayton⁵³ states that pre-heating up to 250°C (482°F) may easily be obtained, but considers it wise not to exceed 180°C (356°F), otherwise damage will be caused to the grate bars and to refractories in the furnace, on account of the increased temperature of combustion.

However, special refractories are made for very high temperatures; but their cost would appreciably reduce the advantage of the additional gain in temperature sought.

These remarks apply only to older types of furnace. With spreader-stoker furnaces and water walls, the temperature is no longer the limiting factor. It is then the cost of the pre-

heater that must be balanced against the saving in bagasse realised; Fig. 41.24 shows that each 10° of further gain in air temperature requires a greater and greater heating surface. There will thus be, as for an economiser, an optimum rapidly reached as a function of the respective magnitudes of the financial charges involved for the air-heater and the value of the bagasse saved. The two corresponding graphs could be drawn and would establish the point where they show the most attractive saving.

Increase in combustion temperature

What is the effect of the increase in the temperature of the combustion air on the temperature of combustion?

Referring back to eqn. (41.44), we might consider that the combustion temperature would increase by exactly the same amount as the increase in air temperature.

However, this formula contains an approximation which was justified when the temperature of the fuel was the same as that of the air and varied with it. This is not the case here, so we shall repeat the calculation. We shall designate:

t_0 = the temperature of the bagasse and the ambient air

t = the temperature of the heated air

T_0 = the temperature of combustion which would be obtained with the cold air

T = the temperature of combustion obtained with the heated air.

Replacing T by T_0 in the second term, eqn. (41.41) gives the combustion temperature T_0 in the case of cold air.

If the air is heated, the supplementary terms to be added to the two sides of the equation should themselves be equal. Hence:

$$\int_{t_0}^t P_a c_a dt = \Sigma \int_{T_0}^T PC At \quad (41.77)$$

or, considering the mean specific heat:

$$P_a [c_a]_{t_0}^t (t - t_0) = \Sigma P [C]_{T_0}^T (T - T_0) \quad (41.78)$$

Hence:

$$\frac{T - T_0}{t - t_0} = \frac{P_a [c_a]_{t_0}^t}{P [C]_{T_0}^T} = \frac{0.24 P_a}{0.33 P_o} = \frac{0.24 \times 5.76 (1 - w)m}{0.33 [5.76 (1 - w)m + 1] + 0.33}$$

or:

$$\frac{T - T_0}{t - t_0} = \frac{1}{1.375 + \frac{1}{4.19 (1 - w)m}} \quad (41.79)$$

Giving extreme values to w and to m , it will be seen that the ratio:

$$\theta = \frac{T - T_0}{t - t_0} \quad (41.80)$$

lies between:

0.57 and 0.62

For mean values: $w = 0.46-0.48$; $m = 1.4 - 1.5$; we have:

$$B = 0.6$$

In other words, in a bagasse furnace, *the increase in the combustion temperature due to pre-heating of the air is 60% of the increase in air temperature.*

Corrosion

Dew-point. Air-heaters are generally fabricated from steel. The dew-point of the combustion gases lies in the region of $60 - 65^\circ\text{C}$ ($140 - 150^\circ\text{F}$), hence it may be supposed that the danger of condensation on the walls of the heater, and consequently of corrosion, *would* be very limited, since the gases leave the air-heater at a temperature which is always greater than 100°C (212°F). In practice, points always exist where pockets of gas are formed, the temperature of which can then drop to the dew-point.

Recirculation. To avoid such risk of corrosion, it is arranged that the temperature of the air itself does not fall below 65°C (150°F). This is obtained by using "recirculation" of part of the hot air.

This consists simply of making a certain proportion of the hot air, leaving the heater, describe a closed circuit: it is taken and returned to the air inlet.

Calculation of proportion of air to be recirculated. Let x be the fraction of the air used which is to be recirculated (Fig. 41.25). We have then:

$$1 t_0 + xt = (1 + x)t'_0 \quad (41.81)$$

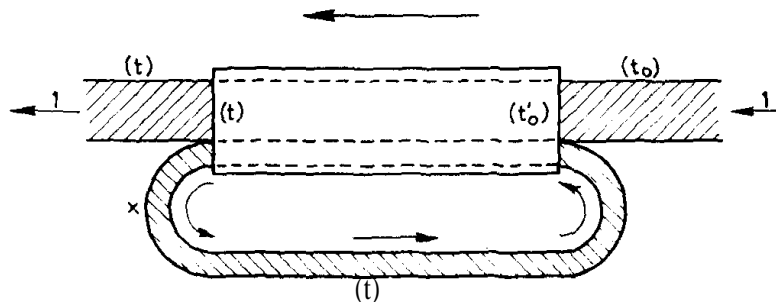


Fig. 41.25. Proportion of air recirculated.

whence:

$$x = \frac{t'_0 - t_0}{t - t'_0} \quad (41.82)$$

t'_0 = dew-point (approximately 65°C; 150°F)

t_0 = temperature of ambient air (generally 30°C; 86°F)

t = temperature of hot air leaving the heater, in °C (°F).

Or, in the general case:

$$x = \frac{65 - 30}{t - 65} = \frac{35}{t - 65} \quad \left(\frac{150 - 86}{t - 150} = \frac{64}{t - 150} \right) \quad (3)$$

It is seen that, for: $t = 100^\circ\text{C}$ (212°F), the quantity of air to be passed through the heater will be double that used for combustion.

Consequences of recirculation. When the hot air temperature t is sufficiently high, the *recirculation* presents little difficulty: the efficiency of the heater is not appreciably modified, since the increase in velocity *partially* compensates for the decrease in temperature difference between gas and air. It is the cross-section of air passage and air velocity which must mainly be taken into account with recirculation.

Measures to prevent corrosion. In spite of recirculation, premature corrosion of air-heaters is often experienced, particularly at the gas exit. To avoid having to replace the whole heater in this case, it is preferable to fabricate it in two portions, the second comprising about one-third or one-quarter of the whole; it is then sufficient to replace this second section when it develops holes.

Lagging

It is necessary to insulate the hot air duct passing from the air-heater outlet to the furnace, carefully otherwise a large proportion of the heat recovered will be lost.

Maximal economy

The maximal economy which an air-heater can give does not depend, as with an economiser, on the steam pressure of the boilers, but only on the maximum temperature permissible for the air (with old furnaces) or on the optimum dimension of the heater (water-walled furnaces). Generally, the maximum improvement in efficiency is of the order of 12%.

It will be seen (c/. Table 41.27) that the maximum economy obtainable with a heat exchanger in the flue gases is greater:

- (a) With an air heater, when the boilers are of low pressure
- (b) With an economiser, when the boilers are of high pressure.

CHOICE BETWEEN ECONOMISER AND AIR-HEATER

The economy realised is determined by the temperature to which the flue gases can be reduc-

ed. Any comparison between the economiser and the air-heater should thus be made for equal final temperatures of the gases going to the stack.

We may remark that, with good modern boilers, each of these two heat exchangers permits the maximum economy practicable to be attained, which corresponds to flue gases leaving at about 200°C (400°F); a temperature approaching 150°C (300°F) could be obtained if such heat recovery could be financially or practically justified.

Consequently, the factors which guide the choice to be made are the following:

A. Heating surface. (a) The heat transfer coefficient should be much better in an economiser (gas to water) than in an air-heater (gas to gas). In practice, the difference is small: approximately 25% in favour of the economiser.

(b) The mean difference in temperature between heating fluid and heated fluid, which influences the logarithmic term in the formulae (41.71) and (41.76), is generally greater for the air-heater, so reducing the heating surface required.

Taking these two factors into account; for equal technical economy, the heating surfaces are approximately equivalent, that of the air heater being only 20 or 25% greater than that for the economiser.

B. Cost. (a) The economiser operates under pressure, and is of cast iron. The air-heater works at atmospheric pressure, hence is not subject to pressure, and is of steel. It should therefore cost much less.

(b) However, it requires an additional fan, in the case where forced draught is not already used.

Finally, the purchase price is rather definitely in favour of the air-heater for equal heating surface.

C. Maintenance and working life. Both types of equipment require little maintenance. Every two or three years, it is necessary to dismantle the economiser to remove soot and particles of rust caught between the fins of the tubes; but, being made in thick cast iron, it lasts much longer than the air-heater, which is made of thin sheet steel. One may estimate approximately 20 years life for an economiser and 10 years for an air-heater.

Furthermore, the furnace temperature being lower with an economiser: (1) the grate bars are kept cooler and thus will have a longer life; (2) as the fusion temperature of bagasse ash is often rather low, there is less risk of clinker formation.

D. Associated advantages. The air-heater improves combustion and generally permits a better regulation of excess air. This was its greatest advantage, but it is much less marked with modern boilers, where combustion is already very efficient. It also requires additional ducts for air and flue gases, as well as demanding refractories of better quality; these factors increase the cost of installation.

The economiser has in its favour the lower pressure drop in flue gas, as well as reduced power for fans. The higher temperature of feed water favours stability of water level.

Finally, while the cost of the air heater may be slightly lower, the complete installation cost remains much the same for both cases.

Fuel Oil. Most boilers are equipped with fuel-oil burners which are used at the start-up and in places where it is required to supply power for outside uses in the slack season. In these cases, the air-heater has disadvantages:

(a) In operation with the gas and fuel oil together, it is necessary to heat the combustion air to about 80°C before the air-heater, to avoid corrosion due to sulphuric acid produced by combustion of the sulphur content of the fuel oil. This pre-heating uses some steam.

(b) Hot air cannot be used in the oil burners, since the primary air fan should supply a pressure of 500 - 800 Pa at high rating, whereas the fuel-oil fan should supply 2000 - 2500 Pa at low output. These two pressures being incompatible means that the oil burners have to be supplied with cold air.

(c) In operation with oil only, the heater is not used and the efficiency is low.

Economic efficiency. In general, the objective is to reduce the flue gas temperature to 150–200°C at the base of the chimney. It is scarcely economic to take this any lower.

Conclusion. In the final count, faults and advantages partially balance, but with some advantage in favour of the economiser. The choice will sometimes be guided by a matter of detail, of convenience of installation or facility of insertion in a new unit. A horseshoe furnace, requiring forced draught, will for example favour the air-heater.

However, from the fact of its performance and its life, and in the case of high-pressure boilers (above 17.6 kg/cm²; 250 p.s.i.), the author definitely prefers the economiser.

A neat solution consists of installing both types of equipment in the path of the flue gases. We may for example place the economiser first, followed by the air-heater. This is the arrangement which is theoretically most logical: it is of advantage to stop the heat exchange in any heat exchanger when the temperature of the hot fluid approaches too close to that of the cold fluid, and then to transfer to exchange of heat with a cooler fluid. However, by placing the air-heater first in the gas circuit (Fig. 41.26), it may be possible to dispense with recirculation; this is a great advantage and, for certain types of boiler, can be conducive to a simpler installation.

Such use of the two heat exchangers presents several advantages:

(a) A large safety margin for the economiser, since only part of the possible temperature rise is being sought.

(b) No risk of deterioration of refractories due to the high air temperature, for the same reason.

(c) In the case of a breakdown of the first heat exchanger, there is a possibility that the second exchanger will compensate for it to a certain extent, consequent on the increase of temperature in the gases entering it.

Efficiency

When the efficiency of an economiser or air-heater is calculated, it is sometimes found to reach or exceed 1, instead of lying in the neighbourhood of the value expected: $a = 0.90$.

This phenomenon is explained by the fact that part of the unburnt solids and of the hydrogen contained in the gases continue to burn while passing through the air-heater. (There would be no question of CO, which requires much higher temperatures.)

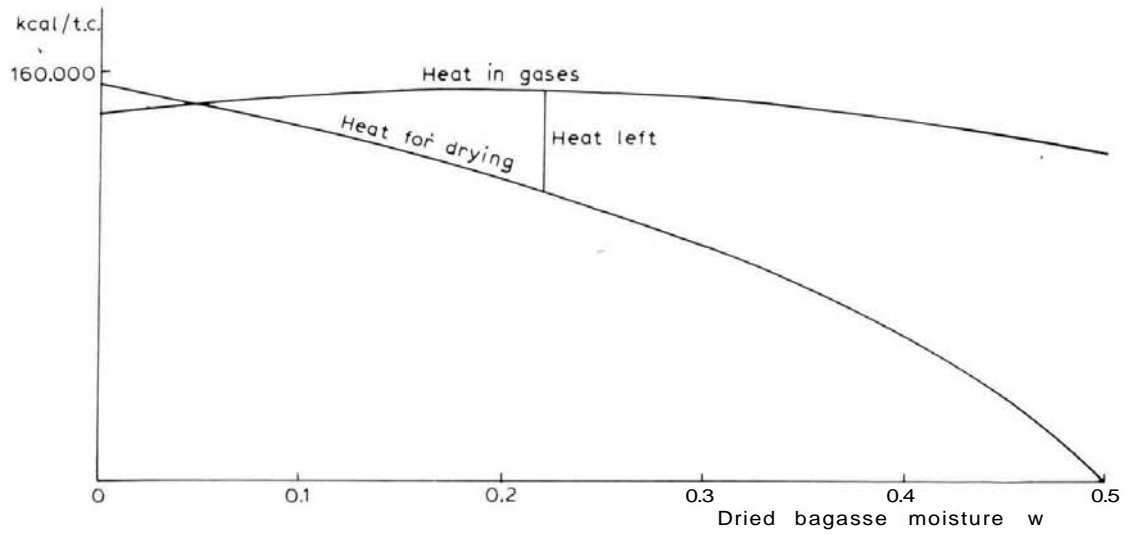


Fig. 41.25a. Bagasse drying. Heat available in flue gases and heat used for dryer as a function of final bagasse moisture when drying 100% of the bagasse.

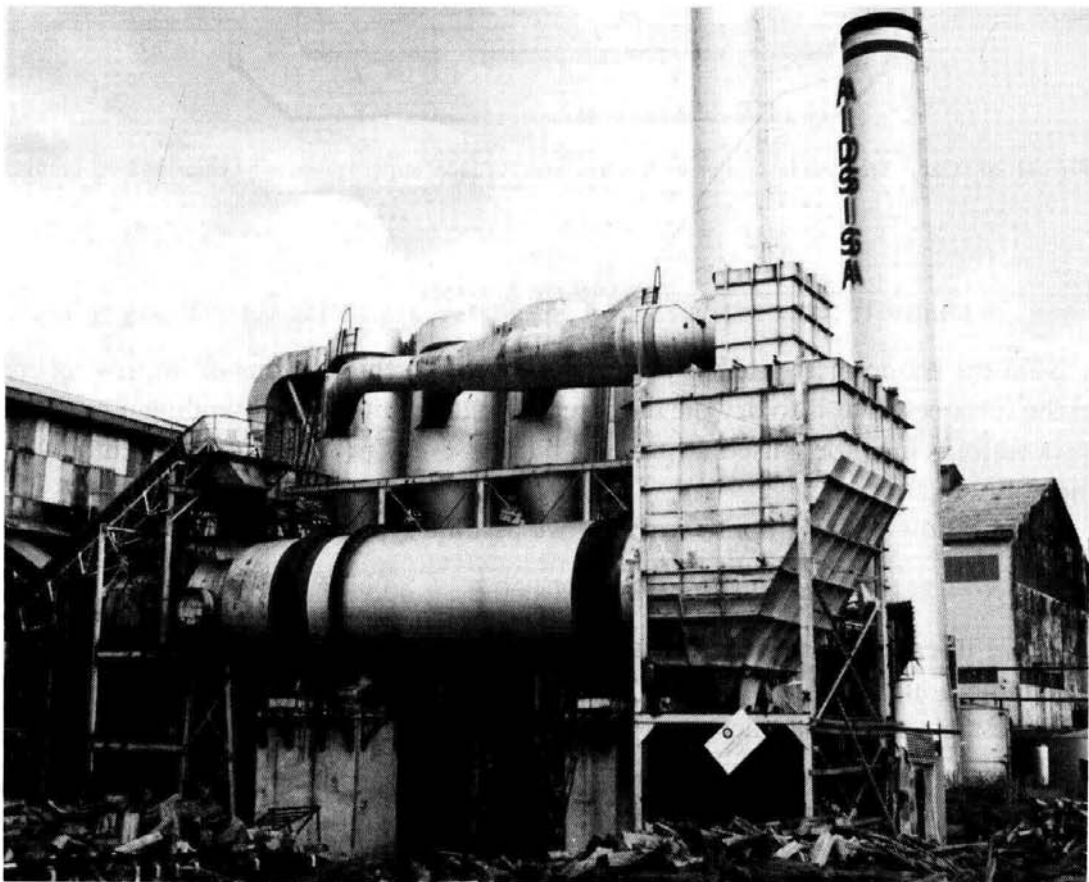


Fig. 41.25b. Bagasse dryer.

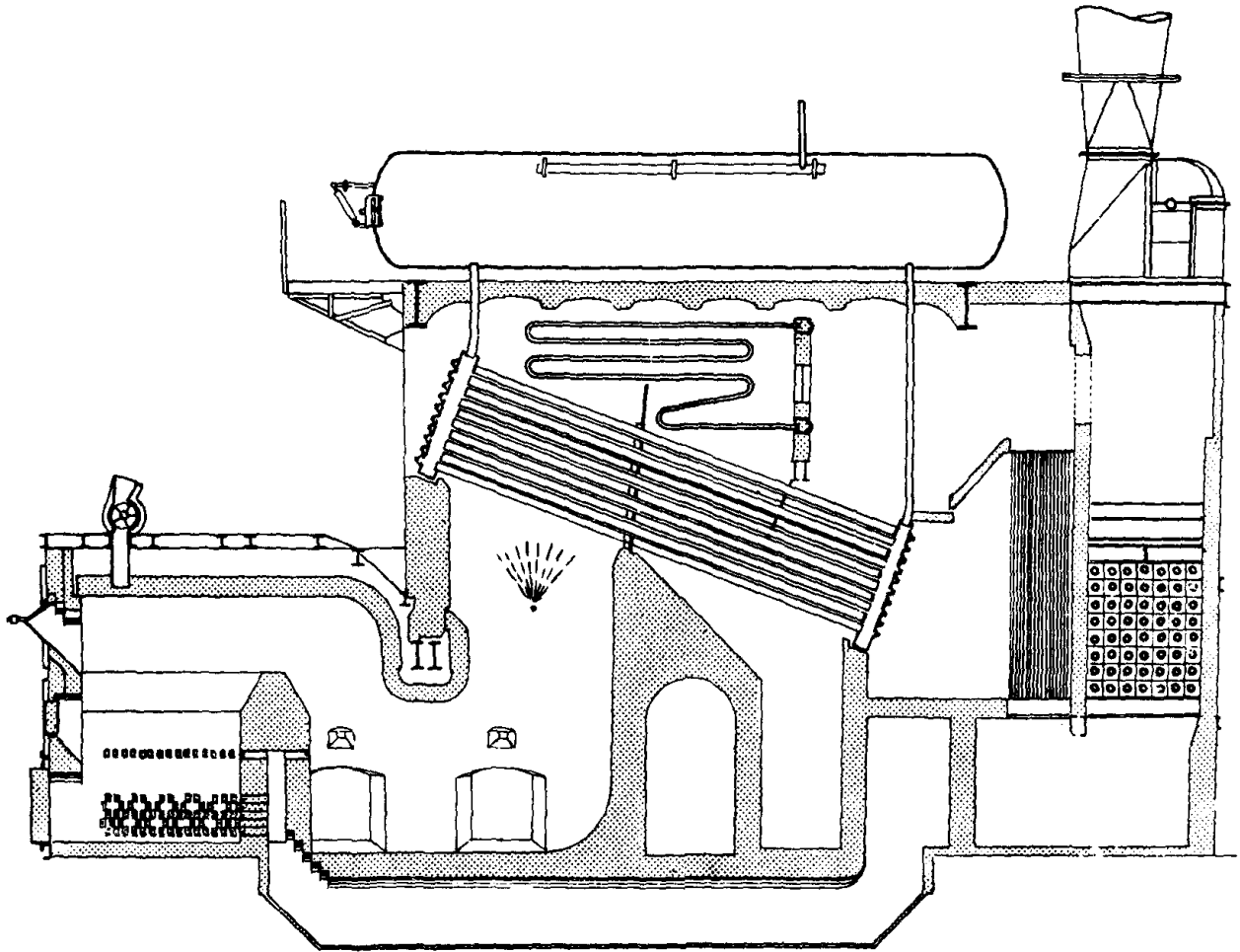


Fig. 41.26. Cail-Steinmuller boiler with horseshoe furnace, superheater, air-heater and economiser.

DRYING OF BAGASSE

Apart from the economiser and the air heater, there is a third means of improving combustion in the furnaces, that is to dry the bagasse leaving the mills before sending it to the boilers. This procedure is much less used at present, but has every prospect of being employed more and more in the future.

To consider how drying can be realised, we assume typical or frequent conditions. Suppose:

Fibre in cane:	•	$l = 0.15$
Ambient air:		$t = 30^{\circ}\text{C}$
Excess air to the furnace:		$m = 1.4$
Bagasse at 50% moisture per t.c.:		$B_0 = 300 \text{ kg}$
Moisture in bagasse leaving the mills:		$w = 0.5$
Ash % in bagasse:		$c = 2.5\%$
Temperature of gas admitted to the outlet from the dryer:		$T = 120^{\circ}\text{C}$
Temperature of gases leaving the boiler tubes:		$T_0 = 350^{\circ}\text{C}$

B_0 and T_0 are values before installation of the dryer. The temperature T_0 depends on various factors and in particular the moisture of the bagasse. Moisture content must thus be taken into account and, taking a conservative figure, we assume: $T_0 = (400 - 100w)$ °C.

Similarly, the weight B_0 of bagasse will obviously vary with moisture content to which it is reduced starting from the 50% assumed. For unit weight of bagasse of 50% moisture, we would have:

$$\begin{aligned} \text{At 50\% moisture: } & 0.50 + 0.50 = 1 \quad \text{hence: } 300 \text{ kg/t.c.} \\ \text{At 40\% moisture: } & 0.50 + 0.40 = 0.9 \quad \text{hence: } 270 \text{ kg/t.c.} \\ \text{At 30\% moisture: } & 0.50 + 0.30 = 0.8 \quad \text{hence: } 240 \text{ kg/t.c.} \\ \text{At } w \text{ moisture: } & 0.50 + w \quad \text{hence: } 300(0.5 + w) \text{ kg/t.c.} \end{aligned}$$

Water to be evaporated. To determine the quantity X of water to be evaporated per kg of bagasse to reduce it to any moisture content w , starting from a moisture of 0.5, this quantity will be such that:

$$w = \frac{0.5 - X}{1 - X} \quad \text{hence: } X = \frac{0.5 - w}{1 - w} \text{ kg/kg of bagasse.}$$

Or:

$$\begin{aligned} X = 0 & \quad \text{for } w = 0.5 \\ X = 0.167 & \quad \text{for } w = 0.4 \\ X = 0.286 & \quad \text{for } w = 0.3 \\ X = 0.375 & \quad \text{for } w = 0.2 \\ X = 0.444 & \quad \text{for } w = 0.1 \\ X = 0.500 & \quad \text{for } w = 0 \end{aligned}$$

Weight of gas. The weight of gas leaving the boiler has the value (formula 41.30 allowing for ash content):

$$P = 5.76(1 - w) 1.4 + 1 = 9.06 - 8.06w \text{ kg/kg of bagasse}$$

hence the total weight of gases:

$$BP = 300(0.5 + w)(9.06 - 8.06w) = 1359 + 1509w - 2418w^2 \text{ kg/t.c.}$$

Heat required. The bagasse must be heated to the final temperature at which it leaves the dryer. The specific heat of fibre has a value:

$$c_f = 0.266 + 0.00116t$$

Between 30 and 120°C, it varies between 0.301 to 0.405. Since this is a relatively insignificant term, we assume a mean value at $c_f = 0.36$; the heat required to heat the fibre is thus: 300

$\times 0.5 \times 0.36 (120 - 30) = 4,860$ kcal/t.c. We also require to heat the water contained in the bagasse and evaporate it. The total heat of this vapour (table 41.1) at 120°C is 646. We thus have:

$$(646 - 30)X = 316 \cdot \frac{0.5 - w}{1 - w} = \frac{308 - 616w}{1 - w} \text{ kg/kg of fresh bagasse}$$

or:

$$300 \frac{308 - 616w}{1 - w} = \frac{92,400 - 184,800w}{1 - w} \text{ kcal/t.c.}$$

The total heat necessary is thus:

$$\text{Ch.N} = 4,860 + \frac{92,400 - 184,800w}{1 - w} = \frac{97,260 - 189,660w}{1 - w} \text{ kcal/t.c.}$$

The heat available (Ch.D) has the value, assuming an efficiency of 0.85 (if the dryer is insulated: otherwise 0.83):

$$BP(375 - 120)0.285 \times 0.85 = 61.8 BP$$

or:

$$61.8 (1.359 + 1.509w - 2.418w^2) \text{ kcal/t.c.}$$

or:

$$\text{Ch.D} = 83.986 + 93.256w - 149.432w^2 \text{ kcal/t.c.}$$

Hence the table of heat quantities necessary (Ch.N) and disposable or available (Ch.D):

w	w_2	<i>Ch.N</i> (kcal/t.c.)	<i>Ch.D</i> (kcal/t.c.)	<i>Balance disposable</i> <i>for drying</i> (kcal/t.c.)
0.5	0.25	4,860	93,156	93,156
0.4	0.16	35,660	97,379	61,719
0.3	0.09	57,660	98,514	40,854
0.2	0.04	74,160	96,660	22,500
0.1	0.01	86,993	91,817	-4,824
0	0	97,260	83,986	-13,274

There is no advantage in drying the bagasse below 10% moisture, as it would be liable to increase to 10% by absorption of atmospheric humidity. The drying operation could, for ex-

ample, be taken to 15 or 20%, and the remaining available heat in the gases could be used in another heat exchanger such as a small economiser. The operation could indeed well commence with the economiser, the gases from which would go to the bagasse dryer. This solution would indeed present the advantage that the economiser would show a very good return (see p. 970).

The volume of the dryer is approximately 12–15 m³/t/h of water to be evaporated.

The drying of bagasse is rather expensive, but it offers the great advantage of being the procedure which permits the flue gases to be brought to the lowest temperature, and thus giving the equipment the greatest recovery of heat from the flue gases. Whereas economisers and air heaters allow reduction of the gas temperature to 150–200°C, the bagasse dryer allows cooling of the gases to 90°C, the only limit being imposed by the necessity to avoid cooling to the dew point of 60–70°C. However, it is not recommended to attempt such cooling, on account of the dimensions and costs of the dryer, which become excessive.

Another advantage of the dryer is that the bagasse introduced to the furnace is heated, which facilitates combustion and rapid ignition. This is not taken into account in the calculations, but does contribute to improved combustion.

On the other hand, dry bagasse is readily ignited and precautions should be taken in the dryer and in the transfer of bagasse to the boilers. These consist especially in placing a steam valve at the entry to the dryer so as to extinguish readily any fire which may break out in the dryer.

The velocity of gas in the dryer should not exceed 2.5 m/s, to avoid entraining fine particles. It is generally kept between 2 and 2.5 m/s.

The main disadvantage of bagasse drying is its cost, which may attain three times that of economisers or air heaters. For this reason it is not often practised; but it is the solution which should be considered when it is desired to conserve bagasse for slack-season use and in particular for conversion to pellets.

However, if the whole of the bagasse is dried, these velocities lead to dryers of huge dimensions, cumbersome and expensive. In this case a compromise will be chosen where a dust separator is interposed before the chimney.

A logical solution consists of passing the gases first to a heat exchanger, economiser or air-heater, which forms part of the boiler unit and reduces gas temperatures to 280–300°C. The gases are then sent to the dryer, which can be of rotating-drum type resembling the “Roto-Louvre” type of sugar dryer, furnished with vanes which move the bagasse along the length of the rotating cylinder. On account of its length, it can be divided into two sections, the first taking the gas away from the boiler, the second returning it to the base of the chimney.

It will obviously be desirable to provide a system of bagasse feeding and removal at entry to the dryer and exit from it, which effectively prevents air and gas leakage, following, for example, the principle indicated in Fig. 41.9.

Comment. We have given the complete calculation, for drying bagasse to zero moisture content, since progress in the industry and in technique may one day render such total drying desirable. However, for combustion of bagasse in the spreader-stoker furnaces used at present, drying is generally not taken below 30% in the dried bagasse, since a drier fuel would

involve risk of higher combustion temperatures which could cause deposit of fused ash on the boiler tubes.

On the other hand it would appear that drying by a combination of pneumatic conveying and fluidised bed operation would reduce operating costs relative to those of a drum drier, and that separation of the dried bagasse into fine and coarse fractions would permit of handling and combustion better adapted to the different fractions. The dimension for separation of the two fractions would correspond to a length of 1.5–2 mm.

BOILER SETTINGS

Bricks

Most factories use the standard European brick of 220 X 110 X 60 mm (9 x 4½ x 2⅜ in.).

Resistance to temperature

It is now possible to obtain temperatures of 1,340°C (2,450°F) in ordinary furnaces with cold air, 1,450°C (2,650°F) in furnaces supplied with pre-heated air.

We should choose:

- (a) In the former case, bricks of at least 20–25% of alumina
- (b) In the latter case, bricks of at least 30–35% alumina.

The prices of these two types of refractory are about 25 and 50% higher, respectively, at the point of manufacture, than the price of ordinary bricks of 15—18% of alumina. However, when they are purchased at a great distance from the sugar factory, this difference in cost price becomes less important, since expenses of handling, transport, and freight are the same for all types of brick, and consequently independent of the quality.

Expansion joints

In all brick walls of the furnace and the boiler setting, it is necessary to allow expansion joints of 0.5 cm (3/16 in.) every 60 cm (2 ft.). They will not, however, be placed in the walls of the horseshoe or a furnace of hearth type.

In the furnace arches, the bricks should be arranged in separate rows. The arch thus becomes a series of independent small arches in juxtaposition, of length equal to that of one arch brick. This has the advantage of avoiding half-bricks. Thus, an expansion joint of 0.5 cm (3/16 in.) can be arranged between these small arches, every three arches.

Bricklaying. In Europe, a bricklayer lays 700 firebricks in 8 hours. In tropical countries, the figure sometimes falls to between 100 and 400.

Drying out new brickwork

When construction of furnaces or flues is finished, they should be dried out very carefully, by lighting a small fire, which will be increased progressively during at least three days.

Screen walls.

In modern boilers, the heavy brickwork side walls are replaced by "screen walls", in other

words, the wall is omitted, the outer rows of tubes are joined together by welded steel plates, and the outer side is insulated. This simplifies the construction and effects a substantial economy.

Cross-section of flues

(A) *Velocity of gases.* A cross-section should be adopted for flues such that the velocity of the gases is:

(a) In the case of natural draught: 4 – 10 m/s (13 – 33 ft./s); preferably 5 m/s (15 ft./s)

(b) In the case of forced draught: 8 – 16 m/s (25 – 50 ft./s); preferably 10 – 12 m/s (30 – 40 ft./s)

B *Volume of gases.* Equation (41.35) gives us:

$$V_g = 4.45 (1 - w)m + 0.572 w + 0.672 \text{ Nm}^3/\text{kg of bagasse} \\ (71.3 (1 - w)m + 9.17 w + 10.75 \text{ Ncu.ft./lb.})$$

With $w = 0.48$ and $m = 1.5$:

$$V_g = 4.42 \text{ NmVkg bagasse (70.8 Ncu.ft./lb.)}$$

at N.T.P. Or at 250°C (480°F):

$$V_{gt} = 4.42 \times \frac{273 \times 250}{273} = 8.5 \text{ m}^3/\text{kg bagasse (135 cu.ft./lb. bagasse)}$$

Say: 8-9 mVkg (130 – 140 cu.ft./lb.) for mean values of w and m .

(C) *Section of flues.* In the general case of forced draught, and basing calculations on a gas velocity of 11 m/s (36 ft./s) and on a volume of 9 m³ (140 cu.ft.) of gases per kg (lb.) of bagasse, the cross-section of the flues should be:

$$s = \frac{9,000}{3,600 \times 11} = 0.23 \text{ m}^2/\text{t bagasse/h (2.42 sq.ft./t/h)} \quad (41.84)$$

If it is desired to relate this cross-section, not to the weight of fuel burnt but to the weight of steam produced, it will be necessary to multiply the value just found by the factor / in Table 41.22. If for example we adopt a factor of 2.17, we would have:

$$s = 0.50 \text{ mVt steam/h (5.3 sq.ft./t/h)} \quad (41.85)$$

In order to maintain the temperature and the rate of combustion, it is necessary to pass the

required quantity of air through the furnace and over the fuel bed. Since the path of the gases is complex, with many resistances to overcome (passing through the fuel bed and between the boiler tubes, sudden changes of direction, etc.), and since it is necessary to give the gases a rather high velocity, as seen in the foregoing figures, maintaining this flow of gas demands a certain expenditure of energy which is normally supplied in the form of a motive pressure.

This actuating pressure may be obtained from two sources:

- (a) From the thermal energy produced by the combustion: this is *natural draught*
- (b) From an external source, fan or some other means; this is *forced draught*.

We shall disregard natural draught, which is no longer found in the sugar factory and which is discussed in the first edition of this book (p. 730). Requirements of a chimney for the general case of mechanical draught will be discussed.

Draught in the flues

The draught or suction is the difference between the outside pressure and the pressure in the interior of the flue. It is expressed in mm or inches of water (1 in. of water (= 1 in. w.g.) = 0.036 p.s.i.), and is easily measured in an existing installation, with the aid of a small U-tube containing water (Fig. 41.27).



Fig. 41.27. Measure of draught.

The chimney

A. Velocity of gases leaving the chimney. The velocity v of gases leaving the chimney involves a loss of energy. It is therefore undesirable to overdo it. With natural draught, it was always necessary to have sufficient velocity to avoid disturbance to the operation of the chimney by the vertical component of the prevailing winds; for this purpose a value v_s of 4.5-5 m/s (15-16 ft./s) was employed. For mechanical draught, the exit velocity, with short chimneys, helps to disperse more effectively the "fly ash", which is often a nuisance to the factory and its neighbourhood. Generally a velocity of 12-16 m/s (40-50 ft./s) is used.

B. Cross-section of the chimney. The choice of velocity of gases leaving the chimney involves the determination of its cross-section at the top. We know the output of gas to be allowed for:

$$Q = \frac{B \times V_{gt}}{3,600} \quad (41.86)$$

Q = volume of gases to be handled by the chimney, in m^3/s (cu.ft./s)

B = weight of bagasse burnt in the furnaces served, in kg/h (Ib./h)

V_{gt} = volume of combustion gases, given by eqn. (41.35), and converted to the temperature and pressure ruling at the top of the chimney.

$$V_{gt} = V_{g-} \times \frac{273 + t}{273} \quad \left(\frac{V_g \times \frac{459 + t}{459 + 32}}{\right)$$

If the weight of steam produced is known more accurately than that of bagasse burnt, the term B in eqn. (41.86) may be replaced by p/f :

p = weight of steam produced by the boilers, in kg/h (Ib./h)

f = factor calculated or given by Table 41.22.

The temperature of the gases at the top of the chimney may be taken equal to:

$$t_s = t_c - a(t_c - t_a) \quad (41.87)$$

t_s = temperature of the gases at the top of the chimney

t_c = temperature of the gases at the base of the chimney

t_a = ambient temperature

a = coefficient having a value:

0.001 H (0.0003 H) for brick or stone chimneys

0.002 H (0.0006 H) for chimneys of reinforced concrete

0.003 H (0.0009 H) for steel chimneys

H = height of chimney, in m (ft.).

C. Height of the chimney. The height of the chimney is important only with natural draught (see first edition, p. 733). With mechanical draught, it is determined mainly by the necessity of dispersing the fly ash sufficiently far to avoid a nuisance to the factory and places nearby. Generally a height of 30-60 m (100-200 ft.) is adopted, or, better, 15-20 m (50-65 ft.) above the ridge of the factory roofs.

Mechanical draught systems

There are three principal systems of mechanical draught:

A. Forced draught. The air may be fed under pressure below the grate with step grates, or in the air distribution chamber of horseshoe or Ward furnaces or spreader-stokers.

This system has the advantage of producing atmospheric pressure, or slightly above, in the combustion chamber, and consequently avoiding any entry of air by leakage, in spite of cracks or lack of tightness in the brickwork.

Forced draught is used especially in the cases where an air-heater is installed, as well as in horseshoe and Ward furnaces, and with spreader-stokers.

B. Induced draught. This is the commonest system; the fan is placed above the flues, between the boiler and the chimney.

C. Ejector draught. This is another form of induced draught; the suction is produced by a steam nozzle discharging into the chimney and producing a Giffard effect on the gases, or preferably by a fan placed outside the flue circuit and producing the Giffard effect, by withdrawing part of the gases from the flue and returning them by a nozzle discharging into a section of the chimney of reduced diameter.

Pressure or suction required. Calculation of pressure and suction along the gas path is complex; it is a function of the rates of flow, cross-sections, obstructions, changes of direction, lengths of various sections of passage, etc. The following figures, for example, give data for a straight-tube boiler with economiser and air-heater, with forced and induced draught:

(a) *Forced draught.* The aim is to establish in the furnace a pressure of:

	30 mm	1¼ in.
--	-------	--------

to this must be added pressure drops in:

air duct:	15 mm	½ in.
changes of direction:	30 mm	1¼ in.
across air heater (air side):	20 mm	¾ in.
	—————	—————
	95 mm	3¾ in.

(b) *Induced draught.* At the top of the furnace there is required a suction of:

	3 mm	J in.
--	------	-------

pressure drops are:

across the boiler tubes:	35 mm	1⅜ in.
across the economiser:	8 mm	⅜ in.
across air-heater (gas side):	11 mm	½ in.
flues:	9 mm	⅜ in.
	—————	—————
	66 mm	2¾ in.

For these two duties fans will be installed for safety with a 20% margin, say 115 mm (4½ in.) for the forced draught and 80 mm (3½ in.) for induced draught.

In South Africa⁵⁴ it is recommended that, to obtain flexible operation of the boiler, a fan should be installed capable of furnishing, in continuous forced operation, margins appreciably greater than those indicated above:

(a) For forced draught: an air output of 115% against a pressure of 132%

(b) For induced draught: an air output of 120% against a pressure of 144% of the normal values. Logically, the draught should in fact vary as the square of the output (1.32 = 1.15²; 1.44 = 1.20²).

Power requirements

If a fan has to supply a volume of gas V m³/s (cu.ft./s) while maintaining a draught of d inches of water, the power used will be:

$$T = Vd \quad (T = 5.2 Vd) \quad (41.88)$$

T = nett power to be supplied.

The actual power required by the fan will then be:

$$\begin{aligned} T &= \frac{Vd}{e} \text{ kgm/s} && \left(5.2 \frac{Vd}{e} \text{ ft.lb./s} \right) \\ &= \frac{Vd}{75e} \text{ h.p.} && \left(\frac{Vd}{100e} \text{ h.p.} \right) \end{aligned} \quad (41.89)$$

e = efficiency of the fan.

This efficiency is very low, and varies substantially with the power of the fan. We may take:

Small fans 0.20-0.50, average: 0.30

Large fans 0.40-0.70, average: 0.50

It must not be forgotten that the power T is only the power supplied to the shaft of the fan. For the power supplied to the fan motor, it is necessary to allow for the efficiency of the motor, and also that of the belt, where belt drive is used.

Influence of gas temperature. It will be noted that the power T indicated above depends only on the volume V of gases handled. Consequently, for a given output in terms of weight of gas, more power is required when the gas is hot than when it is cold.

Practical formula for power. It is difficult to choose or to know *a priori* the efficiency of the fan, which enters into eqn. (41.89). Furthermore, a small variation would be enough to introduce a serious error, on account of the very low efficiencies under consideration.

Shillington⁵⁵ has published figures more particularly applicable to induced-draught fans of the Prat type, from which the following approximate formula for power may be derived:

$$T = \frac{Bd(273 + t)}{8,400} \quad \frac{Bd(273 + t)}{V \cdot 600} \quad (41.90)$$

T = power required at the fan shaft, in h.p.

B = quantity of bagasse consumed by the furnace or furnaces served by the fans, in tonnes/h

d = draught at suction of fan, in mm (in.) of water (+ or -)

t = gas temperature at suction of the fan, in °C (°F).

As previously, the weight of bagasse B may be replaced by the weight of steam p produced by the boiler, using the factor/given by Table 41.22 for steam produced per unit of bagasse.

Example. We require the power necessary to drive a general fan serving all the boilers in a factory crushing 100 t.c.h. and producing 280 kg (625 lb.) bagasse/t.c. but burning only 210 kg/t.c. (470 lb./t.c.)

Draught required: $d = 125$ mm (5 in.) at suction of fan.

Temperature of gases entering the fan: $t = 225^\circ\text{C}$ (440°F).

We have:

$$B = \frac{100 \times 210}{1,000} = 21 \text{ t/h} \quad (100 \times 470 = 47,000 \text{ lb./h})$$

and:

$$T = \frac{21 \times 125 (273 + 255)}{8,400} \left\{ \frac{21 \times 5 (460 + 440)}{600} \right\} = 157 \text{ h.p.}$$

which will require a motor of at least 175 h.p.

Shillington recommends installing a fan of ample capacity, since, if there is need to force the boilers, the fan efficiency falls very rapidly when normal output is exceeded, whereas it drops only slowly when output is decreased below that corresponding to the optimum efficiency. In other words, a large fan, having a large margin of capacity, is preferable to a smaller one working at full capacity.

Regulation of draught. To save power and especially to reduce wear of the fan blades, it is advisable to regulate the draught by adjusting not the damper but the speed of the motor or turbine driving the fan.

Abrasion. Induced-draught fans are subject to marked wear due to the abrasion produced by solid particles entrained with the flue gasses. These fans generally run at 700–800 r.p.m. It is advisable to use the lowest speed necessary, since the wear is proportional to the 2.5 power of the rotational speed. Replaceable fins may be attached to the fan blades; these are of very hard metal of high tensile strength and alloyed with tungsten carbide.

Fly ash removal

The high draught used in modern installations often leads to a high proportion of unburnt solid particles in the flue gases, which settle about the factory and its neighbourhood and generally come under the control of legal restrictions.

Three types of equipment are used for reduction of fly ash content of the flue gases:

(a) **Mechanical dust removal.** The equipment manufacturers supply fly ash removers for installation in the flues, the action being based on changes of direction or centrifugal force. Such devices are generally effective, sometimes to an efficiency above 96%, but they involve a substantial draught loss (700–800 Pa or 3 in. w.g.).

(b) **Electrostatic dust removers.** This method consists of attracting the solid particles by passing them close to the surface of electrodes between which a potential difference has been established. Power consumption is low, but the cost is high and the equipment rather cumbersome. Very high efficiencies may be obtained.

(c) **Removal by water.** Generally it is considered satisfactory in the sugar factory to pass the flue gases into a large chamber where their velocity is reduced and where they are subjected to a water spray. The wetted particles fall and are caught at the bottom in a stream of water. The effect may be accentuated by interposing baffles in the path of the gases. The efficiency of separation may reach 95%. The finer particles which remain are carried well away by the wind and the practical effect obtained is very superior to the theoretical efficiency. This system substantially reduces the flue gas temperature, which partially compensates for the excess power required by the fan.

Water requirements may be reckoned as 3 - 4 kg of water per kg of steam, say 1.6- 2 kg of water per Nm³ of gases. This water consumption may be reduced if a slightly lower efficiency of operation can be tolerated.

Pneumatic conveying of bagasse. While it is not intended to discuss this matter, it may be commented in passing that, to transport 1 kg of bagasse or bagacillo, approximately 2 kg of air must be used. If this bagasse is to be burnt at the delivery of the conveyor duct, it will thus arrive with a weight of conveying air equal to approximately half the air necessary for combustion.

BOILER FEED WATER

Returned condensates

The sugar factory has a large number of condensates available from the various heat exchangers: multiple effects, juice heaters, vacuum pans, etc.

These condensates are generally pure, since they have been boiled and condensed as distilled water. It is necessary, however, to classify them according to their origin:

(a) Water derived from condensation of the steam coming directly (heat exchangers heated by live steam) or indirectly (heat exchangers using exhaust steam) from the boilers.

(b) Condensates originating from vapour derived from the juice: condensate returned from the second and subsequent vessels of the multiple effect, from juice heaters or other heat exchangers heated by vapour bled from the multiple effects.

Condensates of the first type involve little risk of being contaminated. Since they originate from steam under pressure, even if a tube develops a leak in the first effect, for example, it will be steam that will leak into the juice, rather than the opposite. The greatest risk originates in the juice heaters, where the possibility of a split tube may admit juice under pressure into the steam space.

Condensates of the second type are more dangerous. They involve first the same risk of direct contamination by juice, aggravated by the fact that the juice vapours are most often under vacuum and consequently more subject to entry of juice under pressure through leaks.

However, the main risk is that these condensates may contain sugar originating from entrainment in the evaporators. Even if this is present only in imperceptible traces, these traces will be returned to the boilers and will end up by accumulating on the boiler tubes a harmful and dangerous carbonaceous deposit.

It is important then to separate the returned condensates carefully:

- (a) From direct or exhaust steam
- (b) From vapour derived from juice.

Utilisation of condensates. Condensates of the first group only should be sent to the feed-water tank.

Those of the second group may be utilised for imbibition, washing of filter cakes, dilution of molasses, etc. If there is an excess of them, it is preferable to discard the excess rather than send it to the boilers.

Make-up water

The steam in the sugar factory describes a closed circuit: evaporated in the boilers, condensed in calandrias, it returns to the boilers by way of the feed-water tank.

It would therefore be possible to use the steam in this closed cycle, without addition of water, if it were not for the following losses, which must be compensated:

(a) Steam lost to atmosphere:

Leakages at joints and stuffing-boxes
 Operation of safety valves
 Operation of soot blowers
 Engines exhausting to atmosphere: steam derrick, etc.
 Steaming out of filter presses
 Cleaning of mills and cush-cush plant with steam
 Steam used at centrifugals.

(b) Water lost to the drain:

Washing out of piping
 Blowing down of boilers.

(c) Steam partially lost in the molasses:

Washing out of pans
 Dilution of molasses.

The total of these losses of steam or condensates represents, according to circumstances, from 10 to 20% of the steam produced at the boilers.

In order to replace this quantity, it is necessary to return to the feed-water tank either:

- (a) Cold water, or
- (b) Part of the condensates of the second group above.

These condensates in effect originate from the juice, that is they enter into the factory with the cane. They thus consist of water from an outside source.

However, on account of the risks involved, their use is reduced to a minimum, and they

are taken from the point which presents least danger; that is the second vessel, the condensates from which present the minimum risk of containing sugar due to entrainment.

In order to keep the contribution of condensate from the second effect to the minimum strictly necessary, this condensate should be discharged into a tank alongside the feed-water tank and arranged so as to deliver into the latter only by a float-operated valve which opens when the level in the feed tank has dropped below a certain limit.

The addition of second-effect condensate should normally be sufficient to supply the make-up necessary.

Sundry precautions. To guard against entrainment of water in the steam, Fives Cail - Babcock boilers are equipped with cyclone separators in the steam which ensure the production of steam of high dryness fraction.

To allow for variations in the steam load of the factory, the regulation of the level in the feed water tank integrates the three factors: level of water in the supply tank, output of steam, and output of water.

The boiler may be furnished with a salinometer indicating any abnormal salt content in the boiler water; this figure should not exceed 1800–2200 mg/l, according to pressure.

The silica content of feed water should not exceed 75 p.p.m. for a pressure of 20 bar (290 p.s.i.g.), 40 p.p.m. for 35 bar, 30 p.p.m. for 45 bar or 20 p.p.m. for 50 bar.

De-aeration. For boilers at high pressure (above 35 kg/cm²; 500 p.s.i.), the feed water should be treated to remove dissolved gases, especially carbon dioxide and oxygen, which would attack the tubes at the high temperatures involved.

For this purpose, the water is passed into a de-aerator where it is atomised in steam under low pressure (0.3 kg/cm²) or at a pressure rarely above atmospheric, where the injected steam is raised to 1.4 kg/cm². It is then treated with a suitable reagent such as sodium sulphite activated to work more rapidly, or hydrozine aluminium sulphate or other suitable reagents.

The de-gasser serves at the same time as storage for de-aerated water. The steam occupying the free space in the tank prevents possible re-absorption of gases.

De-mineralisation. Unless the available make-up water is exceptionally pure, high-pressure boilers also require demineralisation of the feed water, *i.e.* salts of calcium, sodium, magnesium, etc., must be removed. This operation is carried out before the de-aerator in a double ion-exchanger, one for cations, the other for anions. It may also be effected more simply with sulphuric acid and soda.

Capacity of feed-water tank

To avoid the necessity of supplementing it with cold water, it is necessary to allow in the feed-water tank a reserve capacity of water sufficient to cope with the sudden demands of the boiling house.

Tromp⁵⁶ estimates that the fluctuations in instantaneous steam consumption in a sugar factory amount to 25% above and below the mean. These fluctuations generally do not last for more than one hour and, if the peak represents $\pm 25\%$ on normal consumption, the mean increase or decrease in consumption during the hour in which it occurs represents only

$\pm 15\%$ of the average consumption for the day or for the week. Assuming a high consumption of 650 kg steam/t.c., we see then that the difference between maximum consumption and minimum return of condensates represents approximately:

$$650 (1.15 - 0.85) \approx 200 \text{ kg water/t.c.}$$

However, the time elapsing between the steam leaving the boiler and the return of the corresponding condensate to the feed tank represents only $\frac{1}{4} - \frac{1}{2}$ h.

It will be seen then that it will be amply sufficient to provide a feed tank of capacity equal to 100 kg/t.c.h. (225 lb./t.c.h.), in order to cope with fluctuations in steam consumption without having recourse to an avoidable addition of cold water:

$$c = 100 A \quad (22A) \quad (41.91)$$

c = capacity of the feed-water tank, in litres (Imp. gallons)

A = crushing rate of the factory, in t.c.h.

A factory of 100 t.c.h. should therefore have a feed tank of 10 m³ (2,200 gallons) capacity.

We may comment in passing that this reserve capacity does not depend on the steam economy of the factory: it should be as large for a factory using 450 kg (1,000 lb.) steam/t.c. as for one using 650 kg/t.c. (1,450 lb./t.c.), since the variations in consumption, due predominantly to the pans, are the same in the two factories if they are expressed in weight of steam per tonne of cane.

It is of advantage to provide the feed tank with two float-operated valves: the first controlling the addition of water from the second vessel, the second the addition of cold water. It will then be necessary to allow a sufficient margin between the different levels, say approximately $\frac{1}{3}$ of the height of the tank:

(a) Between the overflow level and the opening of the float-operated valve controlling condensate from the second vessel

(b) Between the opening of this condensate valve and the point of opening of the float-operated valve for cold water

(c) Between the opening of the cold-water valve and the bottom of the tank.

Feed-water pumps

The law requires that there should be at least two feed-water pumps. Generally the pump in normal usage is turbine driven and the stand-by pump is motor driven.

At the delivery side of the pump, the stop valve should be supplemented by a non-return valve the purpose of which is to avoid the risk, when the pump is stopped, of the water running back from the boiler towards the feed tank if the delivery valve does not close properly.

The feed pump should be capable of an output of at least 25% greater than the requirement in continuous forced operation of the boilers which it supplies, made up as follows:

10% for emergency making up of water level

10% to allow for an exceptional load for one hour

5% to compensate for blowing down

It should be able to operate at a pressure at least 25% higher than the certified pressure of the boilers; we shall calculate this in detail. This pressure comprises:

(a) Certified pressure, for example:	30 kg/cm ²	425 p.s.i.
(b) An excess pressure to free the safety valve and which may be taken as about 3% of working pressure:	1 kg/cm ²	15 p.s.i.
(c) Head losses in the piping, to be calculated as in Chap. 45, for example:	1 kg/cm ²	15 p.s.i.
(d) Gravity head corresponding to height of the boiler above the pump, say 10 – 12 m (30 – 40 ft.):	1.2 kg/cm ²	15 p.s.i.
(e) Head loss in the economiser to be calculated, say:	1 kg/cm ²	15 p.s.i.
(f) Head loss in the regulated valve, of the order of:	2 kg/cm ²	25 p.s.i.
(g) A safety margin which is taken as about 5% of the total preceding:	1.9 kg/cm ²	30 p.s.i.
	38.1 kg/cm ²	540 p.s.i.

On account of the temperature (90 – 95°C; 195 – 205°F) of the feed water, it is desirable that the feed-water tank should be situated at least 1–2 m 3–6 ft. above the pump, preferably 2.5 m (8 ft.) (see Chap. 44).

Feed-water pipes. The diameter of pipes should be chosen so as to obtain the following velocities:

- (a) Suction pipe, from tank to pump: 1 m/s (3 ft./s)
- (b) Delivery pipe, from pump to boiler: 3 m/s (10 ft.s).

However, a velocity of 4.5 m/s (15 ft./s) is now used in modern high pressure boilers".

Influence of water temperature on fuel consumption

The last column of Table 41.21 shows that an increase of 6°C (11°F) in the feed-water temperature corresponds approximately to a saving in fuel of 1%.

There is another advantage in maintaining the feed water at a high temperature: the water dissolves correspondingly less oxygen when it is hotter. Thus the risk of corrosion is reduced.

Reaction of feed water

The corrosive properties of feed water depend on its pH. It is at a minimum, not for a pH of 7, but for a slightly alkaline value.

A certain alkalinity is maintained in the feed water to prevent possible corrosion, and this is done by introducing hydroxyl (OH) ions into the feed water.

When a pH of 9 is reached, a protective film of ferrous hydrate is formed on the metal. On the basis of this fact, Brola⁵⁸ recommends that a pH of 9.5 be maintained.

An English recommendation⁵⁹ is pH 8.3 as the lower limit, and it is stated that it is mainly in the economiser that corrosion is manifest.

Leggett⁶⁰, in India, recommends that a pH of 7.6 be maintained in the feed tank, which corresponds to 8.6-9.0 in the boilers.

Gregory⁶¹, in Cuba, recommends a pH of 10.5 — 11.

The present author would recommend maintaining a pH of 8.5 in the feed tank. For the water in the boiler, the minimal pH, measured at 25°C, should be 11.0.

Alkalisising agents. In order to introduce hydroxyl ions into the feed tank, we may employ:
Caustic soda, NaOH
Sodium carbonate, Na₂CO₃
trisodium phosphate.

Neumann⁶², in Java, recommends trisodium phosphate, which offers certain advantages over the others; but it is not suitable for high pH values, which would require excessive quantities. The boiler water should have a ratio Na₂PO₄/NaOH > 1.

STEAM ACCUMULATORS

The continual variations in momentary steam consumption, and the consequent fluctuations in steam pressure, have led certain factories to install accumulators. These are vessels serving to regulate the output of steam. Contrary to the impression which the name would give, it is not actually steam which they store, but water; but it is indeed steam which they deliver.

Principle

A steam accumulator is a pressure vessel, a large cylindrical drum resembling a boiler drum, filled with water, and communicating on one side with the boilers, on the other side with the low pressure or exhaust steam piping. The connections are made by means of valves and regulators located in such a way as to permit circulation of steam only in the sense which we shall describe.

Let:

P = normal boiler pressure

P' = pressure slightly lower than P , to which it is arranged that steam may drop in the boilers

p = normal pressure at which steam is used for manufacture; this is generally the exhaust steam pressure

p' = pressure slightly higher than p

p'' = pressure slightly lower than p .

The regulators and valves are arranged so that:

(a) When the h.p. steam is at a pressure between P and P' , the accumulator allows steam to enter from the boilers, but does not deliver any steam. It thus becomes charged.

(b) When the pressure falls below P' , the connection with the boilers is closed.

(c) When the exhaust steam pressure falls below p'' , the accumulator delivers steam to the low-pressure line, and thus discharges.

(d) When the pressure in the exhaust system rises above p' , the connection between it and the accumulator is closed.

The two pressures p' and p'' are chosen relative to p in such a way as to avoid too frequent opening and closing of the valves and to provide a margin of stable operation.

The principle of the accumulator is simple. It stores water at the temperature of saturated steam corresponding to the boiler pressure. If the boilers produce superheated steam, it will become saturated when it is forced into the water in the accumulator. In order to obtain an effective and silent mixing of the steam with the water, the steam is introduced into the vessel by conical or bell-shaped nozzles. When a drop in pressure occurs in the accumulator, a portion of the water evaporates practically instantaneously: it is the sensible heat of the whole mass of stored water which is transformed into latent heat and so furnishes the heat of vaporisation of the portion evaporated.

The accumulator should obviously be very carefully insulated.

Calculations for a steam accumulator

Let:

P = normal boiler pressure

P' = pressure slightly lower than P to which the boiler pressure is allowed to drop

p = back pressure, or exhaust pressure (in the calculation, we shall assume for simplification: $p = p' = p''$)

R' = latent heat of steam at pressure P'

r = latent heat of steam at pressure p

T' = temperature of saturated vapour at pressure P'

t = temperature of saturated vapour at pressure p

x = any temperature between T' and t

q = the quantity of steam furnished by one kg (lb.) of water, when its temperature drops from T' to t

Q = the quantity of steam furnished by the accumulator, when its temperature drops from T' to t

V = the volume of water contained by the accumulator

V = gross total interior volume of the accumulator.

When the accumulator delivers steam, and when the temperature of the water which it contains drops from x to $(x - dx)$, each pound of water will supply a quantity of steam dq such that:

$$dq (607 - 0.7x) = - 1.013 dx \quad (dq [1,093 - 0.7(x - 32)] = - 1.013 dx)$$

whence:

$$dq = \frac{- 1.013}{607 - 0.7x} dx \quad \left(dq = - \frac{1.013}{1,115 - 0.7x} \cdot dx \right)$$

For a temperature drop from T' to t , each kg (lb.) of water will give (metric units):

$$q = - 1.013 \int_{T'}^t \frac{dx}{607 - 0.7x} = \frac{1.013}{0.7} [\ln (607 - 0.7x)]_{T'}^t$$

or:

$$q = 1.45 \ln \frac{607 - 0.7t}{607 - 0.7T'} = 3.33 \log \frac{r}{R'} \quad (41.92)$$

However, during this time the accumulator has delivered Q kg(lb.) of water, and the total quantity of steam which it can furnish at the moment it passes through temperature x has a value:

$$dQ = V(1 - q) dq$$

whence:

$$Q = V \int_{r'} (1 - q) dq = V \left(q - \frac{q^2}{2} \right)$$

or:

$$Q = Vq \left(1 - \frac{q}{2} \right) \quad (41.93)$$

Numerical example. In a factory of 100 t.c.h., the steam consumption of which is 500 kg (1,120 lb.) per tonne cane, it is desired to allow for periods of half an hour during which steam consumption exceeds the mean consumption by 20% = 100 kg/t.c. (224 lb./t.c.).

We assume that the boilers are sufficient to supply the required steam for the remainder of the time, without fall in pressure.

Other conditions are as follows:

$$\begin{aligned} P &= 28 \text{ kg/cm}^2 \text{ (400 p.s.i.)} & P' &= 25 \text{ kg/cm}^2 \text{ (350 p.s.i.)} \\ & & T' &= 225^\circ\text{C (435}^\circ\text{F)} \\ & & R' &= 439 \text{ kcal/kg (791 B.T.U./lb.)} \end{aligned}$$

$$\begin{aligned} p &= 1 \text{ kg/cm}^2 \text{ (15 p.s.i.)} \\ t &= 120^\circ\text{C (250}^\circ\text{F)} \\ r &= 526 \text{ kcal/kg (945 B.T.U./lb.)} \end{aligned}$$

Solution. The mean steam consumption amounts to:

$$100 \times 500 = 50,000 \text{ kg/h (112,000 lb./h)}$$

and the excess expected during the half-hour of peak demand to:

$$100 \times 100 \times \frac{1}{2} = 5,000 \text{ kg (11,200 lb.)}$$

The quantity of steam which each kg (lb.) of water stored in the accumulator can furnish in dropping from T' to t will be:

$$q = 3.33 \log \frac{526}{439} = 0.2615$$

We require then an accumulator the contents of which, V , will be given by:

$$5,000 = V \times 0.2615 \left(1 - \frac{0.2615}{2} \right) = 0.227 V$$

whence:

$$V = \frac{5,000}{0.227} = 22,000 \text{ kg} \quad (50,000 \text{ lb.}) \text{ water}$$

At 225°C, the specific volume of water is 1.1991 l/kg, hence:

$$V = 22,000 \times 1.1991 = 26,380 \text{ l} \quad (960 \text{ cu.ft.})$$

The water contained by the accumulator represents only about 80% of its total volume, on account of the necessity of allowing a margin to avoid entrainment due to possible priming. The total volume of the vessel will then be:

$$V = \frac{26,380}{0.8} = 33,000 \text{ l} \quad (1,200 \text{ cu.ft.})$$

say a cylinder of approximately 2 m (7 ft.) in diameter and 10.5 m (31 ft.) in length designed for a pressure of 28 kg/cm² (400 p.s.i.).

STEAM-REDUCING VALVES

In a well-balanced sugar factory, the quantity of steam necessary for manufacture is much greater than that required for production of power. Since all the steam is generally produced at high pressure, there is then normally a large excess of high-pressure steam to be passed into the low-pressure system.

This operation is effected by means of pressure regulators, which are actuated by the low-pressure-steam system: as soon as the pressure, which is generally the exhaust or back pressure, falls below a fixed value, the regulator opens and admits steam (Fig. 41.28).

These regulators generally consist of a **diaphragm**, the underside of which is subjected to the pressure to be regulated. The pressure exerted by the steam on this diaphragm is balanced by an adjustable counterweight. When this counterweight overcomes the steam pressure, the lever descends under the action of the weight and thus actuates a needle valve. This admits water under pressure to an actuating piston, which controls the opening of the valve admitting high-pressure steam.

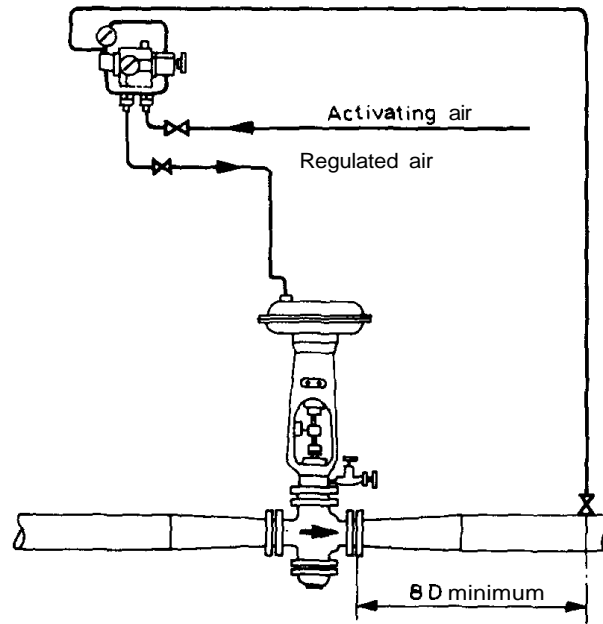


Fig. 41.28. Steam-pressure reducer.

In order that this valve shall work at a normal opening, it is essential that it should be of a diameter not only much smaller than that of the downstream pipe for the expanded vapour but also much smaller than that of the upstream high-pressure-steam line. Otherwise it would operate in an almost closed position, and would rapidly wear both the valve and the seat. Reducing valves are manufactured by SEREG-Schlumberger in France, and Maisoneilan, Fischer and Porter, and Honeywell in the U.S.A.

Temperature of the reduced steam

The final state of the reduced steam is given by the Mollier diagram (Fig. 41.5). The horizontal lines in this diagram correspond to changes without external work, at constant total heat, and this is the case with pressure regulators of reducing valve type.

If, for example, superheated steam at 25 kg/cm^2 (350 p.s.i.g.) and 370°C (700°F) is to be reduced to 1 kg/cm^2 (15 p.s.i.g.), the point is taken on the diagram corresponding to 26 kg/cm^2 (364.7 p.s.i.a.) and 370°C (700°F), and the horizontal line through this point will be followed to the isobar 2 kg/cm^2 (29.7 p.s.i.a.). The isotherm through this point is 355°C (670°F), the temperature sought for the expanded vapour.

DE-SUPERHEATING

The high-pressure steam is superheated in all electrified factories possessing one or more turbo-alternator sets.

We have just seen that expansion through a valve reduces the steam temperature only slightly, since the isothermal lines on the Mollier diagram are almost horizontal. The reduced steam is therefore superheated.

If the quantity of make-up steam necessary for manufacture is relatively small, this superheat will have no great disadvantage: it will serve mainly to reduce the moisture content of the exhaust steam, or to give it a slight superheat. We have already seen (p. 495) that this superheat presents no great disadvantage, and would even be advantageous, provided it does not exceed $10 - 30^{\circ}\text{C}$ ($50 - 90^{\circ}\text{F}$).

However, the make-up is generally much too great for the superheat to remain as low as this, and it is thus necessary to de-superheat the reduced steam. Hence the necessity, in this case, to follow the pressure regulator by a de-superheater (Fig. 41.29).

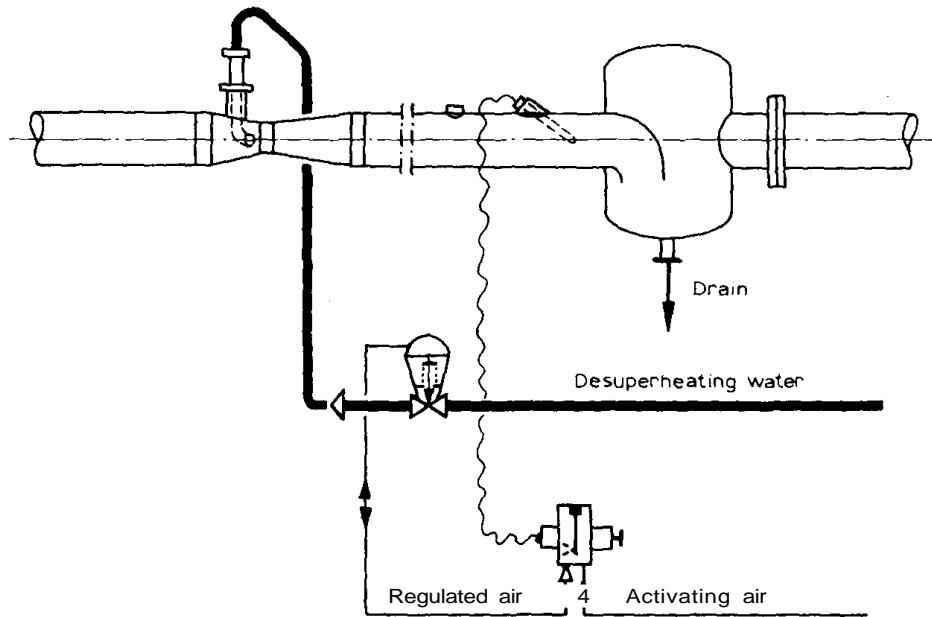


Fig. 41.29. Steam desuperheater.

Calculations for de-superheater

Suppose it is required to de-superheat the steam furnished by the regulator above, leaving only 50°C of superheat.

At the entry to the de-superheater, the steam at 1 kg/cm^2 and 355°C contains 760 kcal/kg according to the diagram or the table ($1 \text{ kg/cm}^2 \text{ gauge} = 2 \text{ kg/cm}^2 \text{ abs.}$).

At this pressure, the temperature of saturated steam is approximately 120°C . If we wish to leave 50°C of superheat, it will then be necessary to reduce the steam to 170°C .

The diagram shows that vapour at $2 \text{ kg/cm}^2 \text{ abs.}$ and 170°C contains 671 kcal/kg .

Per kg of vapour passing through the de-superheater, it is then necessary to remove:

$$760 - 671 = 89 \text{ kcal/kg} \quad (157 \text{ B.T.U./lb.})$$

For this purpose, we introduce into the de-superheater water from the boiler, which is at 25 kg/cm^2 and 225°C , and possesses a latent heat of vaporisation: $r = 439 \text{ kcal/kg}$. As it flashes into vapour at the reduced pressure, it will absorb heat.

It will be necessary then to introduce:

$$\frac{89}{439} = 0.203 \text{ kg water}$$

per kg of steam (0.203 lb./lb.) to be de-superheated.

If the quantity of vapour expanded per hour is 10,000 kg, this will require:

$$10,000 \times 0.2 = 2,000 \text{ kg water/h}$$

and we shall have 12,030 kg/h steam which is partially de-superheated.

The de-superheating water is finely atomised in the de-superheater, the successive opening of the several atomisers in the battery being actuated by a long thermostat element immersed in the outlet pipe carrying the de-superheated steam.

TYPES OF BOILER

Twenty years ago, there would still be found, in older sugar factories, two old types of steam boiler:

- (a) The "elephant" fire-tube boiler (Fig. 41.30)
- (b) The semi-tubular or multi-tubular boiler (Fig. 41.31).

These have no more than a historical interest.

Modern boilers are all of water-tube type and are divided into two principal types:

- (a) Boilers with straight tubes, set at a slight slope with headers or collectors (Fig. 41.26).
- (b) Boilers with bent tubes, vertical or at high slope, fixed to the boiler drums (Figs. 41.15 or 41.16); these boilers are sometimes fitted with water-wall furnaces.

A, Straight-tube boilers. These may have the tubes in square arrangement (*i.e.* in horizontal and vertical rows) or in staggered formation, the latter arrangement giving a better mixing of the combustion gases, better utilisation of the heating surface of the tubes, and less deposit

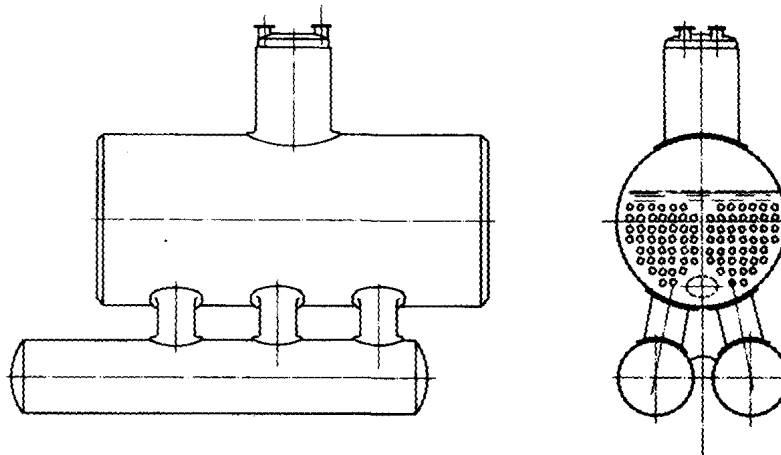


Fig. 41.30. "Elephant" boiler with fire-tubes.

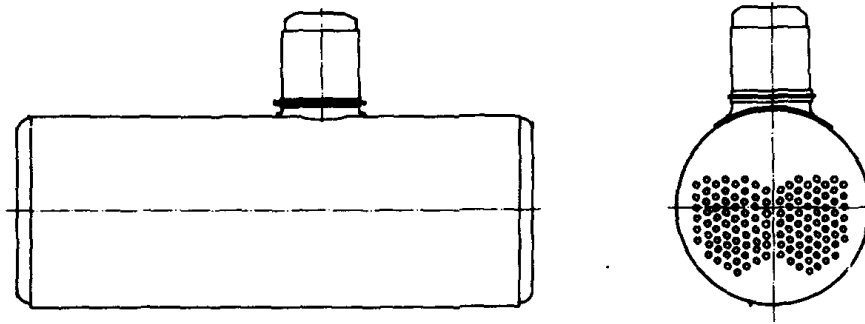


Fig. 41.31. Semi-tubular boiler.

of soot on the front and back faces of the tubes. Replacement of tubes is easy, but requires that space should be available in front of the boiler for a distance equal to the length of the tubes; this is not always easy, on account of the bagasse conveyor and distributor.

B. Boilers with bent tubes. This type promotes more rapid circulation of the water, on account of the steeper inclination of the tubes. Replacement of the tubes is more difficult; to insert a tube of the back row, it will be necessary either that the pitch of the tubes is appreciably greater than twice their diameter, or that they should be arranged with a spacing alternately wide and narrow.

Calculation of heating surfaces of the tube assemblies is made in the same way as indicated for superheaters (*cf.* p. 961), the heat being transmitted being that necessary to raise the sensible heat of the water from the temperature at the economiser outlet to the saturation temperature, and to vaporise this water at that temperature to the dryness fraction required for entry to the superheater.

The heat transfer coefficients of the two types of boiler do not differ greatly. They range between 30 and 50 kcal/m²/h/°C (6 – 10 B.T.U./sq.ft./h/°F).

Tubes

Boiler tubes are made in diameters of 2, 2 $\frac{1}{4}$, 2 $\frac{1}{2}$, 2 $\frac{3}{4}$, 3, 3 $\frac{1}{4}$, 3 $\frac{1}{2}$, 3 $\frac{3}{4}$ and 4 in., mainly of 2 $\frac{1}{2}$ and 3 in. Bent tubes are often made in diameters smaller than those for straight tubes. The diameter has some importance, as it is possible to locate in the same general layout of boiler, and in the same space, a greater heating surface with small tubes than with large tubes. Fives Cail-Babcock, for example, make models BR1 and BR3 with tubes of 2 $\frac{1}{2}$ in. exterior diameter. However, if they are specifying a boiler without economiser, they propose 2-in. tubes. This permits of placing in the same available space a tube assembly with about 20% greater heating surface, which reduces the flue gas temperature by about 20°C and permits of better results from the boiler without economiser.

Tubes are generally made of A37 steel.

Series of boiler sizes

The following example gives a series of boilers supplied by one organisation.

Fives Cail - Babcock boilers are normally made, for use with bagasse, up to 50 bar. There are 4 types:

BC1 with horseshoe furnace (Fig. 41.20)

BR1 and BR2B with spreader-stokers and rocker-type grate (Fig. 41.15)

BR3 with spreader-stoker and travelling grate (Fig. 41.32)

BR22 and 22D with spreader-stoker and rocking grate.

In the model number the first number, n , indicates the number of screen tubes, seen from the front; the second the number of boiler tubes proper, also viewed from the front.

The lengths and widths refer to the boiler itself without economiser or other accessory equipment. Heights are measured from the floor of the boiler room. These boilers all have the same longitudinal section. They consist of rows of tubes side by side. Each row occupies 18.5 cm (7.28 in.), and it follows that the overall width L of the boiler is:

$$L = 0.185n + 1.25 \text{ m} \quad (L = 7.28n + 50 \text{ in.})$$

For the BC1 type, the capacity may be estimated as close to one tonne of steam per row.

$$Q = n$$

Q = evaporation capacity of the boiler, in t/h of steam

n = number of screen tubes as seen from the front.

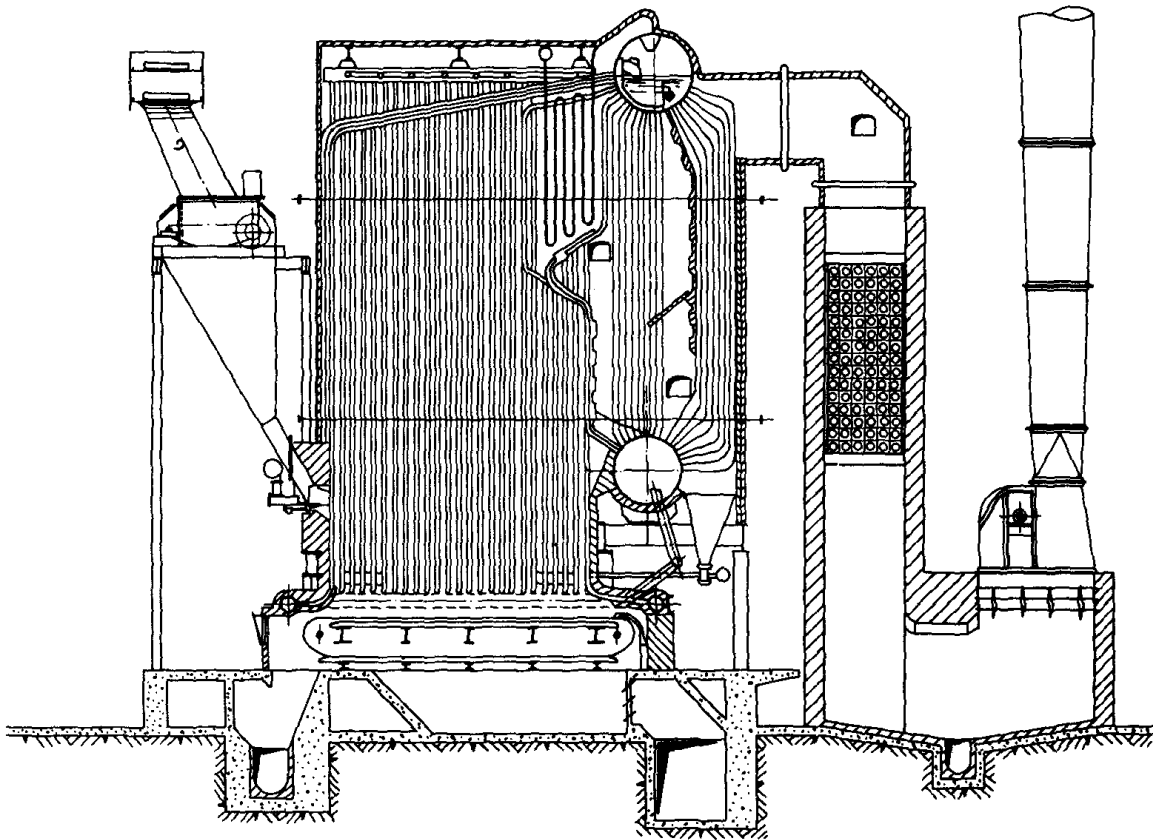


Fig. 41.32. Boiler with spreader-stoker and travelling grate (Fives Cail – Babcock type BR3).

For the BR1 type $Q = 1.25n$, and the BR3B type, about $1.36n$.

For the BR3 type, $Q = 1.65 - 1.90n$.

Table 41.28 gives an extract from the manufacturer's table and includes nearly all the values of n , from 15 to 73.

TABLE 41.28

SPREADER-STOKER BOILERS (FIVES CAIL - BABCOCK)

<i>Model</i>	<i>Width (m)</i>	<i>Steam production (t/h)</i>				
		<i>BR1</i>	<i>BR2</i>	<i>BR3</i>	<i>BR22</i>	<i>BR22D</i>
15/25	4.04	18				
20/32	4.96	26				
23/37	5.52	29				
26/42	6.07			44		
27/43	6.27	34				
30/48	6.81			50		
31/50	7.00	39	48			
33/53	7.37		50	55		
36/54	7.92					89
34/55	7.55	42				
36/58	7.92		55	60		
39/62	8.48		60	66		
40/60	8.67					98
43/68	9.22		65	71		
44/66	9.40					108
46/73	9.77		70	81		
49/73	10.33					122
49/78	10.33		75	91		
52/83	10.88				88	
53/84	11.07		80	100		
58/92	11.99				96	
59/88	12.18					137
64/101	13.10				107	
71/112	14.40				120	
Special				100	≥ 160	≥ 160
Length (m)		7.00	7.70	8.06	8.00	10.10
Height (m)		10.40	11.25	11.80	12.40	14.00

CONSTRUCTION OF BOILERS

Joints in boiler drums

Tromp⁶³ rightly draws attention to the comparison to be made between the longitudinal and transverse stresses involved in the plates and joints of boiler drums under pressure.

(a) **Circumferential joints.** Consider a vertical section of the drum (Fig. 41.33). The section is subject to a bursting force of:

$$F_c = \pi R^2 p$$

F_c = bursting force, in kg (lb.)

R = radius of cylinder, in cm (in.)

p = pressure inside the drum, in kg/cm² (p.s.i.)

This force acts on an area S_c of the metal concerned and the stress in the latter is:

$$f_c = \frac{2F_c}{S_c} = \frac{2\pi R^2 p}{2\pi R e} = \frac{Rp}{e} \quad (41.94)$$

e = thickness of the metal, in cm (in.)



Fig. 41.33. Circumferential joints.

(b) Longitudinal joints. Imagine the drum cut by an axial plane (Fig. 41.34) The section is then subjected to a force:

$$F_l = 2RLp$$

F_l = force acting on a longitudinal section, in kg (lb.)

L = length of the cylinder, in mm (in.)

The longitudinal stress is then:

$$f_l = \frac{2F_l}{S_l} = \frac{4RLp}{2Le} = \frac{2Rp}{e} \quad (41.95)$$

neglecting the additional strength due to the tube plates (or the ends), which is negligible in the median portion of the cylinder.

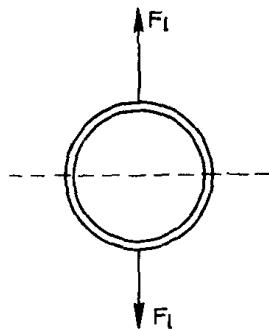


Fig. 41.34. Longitudinal joints.

Conclusion. We see then that the stress acting on the longitudinal joints is double that on the transverse joints:

$$f_l = 2f_c \quad (41.96)$$

The boiler plates should thus be calculated as a function of f_l . In the period when joints were riveted, this was the reason why longitudinal joints had a double row of rivets whereas the circumferential joints had a single row.

Calculation of plate thickness for steam drums

Following on the preceding calculations, the thickness of plate to be used for steam drums may be calculated (French code AFNOR).

(7) Cylindrical portion. We have:

$$e = \frac{pD}{2fz + p} \quad (41.97)$$

e = thickness of plate for cylindrical portion, in mm (in.)

p = certified pressure, in bars (p.s.i.g.)

D = external diameter of drum, in mm (in.)

z = coefficient allowing for the welding of the cylinder (0.7 or 0.85)

f = limit applying at temperature involved, with safety factor of 1.5, in daN/mm².

(2) Thickness of ends. If the end is of basket-handle shape (Fig. 41.35):

$$e = \frac{pDC}{2f} + c \quad (41.98)$$

e = thickness of plate for ends of drum, in mm (in.)

p = pressure, in bars (p.s.i.g.)

C = shape coefficient = $0.25 D/H$ (H = bulge of dished end)

c = additive term for thickness, allowing for possible corrosion

f = limit applying at the temperature involved, in daN/mm².

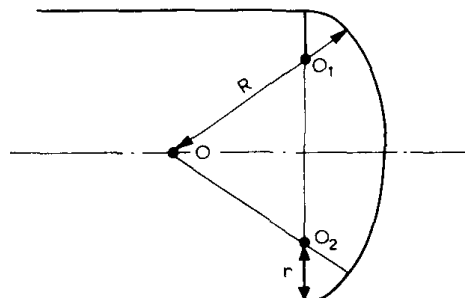


Fig. 41.35. Dished end of boiler drum.

MAINTENANCE OF BOILERS

Slack season. The boilers often deteriorate as much during the slack season as during the crushing season.

To avoid this it is necessary, at the end of the season, to empty the boilers carefully, and to check that no pockets of water remain. The interior will be painted with a rust-preventive paint (or given a coat of used oil), and the boilers will be closed up again on a very dry day, after having introduced a tray of quicklime or calcium carbide, which will absorb the remaining humidity.

Do not forget to remove the tray before the next crushing season!

Inspection. Check whether the tubes connecting headers and drums (straight-tube boilers) or the ordinary tubes (bent-tube boilers) are pitted. If pits are found, and if they are deep, the tubes should be replaced.

These pits are due to attack of the metal by dissolved oxygen, and are particularly to be feared with acid water. Hence, in this case, check the pH of the water.

Check whether the refractory baffles separating the gas passes are in good condition. A short circuit in the path of the hot gases can diminish the efficiency very markedly.

Soot blowers. Before using the soot blowers it should be checked that the piping to the blowers has been well drained. Otherwise, the water blown among the tubes will provoke rapid oxidation in all parts which it reaches.

Cost. The cost of boilers increases approximately proportional to the square root of capacity in t/h of steam.

STEAM BALANCE

There is frequently need to establish the steam balance of the factory. We shall proceed to establish one but, since, if we sought to work out general formulae, we should end up with very long and complicated equations, we propose to take a concrete case and reduce all figures to a crushing rate of 1 tonne per hour. It will be easy to repeat the calculation in any practical case whatever which may arise, by replacing the values adopted by the appropriate figures, and calculating the quantities of steam for the hourly rate concerned.

Calculation of steam balance

We shall assume the following conditions:

Crushing rate of the factory

100 t.c.h.

(a) Steam production

Fibre in cane:	14%
Weight of bagasse:	285 kg/t.c. (640 lb./t.c.)
Moisture in bagasse:	48%
Pol in bagasse:	2%
Excess air in flue gases:	40%
Temperature of gases at entry to the chimney:	200°C (400°F)
Boiler pressure:	28 kg/cm ² g (400 p.s.i.g.)
Temperature of the superheated live steam:	380°C (700°F)
Temperature of water in feed tank:	90°C (194°F)
Coefficients characterising the efficiency of the combustion (<i>cf.</i> p. 937):	$a = 0.97$
	$O = 0.97$
	$\eta = 0.97$

Factory completely electrified.

(b) Steam consumption

Quadruple effect evaporation	
Weight of mixed juice per tonne of cane:	1,000 kg (2,240 lb.)
Brix of clarified juice:	13
Brix of syrup:	65
Temperature of primary juice after mixing:	122°F
Temperature of secondary juice after mixing:	122°F
Vapour bled from third effect:	$P_3 = 52$ kg/t.c. (115 lb./t.c.)
Vapour bled from second effect:	$P_2 = 44$ kg/t.c. (100 lb./t.c.)
Vapour bled from first effect:	$P_1 = 28$ kg/t.c. (63 lb./t.c.)
Temperature of primary and secondary juices at the outlet from vapour heating:	95°C (200°F)
Temperature of heating of primary and secondary juices by exhaust steam:	103°C (218°F)
Back-pressure:	1 kg/cm ² g (15 p.s.i.g.)
Vacuum:	65 cm (26 in.)

Input. The N.C.V. of the bagasse will be (eqn. 41.25):

$$\text{N.C.V.} = 4,250 - (4,850 \times 0.48) - (1,200 \times 0.02) = 1,900 \text{ kcal/kg (3,416 B.T.U./lb.)}$$

The loss of sensible heat in the flue gases will be (eqn. 41.48):

$$q = [(1 - 0.48) (1.4 \times 1.4 - 0.13) + 0.5] (200) = 290 \text{ kcal/kg (534 B.T.U./lb.)}$$

The quantity of heat recovered in the steam will be (eqn. 41.50):

$$M_v = (1,900 - 290) 0.97 \times 0.97 \times 0.97 = 1,470 \text{ kcal/kg (2,630 B.T.U./lb.)}$$

We may comment in passing that:

$$\frac{M_v}{\text{N.C.V.}} = \frac{1,470}{1,900} = 77\%$$

Under favourable conditions, a figure of 80% and even 82% may be attained⁶⁴.

Now, evaporation of 1 kg of steam will require, at 29 kg/cm² abs. and 38°C (Tables 41.3 and 41.1):

$$761 - 90 = 671 \text{ kcal/kg} \quad (1,199 \text{ B.T.U./lb.})$$

and 1 kg of bagasse will thus give:

$$\frac{1,470}{671} = 2.19 \text{ kg steam at } 28 \text{ kg/cm}^2 \text{ and } 380^\circ\text{C}$$

and a tonne of cane can supply, if necessary:

$$285 \times 2.19 = 624 \text{ kg (1,402 lb.) live steam}$$

Output. We shall first calculate the steam consumption of the prime movers, and then the steam consumed in manufacture.

Prime movers. Power requirements for the various sections of the factory are estimated as follows:

(a) Cane handling. this includes power for unloading cane from the delivery trucks, transfer to cane carrier by cranes, lateral tables, etc. Power requirements can be very variable, for instance, negligible in Australia, but high in countries where the fields are stony and where cane is stacked in a pile loaded by various devices on to the carrier. Power required may vary from 0.1 to 1.5 kW/t.c.h. Say 0.5:

	<i>Electric motor drive (kW)</i>	<i>Turbine drive (kW)</i>
(a) Cane handling:	0.50	
(b) Cane carrier:	0.15	
(c) 1st knife-set: varies from 3 to 15, say averaging 6 kW/t.f.h.:	0.85	
(d) 2nd knife-set: varies from 10 to 70, say averaging 15 kW/t.f.h.:	2.10	
(e) 3rd knife-set: now seldom used:	—	
(f) Shredder: varies from 10 to 50 kW/t.f.h., say 20 kW/t.f.h.:		2.80
(g) Mills: varies from 10 to 20 kW/t.f.h. and per mill, say 15 kW/t.f.h., assuming 5 mills, turbine driven:		10.50
(h) Other machines in the factory driven by electric motors: 10-20 kW/t.c.h., say 15:	15.00	
	18.60	13.30

We assume the steam consumption of the double-wheeled turbines for shredder and mill drive to be 13.5 kg/kWh (30 lb./kWh) and that of the turbo-alternator at the Power Station

to be 11 kg/kWh (24.2 lb./kWh). Allowing for the efficiency of the turbo-alternators, the power consumption for the electric motor drives listed above becomes:

$$\frac{18.60}{0.85 \times 0.97} = 22.50 \text{ kW/t.c.}; \text{ at } 11 \text{ kg/kWh steam consumption}$$

Steam requirement is:	247.5 kg/t.c.
For turbines fed directly: 13.30 x 13.5 =	179.5 kg/t.c.
	427 kg/t.c.

This steam is recovered as exhaust, and allowing for losses in turbines and piping, of about 5%, we have:

$$427 \times 0.95 = 406 \text{ kg/t.c.h. in exhaust steam}$$

Manufacture. (a) Vapour bleeding. We assume that the vapour bleeding at the quadruple effect for juice heating (*cf.* p. 589), amounts to 60, 35 and 32 kg/t.c.h. for P_3 , P_2 and P_1 , respectively.

(b) We assume that the juice after heating by first effect vapour is heated to 112°C (218°F) by exhaust steam and that the steam is at 1 kg/cm² = 120°C (223°F). The heating before evaporation thus requires:

$$\frac{1,000 \times 0.9 (112 - 103)}{525 \times 0.95} = 16 \text{ kg of exhaust steam (36 lb) per t.c.h.}$$

The quantity of water to be evaporated from the juice amounts to:

$$E = 1,000 \left(1 - \frac{13}{65} \right) = 800 \text{ kg/t.c.}$$

(c) Vacuum Pans. Assuming that the A pans are discontinuous, and B and C are continuous pans, the steam consumption for the pans will be (*cf.* p. 666),

A m.c.: 140 X 1.47 x 0.275 x 1.5 = 85 kg/t.c.h.	}	
B m.c.: 60 X 1.47 X 0.300 x 1.2 = 32 kg/t.c.h.		142 kg/t.c.h.
C m.c.: 40 X 1.47 x 0.360 x 1.2 = 25 kg/t.c.h.	}	

which we suppose shared in the ratio of: 98 kg of vapour from 1st vessel, 44 from exhaust steam.

We thus have the total vapour bleeding:

3rd vessel:	60
2nd vessel:	35
1st vessel: 32 + 98 =	130

The distribution of evaporation between the vessels will be, designating by X that from the last:

X			
$X + 60$	-----	$\frac{800 - 380}{4} X =$	No. 4 = 105
$X + 60 + 35$		105 kg/h, and evaporations	No. 3 = 165
$X + 60 + 35 + 130$		from the 4 vessels:	No. 2 = 200
$4X + 180 + 70 + 130 = 800$			No. 1 = 330
			800

The steam consumption in the manufacture is thus:

Heating before evaporators:	16		
Evaporation:	330	}	390 kg/h (874 lb/h)
Pans:	44		

To this there should be added steam required for cleaning, steaming-out pans, soot blowing, and loss in leakages, condensation or in blow-off at safety valves, etc. This may be estimated at 60 kg/h, giving a total of $390 + 60 = 450$ kg/h. We already have 406 kg/h from the turbines, leaving a further quantity of 44 kg/h required. The quantity of live steam required will thus be $427 + 44 = 471$ kg/h (1,059 lb./h.). The factory would thus have an excess of bagasse of:

$$\frac{624 - 471}{624} = 24.5\%$$

We shall see that it is possible to do much better.

MAXIMUM ECONOMY - POWER FOR EXTERNAL USES

The financial conditions in which sugar factories operate are sometimes difficult. To ensure stability, the sugar factory should seek to capitalise on products other than sugar and molasses. Now it has one product which it utilises very incompletely: the bagasse. Bagasse can serve as raw material for fibre board or paper pulp. Unfortunately, the economic conditions for such manufacture are often unfavourable. On the other hand, one use for excess bagasse is generally profitable, particularly since the crisis in petroleum supplies; that is, its conversion to electric energy for sale to the public network.

There are also factories which grow their own cane in dry, tropical areas and which have high need for irrigation for the cane in addition to the requirements of the factory. They may find it profitable to produce their own power for pumping, rather than purchasing power.

To consider a definite case, we assume sale to the public network. This solution assumes agreement and cooperation with the public power supply authority. To be truly profitable, it requires on the part of the factory: (a) maximal economy of bagasse; (b) installation of

a power plant for steam generation from the excess bagasse and production of electricity (this should be of substantial size and high efficiency). This equipment could be common to the requirements of both the sugar factory and the external supply; (c) dovetailing of this power plant with that of the factory, in such a way that the requirements of the two systems do not interfere with each other, and so that both obtain the maximum benefit of the potential of the system.

If the necessary financial arrangements are satisfactory, the general solution is straightforward. For the best conditions, we assume the case of a new factory. For an existing factory it will not be difficult to approach the solution which will be proposed for a new factory.

The maximal economy in bagasse will be obtained by the following means:

(a) Adoption of a very high pressure at the boilers: 45, 50, 55 or 60 atm; a value of 50 - 55 is generally sufficient.

(b) Adoption of a maximal exhaust-steam temperature, that is, just below the safety limit for the juice (*cf.* p. 506). We select 130°C (266°F), or 1.75 kg/cm² (25 p.s.i.g.) so as to ensure that the juice does not exceed 127 - 128°C (261 - 262°F).

(c) This temperature suggests the use of downward (falling-film) circulation for the first evaporator vessel, so that the juice passes the zone very rapidly at this temperature.

(d) We make the maximum possible use of vapour bleeding, from the lowest possible vessels of the set, in order to profit by the resulting economies.

(e) We employ continuous vacuum pans, including those for the A m.c., since continuous pans accept the lowest pressure of heating vapours (p. 716) and consume less vapour if live steam agitation is not used. These continuous pans contribute to regular steam demand at the boiling house, and this regularity becomes highly desirable with power supply to outside uses (Fig. 41.36).

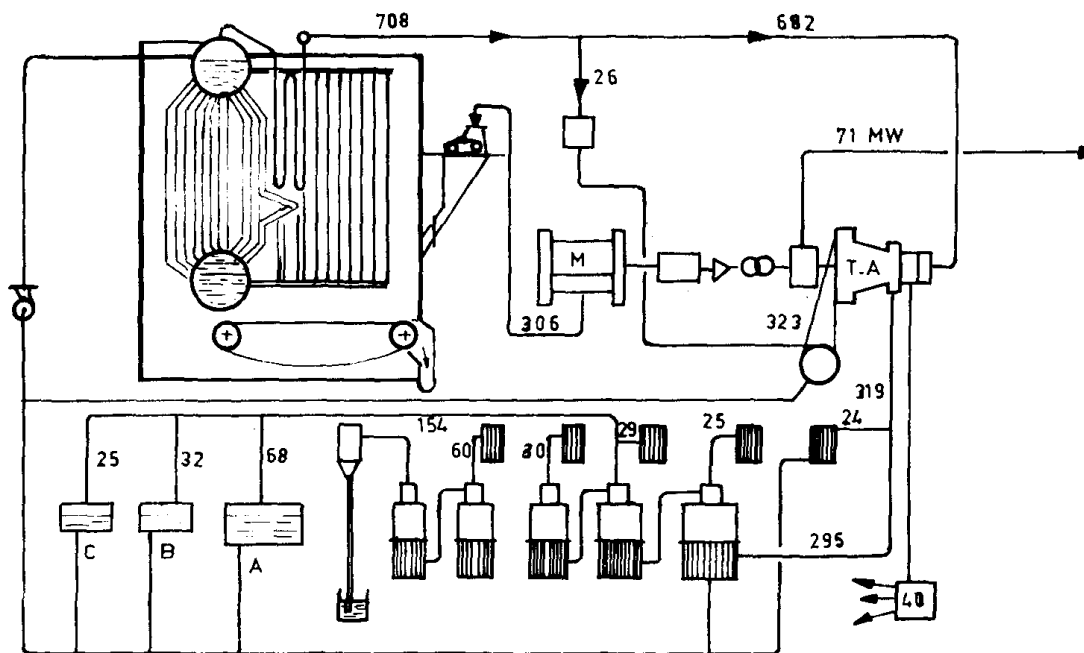


Fig. 41.36. Maximum economy. Diagram.

(f) We provide for a pass-out turbine at the exhaust-steam pressure, in order to allow maximum flexibility for production of process steam for the factory and power for the supply network. The use of a back-pressure turbine is not desirable in this case, though it involves less investment expenditure; an existing factory could retain its back-pressure turbines for use in case of breakdowns. For a new factory in fact, it may be advisable to have two groups of turbo-alternators to avoid trouble in case of an accident to one of the groups in service.

We consider as an example the following case.

Maximum economy for sale of electric power

Factory of 250 t.c.h.

Fibre in cane = 0.15

Weight of bagasse = 300 kg/t.c. (672 lb./t.c.) at 48% moisture and 1.8% sugar

Boiler of 180 t/h of steam at 55 atm with economiser and air heater

Superheat to 440°C (824°F)

Excess air 36%

Efficiency coefficient for combustion: $\alpha\beta\eta = 0.96$

Flue gases leaving the boiler unit = 156°C (313°F)

Temperature of feed water = 105°C (221°F)

Back pressure = 1.75 kg/cm² (25 p.s.i.g.) = 130°C (266°F)

Turbo-alternator of 30 MW, admission 45 atm (46 kg/cm² = 654 p.s.i.g.) 435°C (815°F)

Passout steam at 5 kg/cm² (71 p.s.i.g.) and 1.75 kg/cm² (25 p.s.i.g.)

Tandem of four mills with pressure feeders, 1067 X 2134 mm = 42" x 84"

Quintuple effect with the following scale of temperatures and vapour bleeding:

	Steam		Latent heat		Juice temp.		Vapour bleeding	
	T (°C)	t (°F)	(kcal/kg)	(B.T.U./lb.)	t (°C)	t (°F)	(kg/t.c.h.)	(lb./t.c.h.)
Ex. steam	130	266	519	934	125	257	24	54
1st effect	122	252	524	943	112	234	25	56
2nd effect	113	235	530	954	98	208	29	65
3rd effect	102	216	538	968	82	180	30	67
4th effect	87	189	547	985	65	149	60	134
5th effect	55	131	—	—	—	—	—	—

Brix of clear juice = 14

Brix of syrup = 68

Massecoites in continuous pans: A = 140 l/t.c. = 5.02 cu.ft./t.c.
 B = 60 l/t.c. = 2.15 cu.ft./t.c.
 C = 40 l/t.c. = 1.44 cu.ft./t.c.

Calculations are made in metric units; these are readily transposed into British units with the aid of the values above.

Heat input. The N.C.V. of the bagasse has a value (eqn. 41.25):

$$\text{N.C.V.} = 4250 - (4850 \times 0.48) - (1200 \times 0.018) = 1900 \text{ kcal/kg}$$

The loss of sensible heat in the flue gasses is (eqn. 41.48):

$$q = ((1 - 0.48) (1.4 \times 1.36 - 0.13) + 0.5)156 = 222 \text{ kcal/kg of bagasse}$$

The quantity of heat transferred to the steam will be:

$$M_v = (1900 - 222) 0.96 = 1611 \text{ kcal/kg of bagasse}$$

The steam obtained by combustion of 1 kg of bagasse will be (Tables 41.3 and 41.1):

$$\frac{1611}{787.6 - 105.1} = 2.36 \text{ kg of steam at } 46 \text{ kg/cm}^2 \text{ abs. and } 440^\circ\text{C}$$

and 1 tonne of cane will furnish: $300 \times 2.36 = 708 \text{ kg of live steam.}$

Steam usage. (I) Prime movers. Requirements are approximately:

	<i>Electric drive</i>	<i>Turbine drive</i>
(a) Cane unloading and feed to cane carrier: 0.5 kW/t.c.h.:	125	
(b) Cane carrier and magnetic separator: 0.4 kW/t.c.h.:	100	
(c) 1st knife-set: 3-8 kW/t.f.h., say 4: 4 x 37.5:	150	
(d) 2nd knife-set: 5-15 kW/t.f.h., say 4: 15 x 37.5:	560	
(e) 3rd knife-set: not used:	-	
(f) Shredder: 30-40 kW/t.f.h., say 32 x 37.5:		1200
(g) Mills: 10-20 kW/t.f.h. per mill: 4 x 18 x 37.5:	2700	
(h) Other machines: 10-20 kW/t.c.h., say 12 x 250:	3000	
	6635	1200

The shredder will be driven by a condensing turbine of nine stages; steam consumption for this will be 5.5 kg/kWh, or 6600 kg/h = 26 kg/t.c.h.

The main turbine for the power station is of passout type, with 5, 6 or 8 high-pressure stages and 4 low-pressure stages. The condensor pressure is 0.09 atm abs., its steam consumption is 5 kg/kWh condensing and 9 on passout operation; the latter figure is obtained after deducting power absorbed by H.P. auxiliaries of the boiler not included under "other machines" above.

Allowing for a mean efficiency of electric motors of 0.85 and efficiency of transmission lines of 0.97, the power consumption of the turbo-alternator at the Power House is:

$$\frac{6,635}{0.85 \times 0.97} = 8,050 \text{ kW} = 32 \text{ kW/t.c.h.}$$

(2) *Manufacture.* Clarified juice is heated from 112 to 125°C before the first vessel; for this heating, the steam required will be:

$$P = \frac{1000 \times 0.9}{0.95} \times \frac{125 - 112}{519} = 24 \text{ kg/t.c.h.}$$

The water to be evaporated in quintuple effect is:

$$1,000 \left(1 - \frac{68}{68} \right) = 794 \text{ kg/t.c.h.}$$

Steam consumption for the pans, all continuous, will be (kg/t.c.h.):

$$\begin{array}{l} \text{A m.c.: } 140 \times 1.47 \times 0.275 \times 1.2 = 68 \\ \text{B m.c.: } 60 \times 1.47 \times 0.300 \times 1.2 = 32 \\ \text{C m.c.: } 40 \times 1.47 \times 0.360 \times 1.2 = 25 \end{array} \quad \left. \vphantom{\begin{array}{l} \text{A m.c.:} \\ \text{B m.c.:} \\ \text{C m.c.:} \end{array}} \right\} 125$$

which will be taken from the second effect.

Hence the total vapour bleeding, for juice heating and pans, will be (kg/t.c.h.):

$$\begin{array}{r} \text{4th effect:} \quad 60 \\ \text{3rd effect:} \quad 30 \\ \text{2nd effect: } 29 + 125 = 154 \\ \text{1st effect:} \quad 25 \end{array}$$

and the evaporation will be:

$$\begin{array}{r} X \\ X + 60 \\ X + 60 + 30 \\ X + 60 + 30 + 154 \\ X + 60 + 30 + 154 + 25 \\ \hline 5X + 240 + 90 + 308 + 25 = 794 \end{array} \quad X = \frac{794 - 663}{5} = 26.2$$

Hence the evaporation for the 5 effects (kg/t.c.h.):

$$\begin{array}{r} \text{5th effect} = \quad \quad \quad = 26.2 \\ \text{4th effect} = 26.2 + 60 = 86.2 \\ \text{3rd effect} = 86.2 + 30 = 116.2 \\ \text{2nd effect} = 116.2 + 154 = 270.2 \\ \text{1st effect} = 270.2 + 25 = 295.2 \end{array}$$

794

The steam consumption for manufacture is thus (kg/t.c.h.):

Juice heating:	24
Evaporation:	295
	319

In addition to juice heating and evaporation, the factory requires an additional 40 kg of medium-pressure steam, for cleaning, for steaming out, soot blowers, etc. This steam is taken from the first stage of the turbine, at 5 kg/cm² or between 5 and 6 kg/cm² (70 – 85 p.s.i.g.), according to the stage which can best supply it. The steam consumption of the turbine at this stage is approximately 11 kg/kWh.

Recapitulating, we have the following steam consumption figures:

(a) The shredder, driven by its own turbine, consuming 5.5 kg/kWh or 1200 x 5.5: 6600 kg/h = 26 kg/t.c.h.

(b) The M.P. vapour from 1st. pass-out: 40 x 250 = 10000 kg/h = 40 kg/t.c.h.

(c) Heating and evaporation from 2nd pass-out stage, or: 319 x 250 = 79750 kg/h = 319 kg/t.c.h.

If we supply all the bagasse to the boilers, and pass out the foregoing quantities, we shall have:

H.P. steam; 300 X 250 X 2.36:	177,000 kg/h = 708 kg/t.c.h.
of which the shredder takes directly:	6,600 kg/h = 26 kg/t.c.h.
	170,400 kg/h = 682 kg/t.c.h.
There remains for the turbine:	170,400 kg/h = 682 kg/t.c.h.
The 1st. pass-out station bleeds off:	10,000 kg/h = 40 kg/t.c.h.
	160,400 kg/h = 642 kg/t.c.h.
There remains:	160,400 kg/h = 642 kg/t.c.h.
The 2nd. pass-out takes:	79,750 kg/h = 319 kg/t.c.h.
	80,650 kg/h = 323 kg/t.c.h.
There remains for sale outside:	80,650 kg/h = 323 kg/t.c.h.

At full load, the power consumption will thus be (kW):

Shredder:	26/ 5.5	4.7
1st pass-out:	40/11	3.6
2nd pass-out:	319/ 9	35.4
To condensor:	323/ 5	64.6

$$108.3 \text{ kW/p.t.c.} \times 250 = 27,075 \text{ kW}$$

Of which the factory requires for its own uses: 9,250

Leaving for external uses: 17,825

We have not sought to economise in power requirements of the prime movers in the factory and have allowed more than 36 kW/t.c.h. However, many factories use only 25-30 kW/t.c.h. for the total of their prime movers, thus $25 \times 250 = 6250$ kWh. Requirements for the manufacture remain the same, thus available for outside purposes: $27,025 - 6,250 = 20,825$ kWh.

The operation of this power production unit and its adaptation to possible variations in conditions of the factory operation (though as little as possible) can be assured automatically by a quick-action valve of hydraulic - electronic control and of high sensitivity which regulates the proportion of steam bled off from the turbine.

Where the supply authority requires it, it is possible to add a method of regulation and stabilisation of the power delivered to the public network. This requires that the supply should not be disturbed by stoppages or breakdowns in the factory, which can cause an abrupt drop in power production from 27,000 to 18,000 kW, or by 33%. These are the most severe conditions which can be encountered for regulation of supply.

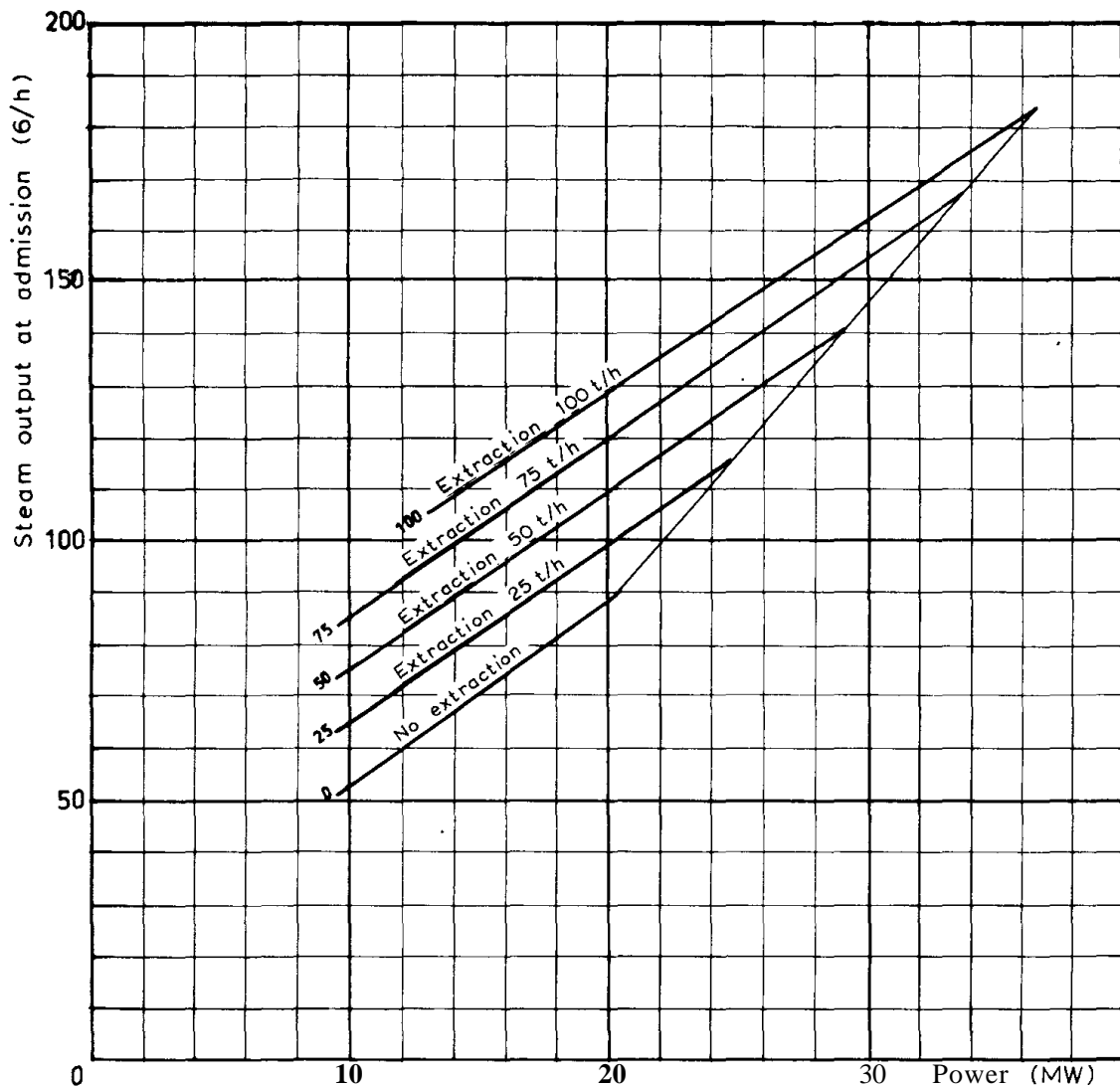


Fig. 41.37. Steam consumption and power as a function of extraction.

We give a graph (Fig. 41.37) of the power and steam consumption figures as a function of the quantity of pass-out steam, in the conditions of the example given.

We may comment that a factory thus equipped can dispose for sale, or for irrigation of its cane lands, 70-80 kW/t.c.h.

SALE OF POWER BY THE SUGAR FACTORY

The production of energy by the sugar factory for the public supply authorities poses certain problems:

(a) The sugar factory is a seasonal industry, whereas the needs of the public supply are necessarily for the whole year. The situation is similar with regard to irrigation needs for a tropical factory which produces the cane which supplies it; with this difference: that the irrigation demand is during the dry season and irrigation loses its importance during the rest of the year. When the public supply is partly derived from a hydro-electric plant, the situation again is similar; the sugar factory then can supply power during the periods of low water flow and thus fill the gap in the supply of hydro-power. Again, the demand for electricity coincides only partially with the period of power production from the factory.



Fig. 41.38. "Promill" pellet mill working.

(b) It is thus necessary to be able to store excess bagasse during twelve months or at least during a longer period than the crushing season. Unfortunately, bagasse is a very light, voluminous material, costly to compress or to store and handle, and presenting some fire risk.

The bagasse may be pressed and stored in bales (*cf.* p. 915), but the operation and labour requirements are expensive. At present (1982) attention is being directed to a solution which seems more elegant: "pelletisation" of the bagasse. This consists of drying, fibrating and compressing the bagasse and extruding it in the form of fingers of the dimensions of a cigarette or large pencil, several centimetres in length and described as "pellets", or sometimes called granules. Equipment for this operation and extrusion is made in France by Promill, in California by C.P.M. (California Pellet Mill), and in Germany by Kahl (Figs. 41.38 and 41.39).

Transformation of bagasse into pellets requires a previous moisture content of 15%, and heating to 80°C. The operation consumes 80 kWh/tonne of pellets.

The pellets are delivered at 8 or 10% moisture; their N.C.V. is thus about 3.800 kcal/kg (6,840 B.T.U./lb.). They cannot be used in boilers with spreader-stoker furnaces since they are too heavy to burn in suspension and fall immediately onto the grate; thus they would have to be broken up before feeding to the furnace. It would seem that they could be burnt



Fig. 41.39. "C.P.M." pellet mill open for cleaning.

in a travelling-grate furnace, as for fine coal, without previous disintegration; but we have no knowledge of tests of this type. They form an excellent fuel, but present difficulties on account of the high ash content, which may be as high as 10–12%. Hence the necessity for a travelling grate, rather than a rocking grate. We may comment that 1 kg of pellets at 10% moisture would give 4.8 kg of steam at 45 atm and 425°C, more than double that from 1 kg of bagasse at 48% moisture. If we compare the steam production per kg of bagasse before and after forming into pellets, the 560 g of pellets derived from 1 kg of bagasse gives 2.7 kg of steam compared with 2.2, or 22% more; but this does not say that it can be burnt in the same furnace.

It is probable that the development of pelletisation will lead to the design of a type of furnace and combustion chamber of a special design permitting efficient combustion of this fuel.

There are some bagasse feeders of Star type (Woodex System), which permit combustion of bagasse pellets in a spreader-stoker type furnace.

TABLE 41.29

STEAM CONSUMPTION OF TURBO-GENERATORS (kg/kWh)

Admission pressure and superheat		Back-pressure		Steam consumption (kg/kWh) ^a		
(kg/cm ²)	(p.s.i.g.)	(kg/cm ²)	(p.s.i.g.)	at ½ load	at load	at full load
18; 325°C	250; 620°F	0.5	7	12.3	11.4	10.6
22; 350°C	300; 660°F	0.6	8	11.6	10.8	10.0
25; 380°C	350; 700°F	1.0	15	11.7	10.9	10.1
25; 380°C	350; 700°F	1.5	20	12.7	11.8	10.9
25; 380°C	350; 700°F	0.07 ^b	1 ^b	6.2	5.7	5.3
45; 440°C	640; 825°F	0.09 ^b	1.4 ^b	5.7	5.3	5.0

^a For lb./kWh, multiply by 2.2.

^b Absolute.

Steam consumption of prime movers

The steam consumption of steam turbines will be calculated as indicated on p. 1063.

Example. Table 41.29 gives steam consumption of a turbo-alternator, for several typical cases.

Losses. Two types of loss have to be taken into account:

(a) Losses in turbines: leaks occur predominantly around the shaft, at the labyrinth packing. We may reckon 3%. This loss is reckoned between admission and exhaust of steam.

(b) Piping: the steam pipes lose steam by leaks at the joints, by condensation and at steam traps, etc. Deerr⁶⁹ estimates that a factory has about 5 m² (56 sq.ft.) of pipe surface per t.c.h., in high-pressure steam piping, and as much in exhaust steam piping. Say a total of 10 m²/t.c.h. (112 sq.ft./t.c.h.)

If these pipes are bare, they would lose:

Exhaust steam: 1,200 kcal/m²/h) (450 B.T.U./sq.ft./h) = 10 kg (20 lb.) steam/t.c.

Live steam: 3,600 kcal/mVh (1,350 B.T.U./sq.ft./h) = 30 kg (60 lb.) steam/t.c.

Properly lagged, these pipes would lose on an average 300 kcal/mVh (110 B.T.U./sq.ft./h), say about 3,000 kcal/t.c. or approximately 5 kg (10 lb.) steam/t.c.

(c) Multiple effect. In estimating that an evaporator of n effects evaporates n kg of water per kg of steam admitted to the first effect, we have taken into account its losses by radiation and convection. However, this is the case of a multiple effect which is correctly lagged. If it is badly lagged, we have seen elsewhere (*cf.* p. 544) the loss which must then be calculated.

Steam consumption per tonne cane. The steam consumption per t.c. varies finally between rather wide limits, according to the degree of steam economy realised in the manufacture, *i.e.* according as operation is in triple, quadruple, or quintuple effect; according to the extent of vapour bleeding used; and also whether or not thermo-compression is employed, etc.

The highest steam consumptions are reached where white sugar is being manufactured, with evaporation in quadruple or, obviously, particularly in triple effect, without vapour bleeding. Steam consumption may then reach 60—65% on cane.

With raw sugar, according to the economy sought at the evaporators and the quantity of imbibition used, steam consumption will amount to 50—60% on cane.

In well-equipped modern factories with high-pressure boilers and superheaters, quadruple effects and a full sequence of vapour bleeding, steam consumption should be between 45 and 50% on cane.

With quintuple effects, thermo-compression or pressure evaporation, it is possible to go as low as 36—45% on cane.

Fluctuations in steam consumption. The variations in momentary requirements of steam in the sugar factory are due to many factors, only one of which is unavoidable in practice: the fluctuations in steam consumption at the vacuum pans. We have seen (p. 663) the tremendous variations in evaporation rate from the beginning to the end of a strike. Since the number of pans is generally between 4 and 6, these individual variations have a very marked effect on the general steam consumption of the factory.

For the pans, we may estimate a total variation of approximately 50% (in other words, from 50 to 150, if the mean consumption is 100). According to the brix value adopted for the syrup, the quantities of dilution and washing water, etc., added at the pan station, the total steam consumption of this station will vary from 5 to 10% on cane (112 - 224 lb./t.c.).

We may comment that it is preferable to express this variation as a percentage on cane rather than to reckon it as a percentage on the total steam consumption, since in practice only the variation in steam consumption at the pans is concerned. As a percentage of the total steam consumption, this variation will generally represent between ± 10 and $\pm 20\%$.

A big factor in the variations in steam demand is the relative dimensions of the individual vacuum pans. A factory having very large pans relative to its capacity, particularly in A masseuite, will show much greater variations in its steam consumption than a factory having a larger number of smaller pans.

Heat balance. If, instead of calculating in weight of steam, the balance is expressed in heat units, a somewhat different table is obtained. The following is an example given by three Taiwan technologists^{69a} for three defecation factories:

	<i>kcal/t.c.</i>	<i>B.T.U./t.c.</i>	<i>%</i>
Turbo-generators:	17,200	69,300	5
Mills:	12,600	50,900	4
Pumps:	4,300	17,200	1
Total for power:	34,100	137,400	10
Evaporation:	151,800	612,200	47
Pans:	90,000	362,900	28
Total for boiling house:	241,800	975,100	75
Other requirements:	470	1,900	0.1
Losses:	10,600	42,600	2.7
Total heat consumption:	287,000	1,157,000	87.8
Heat lost in condenser water:	40,000	160,000	12.2
Total heat output:	327,000	1,317,000	100

Carbonatation factories require 20% more heat, due mostly to requirements at the pans, on account of the low brix of syrup. The three defecation factories in the previous example had a syrup brix of 65 and crushed 2,325 t.c. per day (weighted average).

As general figures, heat consumption in sugar factories is:

Power:	10 - 12 %
Boiling house:	70 - 78 % (40% being for evaporation)
Other uses:	0.1 - 7 %
Losses:	0.1 - 7.3 %
Lost in condenser water:	9.7-13 %

Influence of rate. Heat requirements per t.c. vary when hourly crushing rate increases or decreases. In Taiwan, this has been expressed by:

$$M = 340,000 - 3,400 (A' - A) \text{ kcal/t.c.} \quad (1,350,000 - 13,500 (A' - A) \text{ B.T.U./t.c.})$$

M = total heat requirements for the factory, in kcal/t.c. (B.T.U./t.c.)

A = nominal crushing rate, in t.c.h.

A' = actual crushing rate, in t.c.h.

It is also estimated that a 10% increase in crushing rate leads to an 8% decrease in heat consumption per t.c.

Another report from Taiwan^{69b} states that a 2% increase in crushing rate permits of a saving of approximately 1% in steam consumption per t.c. for electrified factories.

Total power. Similarly, the average total power consumed in a sugar factory is rather variable: Deerr⁷⁰ estimates 18 i.h.p./t.c.h.; Tromp⁷¹ estimates 25 i.h.p./t.c.h., but the latter author comments that it is necessary to calculate this value in each case, since it varies greatly in different installations.

In India⁷², a figure of 32 i.h.p./t.c.h. is given.

Webre⁷³ gives as an upper limit 26 kW = 35 i.h.p./t.c.h.

The high power consumption used in recent years by the South African industry, for cane preparation and the long milling trains, has led to a total figure of 36-40 kW/t.c.h.

Influence of electrification. It is appropriate to comment that electrification itself effects no economy in steam. It involves a triple transformation of energy (thermal energy into mechanical, mechanical energy into electrical, electrical energy into mechanical energy), which affects the steam consumption in each case, and thus loses the benefit of the high thermodynamic efficiency of the steam turbine.

For example, the calculation of steam balance (p. 1012) carried out for a factory with steam-driven mills, steam engine for central vacuum system, and a generating set for pumps and small isolated units only, where the steam engines were drop-valve engines working at 15 kg/cm² (215 p.s.i.), with steam slightly superheated to 250°C (480°F), gave a total steam consumption hardly higher than for the electrified factory: (470 kg/t.c. (1,050 lb./t.c.)).

However, it is electrification that has effected the main progress in steam economy, since it has permitted the elimination of direct-acting pumps and small isolated units by replacing them with motor-driven units.

Effect of brix. In all the preceding considerations, the steam consumption has been related to tonnage of cane. This is normal practice, since the crushing rate in terms of cane is the essential figure for the factory. In reality, however, the steam consumption depends more closely on the total quantity of brix treated than on the actual quantity of cane. For accurate comparisons between factories, or within the same factory, it may be valuable to relate the weight of steam consumed not to the weight of cane handled but to the weight of brix entering the factory, which is of the order of 15% on cane. It may be commented that, for a given factory, an increase in brix may have a positive or a negative effect on steam consumption. In Taiwan^{69b} it is estimated that a 1% increase in brix of mixed juice leads to a 1% decrease in steam consumption per tonne of cane.

In Java an average steam consumption was reported as:

- 3,200 kcal/kg (5,760 B.T.U./lb.) of brix, for defecation factories (raw sugar)
- 3,500 kcal/kg (6,300 B.T.U./lb.) of brix, for sulphitation factories (white sugar)
- 3,300 kcal/kg (5,940 B.T.U./lb.) of brix, for carbonatation factories (white sugar)

reckoned in terms of fuel burnt.

In South Africa¹⁸, the average consumption on the same basis was given as 4,170 kcal/kg of brix, of which 3,817 was obtained from the bagasse and the remainder from supplementary fuel.

GENERAL ARRANGEMENT OF THE STEAM CYCLE

The simplest steam cycle in a modern factory consists of producing the steam in high-pressure boilers and expanding it to the pressure necessary for manufacture, in turbo-alternators (T.A., Fig. 41.40), which produce all the electric power necessary for the factory. When the mills are driven by steam turbines, these are supplied by a branch taken directly from the high-pressure steam piping.

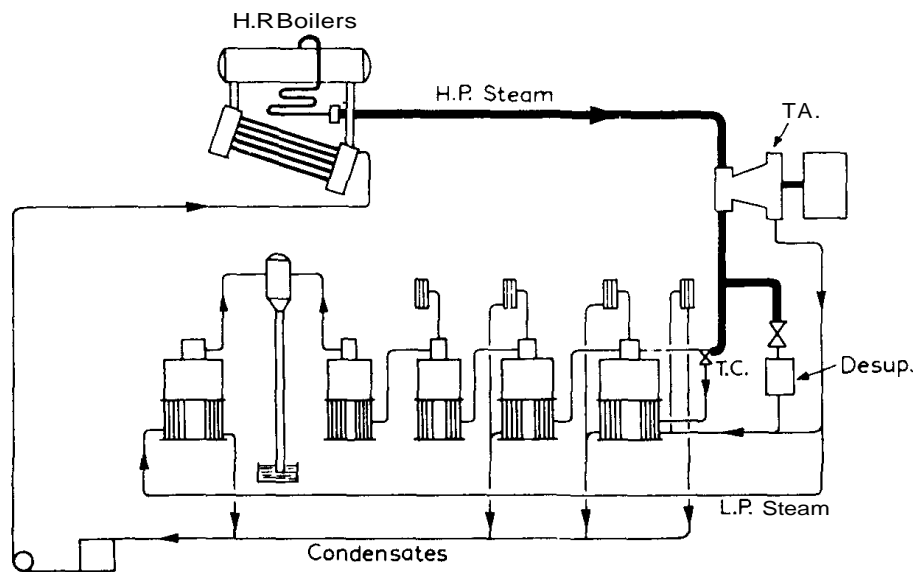


Fig. 41.40. Steam circuit No. 1.

Since the quantity of exhaust steam supplied by these turbo sets is insufficient, the high-pressure circuit is connected to the low-pressure circuit through a steam reducer – de-superheater (Desup.) which supplies the make-up steam necessary. Finally, the high-pressure circuit will be completed by a connection to the thermo-compressor (T.C.).

Where the mills are driven by reciprocating engines, a medium-pressure (M.P.) circuit is provided serving these engines (Fig. 41.41).

A partial reducer - de-superheater (Desup.) should be installed between the H.P. and M.P. circuits, but a simple pressure reducer (R.V.) will be sufficient between the M.P. and L.P. circuits.

Choice of steam pressure

In the sugar factory we have three principal steam pressures:

- (a) The H.P. steam for the turbo sets
- (b) The M.P. steam for reciprocating engines
- (c) The L.P. steam for manufacture.

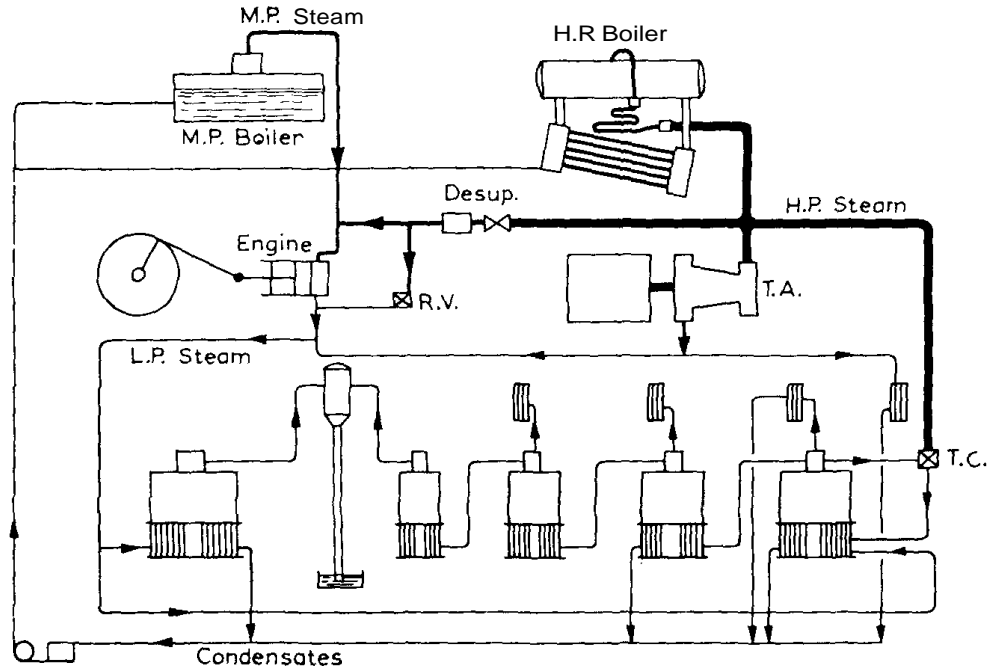


Fig. 41.41. Steam circuit No. 2.

(a) **Choice of high pressure.** To achieve a suitably economical operation of the steam turbines, it is necessary to have a high pressure between 16 and 45 kg/cm² (225 – 640 p.s.i.).

Below 16 kg/cm², the steam consumption of the turbines increases rapidly. Also, the domain of the reciprocating engine readily extends to 15 kg/cm². If this limit were not exceeded, there would be every advantage, as much from the point of view of first cost of the installation as from that of steam consumption, in retaining the drop-valve engine, operating at 15 kg/cm², with a slight superheat to 250°C.

Above 45 kg/cm², the expense of the installation, due to the rapid increase in cost of the boilers, with increasing pressure, reduces the value of the steam-economy obtained.

Furthermore, above 30 kg/cm² (425 p.s.i.), it is generally necessary to treat the boiler feed water, since its quality and purity become more and more critical as the pressure is increased. Such water treatment involves a complication and further expense.

Steam conditions are generally kept within the ranges shown in Table 41.30.

TABLE 41.30

OPTIMAL CONDITIONS FOR *h.p.* STEAM IN THE SUGAR FACTORY

<i>kg/cm²</i>	<i>p.s.i.</i>	°C	°F
18	250	300 – 350	575 – 650
20	280	325 – 375	625 – 700
25	350	350 – 400	650 – 750
30	425	375 – 425	700 – 800
45	640	420 – 450	790 – 840

Within these various values, the advantages and disadvantages compensate each other and the choice may depend on secondary considerations such as space required.

(b) Choice of medium pressure. The M.P. is more especially of interest with regard to reciprocating engines. To obtain economical operation with these, it is necessary to use a pressure between 6 and 15 kg/cm² (85 – 215 p.s.i.).

The highest pressures, 12 – 15 kg/cm² (170 – 215 p.s.i.), should be adopted for factories without turbines, but with an electric generating set. Their steam engines will then function under particularly economical conditions.

However, factories having the three levels of pressure will maintain an M.P. level of 6 kg/cm² (85 p.s.i.) or thereabouts, particularly if they do not use reciprocating engines.

Where reciprocating engines are used, the pressure should not go below 6 kg/cm² (85 p.s.i.), on account of the rapid increase in steam consumption of these engines below that figure.

(c) Choice of low pressure. The low pressure is generally at the same time the upper limit of pressure for the boiling house as well as the back-pressure for the engines.

To avoid development of colour in the juices and to avoid destruction of sucrose, we have seen (Table 32.4) that it is advisable not to exceed 1.5 kg/cm² (20 p.s.i.g.) in an ordinary multiple effect, or 2 kg/cm² (28 p.s.i.g.) in a Kestner.

If pressure evaporation is to be used, it will be necessary to work to these limits or to approach them closely. Otherwise, pressure should be kept between 0.5 and 1.2 kg/cm² (7 - 17 p.s.i.g.). If reciprocating engines are used, the pressure will be kept between 0.5 and 0.6 kg/cm² (7-9 p.s.i.g.). If not, it will be advisable to increase to 0.8 – 1.2 kg/cm² (12- 17 p.s.i.g.).

Sometimes the low-pressure system is divided into two: a higher pressure for coil pans (1.4 kg/cm², for example) and a lower pressure for juice heaters and multiple effects (0.5 kg/cm², for example). This has the disadvantage of complicating the piping system for steam distribution.

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TABLE 41.1 A

PROPERTIES OF DRY SATURATED STEAM (METRIC UNITS)

t	P	d	q	r	X	\bar{a})	v
0	0.00623	0.99987	0	597.2	597.2	0.00485	206.3
5	0.00889	0.99999	5	594.4	599.4	0.00680	147.2
10	0.01251	0.99973	10	591.6	601.6	0.00940	106.4
15	0.01738	0.99913	15	588.8	603.8	0.01282	77.99
16	0.01853	0.99897	16	588.3	604.3	0.01363	73.39
17	0.01975	0.99880	17	587.7	604.7	0.01447	69.10
18	0.02103	0.99862	18	587.1	605.1	0.01536	65.10
19	0.02239	0.99843	19	586.6	605.6	0.01630	61.35
20	0.02383	0.99823	20	586	606	0.01729	57.84
21	0.02534	0.99802	21	585.5	606.5	0.01833	54.56
22	0.02694	0.99780	22	584.9	606.9	0.01942	51.49
23	0.02863	0.99756	23	584.3	607.3	0.02056	48.63
24	0.03041	0.99732	24	583.8	607.8	0.02177	45.94
25	0.03229	0.99707	25	583.2	608.2	0.02304	43.41
26	0.03426	0.99681	26	582.6	608.6	0.02437	41.04
27	0.03634	0.99654	27	582.1	609.1	0.02576	38.82
28	0.03853	0.99626	28	581.5	609.5	0.02723	36.73
29	0.04083	0.99597	29	581	610	0.02876	34.77
30	0.04325	0.99567	30	580.4	610.4	0.03036	32.93
31	0.04580	0.99537	31	579.8	610.8	0.03204	31.20
32	0.04847	0.99505	32	579.3	611.3	0.03380	29.58
33	0.05128	0.99463	33	578.7	611.7	0.03565	28.05
34	0.05423	0.99440	34	578.1	612.1	0.03758	26.61
35	0.05733	0.99406	35	577.5	612.5	0.03960	25.25
36	0.06057	0.99371	36	577	613	0.04171	23.97
37	0.06398	0.99336	37	576.4	613.4	0.04392	22.77
38	0.06755	0.99299	38	575.9	613.9	0.04622	21.63
39	0.07129	0.99262	39	575.3	614.3	0.04863	20.56
40	0.07520	0.99224	40	574.7	614.7	0.05114	19.55
41	0.07930	0.99186	41	574.2	615.2	0.05377	18.60
42	0.08360	0.99147	42	573.6	615.6	0.05650	17.70
43	0.08809	0.99107	43	573	616	0.05935	16.85
44	0.09279	0.99066	44	572.4	616.4	0.06233	16.04
45	0.09771	0.99024	45	571.8	616.8	0.06544	15.28
46	0.10284	0.98982	46	571.2	617.2	0.06867	14.56
47	0.10821	0.98940	47	570.7	617.7	0.07203	13.88
48	0.11382	0.98896	48	570.1	618.1	0.07553	13.24
49	0.11967	0.98852	49	569.5	618.5	0.07918	12.63
50	0.12578	0.98807	50	569	619	0.08298	12.05

t = temperature, in °C
 p = absolute pressure of the vapour, in kg/cm²
 p' = gauge pressure of the vapour, in kg/cm²
 d = density of water at t °C, in kg/dm³
 q = sensible heat to raise 1 kg of water from 0 to t °C, in kcal/kg

r = latent heat of vaporisation of water at t °C, in kcal/g
 X = $q + r$ = total heat of 1 kg of water at t °C
 $\bar{\omega}$ = density of vapour at t °C, in kg/m³
 v = specific volume of vapour at t °C, in m³/kg

TABLE 41.1 A (continued)

t	P	d	q	r	X	$\bar{\omega}$	v
51	0.1322	0.98762	50.9	568.4	619.3	0.0869	11.50
52	0.1388	0.98715	51.9	567.8	619.7	0.0910	10.98
53	0.1457	0.98669	52.9	567.3	620.2	0.0953	10.49
54	0.1530	0.98621	53.9	566.7	620.6	0.0997	10.02
55	0.1605	0.98573	54.9	566.1	621	0.1043	9.584
56	0.1684	0.98524	55.9	565.6	621.5	0.1091	9.164
57	0.1765	0.98478	56.9	565	621.9	0.1141	8.764
58	0.1850	0.98425	57.9	564.4	622.3	0.1193	8.385
59	0.1939	0.98375	58.9	563.8	622.7	0.1247	8.025
60	0.2031	0.98324	59.9	563.3	623.2	0.1302	7.682
61	0.2127	0.98272	60.9	562.7	623.6	0.1359	7.356
62	0.2227	0.98220	61.9	562.1	624	0.1419	7.046
63	0.2330	0.98167	62.9	561.5	624.4	0.1481	6.752
64	0.2438	0.98113	63.9	560.9	624.8	0.1545	6.473
65	0.2550	0.98059	64.9	560.3	625.2	0.1611	6.206
66	0.2666	0.98005	65.9	559.7	625.6	0.1680	5.951
67	0.2787	0.97950	66.9	559.1	626	0.1752	5.709
68	0.2912	0.97894	67.9	558.5	626.4	0.1826	5.478
69	0.3024	0.97838	68.9	558	626.9	0.1902	5.258
70	0.3177	0.97781	69.9	557.4	627.3	0.1981	5.049
71	0.3317	0.97723	70.9	556.8	627.7	0.2062	4.849
72	0.3463	0.97666	71.9	556.2	628.1	0.2146	4.658
73	0.3613	0.97607	72.9	555.6	628.5	0.2234	4.476
74	0.3769	0.97548	73.9	555	628.9	0.2324	4.302
75	0.3931	0.97489	74.9	554.4	629.3	0.2418	4.136
76	0.4098	0.97429	75.9	553.8	629.7	0.2514	3.977
77	0.4272	0.97368	76.9	553.2	630.1	0.2614	3.826
78	0.4451	0.97307	77.9	552.6	630.5	0.2717	3.681
79	0.4637	0.97245	78.9	552	630.9	0.2823	3.543
80	0.4829	0.97183	80	551.3	631.3	0.2933	3.410
81	0.5028	0.97121	81	550.7	631.7	0.3046	3.283
82	0.5234	0.97057	82	550.1	632.1	0.3162	3.162
83	0.5447	0.96994	83	549.5	632.5	0.3282	3.047
84	0.5667	0.96930	84	548.8	632.8	0.3406	2.936
85	0.5894	0.96865	85	548.2	633.2	0.3534	2.830
86	0.6129	0.96800	86	547.6	633.6	0.3666	2.728
87	0.6372	0.96734	87	547	634	0.3802	2.630
88	0.6623	0.96668	88	546.4	634.4	0.3942	2.537
89	0.6882	0.96601	89	545.7	634.7	0.4086	2.447
90	0.7149	0.96534	90	545.1	635.1	0.4235	2.361
91	0.7425	0.96467	91	544.5	635.5	0.4388	2.279
92	0.7710	0.96399	92	534.9	635.9	0.4545	2.200
93	0.8004	0.96330	93	543.3	636.3	0.4707	2.124
94	0.8307	0.96261	94	542.7	636.7	0.4873	2.051
95	0.8619	0.96192	95	542	637	0.5045	1.981
96	0.8942	0.96122	96	541.4	637.4	0.5221	1.914
97	0.9274	0.96051	97	540.8	637.8	0.5402	1.851
98	0.9616	0.95981	98	540.2	638.2	0.5588	1.789
99	0.9969	0.95909	99	539.5	638.5	0.5780	1.730
100	1.0332	0.95838	100	538.9	638.9	0.5977	1.673

TABLE 41.1 A (continued)

t	P	p'	l	r	X	$\bar{\omega}$	v
101	1.0707	0.0375	101	538.3	639.3	0.6179	1.618
102	1.1092	0.0760	102	537.6	639.6	0.6387	1.565
103	1.1489	0.1157	103.1	536.9	640	0.6601	1.515
104	1.1898	0.1566	104.1	536.2	640.3	0.6820	1.466
105	1.2318	0.1986	105.1	535.6	640.7	0.7045	1.419
106	1.2751	0.2419	106.1	535	641.1	0.7276	1.374
107	1.3196	0.2864	107.1	534.3	641.4	0.7514	1.331
108	1.3654	0.3322	108.1	533.6	641.7	0.7758	1.289
109	1.4125	0.3793	109.1	533	642.1	0.8008	1.249
110	1.4609	0.4277	110.1	532.4	642.5	0.8265	1.210
111	1.5106	0.4774	111.1	531.8	642.9	0.8528	1.173
112	1.5618	0.5286	112.1	531.1	643.2	0.8798	1.137
113	1.6144	0.5812	113.2	530.4	643.6	0.9075	1.102
114	1.6684	0.6352	114.2	529.7	643.9	0.9359	1.068
115	1.7239	0.6907	115.2	529.1	644.3	0.9650	1.036
116	1.7809	0.7477	116.2	528.4	644.6	0.9947	1.005
117	1.8394	0.8062	117.2	527.8	645	1.026	0.9752
118	1.8995	0.8663	118.2	527.1	645.3	0.057	0.9462
119	1.9612	0.9280	119.2	526.4	645.6	1.089	0.9183
120	2.0245	0.9913	120.3	525.7	646	1.122	0.8914
121	2.0895	1.0563	121.3	525.1	646.4	1.156	0.8655
122	2.1561	1.1229	122.3	524.4	646.7	1.190	0.8404
123	2.2245	1.1913	123.3	523.7	647	1.225	0.8161
124	2.2947	1.2615	124.3	523.1	647.4	1.262	0.7927
125	2.3666	1.3334	125.3	522.4	647.7	1.299	0.7701
126	2.4404	1.4072	126.4	521.6	648	1.337	0.7482
127	2.5160	1.4828	127.4	520.9	648.3	1.376	0.7271
128	2.5935	1.5603	128.4	520.3	648.7	1.415	0.7068
129	2.6730	1.6398	129.4	519.6	649	1.455	0.6871
130	2.7544	1.7212	130.4	518.9	649.3	1.496	0.6680
131	2.8378	1.8046	131.4	518.2	649.6	1.539	0.6496
132	2.9233	1.8901	132.5	517.4	649.9	1.583	0.6318
133	3.011	1.978	133.5	516.7	650.2	1.628	0.6146
134	3.101	2.068	134.5	516	650.5	1.673	0.5979
135	3.192	2.159	135.5	515.3	650.8	1.719	0.5817
136	3.286	2.253	136.6	514.6	651.2	1.767	0.5661
137	3.382	2.349	137.6	513.9	651.5	1.815	0.5510
138	3.481	2.448	138.6	513.3	651.9	1.864	0.5363
139	3.582	2.549	139.6	512.6	652.2	1.915	0.5221
140	3.685	2.652	140.6	511.9	652.5	1.967	0.5084
141	3.790	2.757	141.7	511.1	652.8	2.020	0.4951
142	3.898	2.865	142.7	510.4	653.1	2.074	0.4823
143	4.009	2.976	143.7	509.7	653.4	2.129	0.4698
144	4.122	3.089	144.8	508.9	653.7	2.185	0.4577
145	4.237	3.204	145.8	508.2	654	2.243	0.4459
146	4.355	3.322	146.8	507.5	654.3	2.302	0.4345
147	4.476	3.443	147.8	506.8	654.6	2.362	0.4235
148	4.559	3.566	148.9	506	654.9	2.423	0.4128
149	4.725	3.692	149.9	505.3	655.2	2.485	0.4024
150	4.854	3.821	150.9	504.6	655.5	2.548	0.3924

TABLE 41.1 A (continued)

p	p'	t	q	r	A	$\bar{o}i$	v
5	3.967	151.11	152.1	503.7	655.8	2.621	0.3816
5.5	4.467	154.71	155.8	501.1	656.9	2.867	0.3488
6	4.967	158.08	159.3	498.5	657.8	3.112	0.3213
6.5	5.467	161.21	162.5	496.1	658.6	3.356	0.2980
7	5.967	164.17	165.6	493.8	659.4	3.600	0.2778
7.5	6.467	166.96	168.5	491.6	660.1	3.842	0.2603
8	6.967	169.61	171.3	489.5	660.8	4.085	0.2448
8.5	7.467	172.12	173.9	487.5	661.4	4.327	0.2311
9	7.967	174.53	176.4	485.6	662	4.568	0.2189
9.5	8.467	176.83	178.9	483.6	662.5	4.809	0.2080
10	8.967	179.04	181.2	481.8	663	5.049	0.1981
10.5	9.467	181.16	183.4	480.1	663.5	5.290	0.1891
11	9.967	183.20	185.6	478.3	663.9	5.530	0.1808
11.5	10.467	185.17	187.7	476.6	664.3	5.770	0.1733
12	10.967	187.08	189.7	475	664.7	6.010	0.1664
12.5	11.467	188.92	191.6	473.5	665.1	6.249	0.1600
13	11.967	190.71	193.5	471.9	665.4	6.488	0.1541
13.5	12.467	192.45	195.3	470.4	665.7	6.728	0.1486
14	12.967	194.13	197.1	468.9	666	6.967	0.1435
14.5	13.467	195.77	198.9	467.4	666.3	7.207	0.1388
15	13.967	197.36	200.6	466	666.6	7.446	0.1343
16	14.967	200.43	203.9	463.2	667.1	7.925	0.1262
17	15.967	203.35	207.1	460.4	667.5	8.405	0.1190
18	16.967	206.14	210.1	457.8	667.9	8.886	0.1126
19	17.967	208.81	213	455.2	668.2	9.366	0.1068
20	18.967	211.38	215.8	452.7	668.5	9.846	0.1016
21	19.967	213.85	218.5	450.2	668.7	10.33	0.09682
22	20.967	216.23	221.2	447.7	668.9	10.81	0.09251
23	21.967	218.53	223.6	445.5	669.1	11.29	0.08856
24	22.967	220.75	226.1	443.2	669.3	11.78	0.08492
25	23.967	222.90	228.5	440.9	669.4	12.26	0.08157
26	24.967	224.99	230.8	438.7	669.5	12.75	0.07846
27	25.967	227.01	233	436.6	669.6	13.23	0.07557
28	26.967	228.98	235.2	434.4	669.6	13.72	0.07288
29	27.967	230.89	237.4	432.3	669.7	14.21	0.07037
30	28.967	232.76	239.5	430.2	669.7	14.70	0.06802
32	30.967	236.35	243.6	426.1	669.7	15.69	0.06375
34	32.967	239.77	247.5	422.1	669.6	16.68	0.05995
36	34.967	243.04	251.2	418.3	669.5	17.68	0.05658
40	38.967	249.18	258.2	410.8	669	19.69	0.05078
45	43.967	256.23	266.5	401.7	668.2	22.25	0.04495
50	48.967	262.70	274.2	393.1	667.3	24.85	0.04024
60	58.967	274.29	288.4	376.6	665	30.21	0.03310
80	78.967	293.62	312.6	346.3	658.9	41.60	0.02404
100	98.967	309.53	334	317.1	651.1	54.21	0.01845
150	148.967	340.56	381.7	243.2	624.9	93.90	0.01065
200	198.967	364.08	431.4	150.7	582.1	161.2	0.00620
225.6	224.567	374.15	501.5	0	501.5	315	0.00318

TABLE 41.1 B

PROPERTIES OF DRY SATURATED STEAM (BRITISH UNITS)

t	P	d	q	r	X	$\bar{\omega}$	v
32	0.0886	62.420	0	1075	1075	0.000303	3305
40	0.1271	62.428	8	1071.3	1079.3	0.000409	2445
50	0.1780	62.411	18.1	1064.9	1083	0.000587	1705
60	0.2563	62.368	28.1	1059.3	1087.4	0.000828	1208
62	0.2751	62.357	30.1	1058.2	1088.3	0.000886	1130
64	0.2951	62.344	32.1	1057.1	1089.2	0.000947	1057
66	0.3163	62.331	34.1	1056	1090.1	0.001011	989.2
68	0.3389	62.318	36.1	1054.8	1090.9	0.001079	926.5
70	0.3630	62.303	38.1	1053.7	1091.8	0.001152	868.4
72	0.3886	62.287	40.1	1052.6	1092.7	0.001228	814.5
74	0.4156	62.271	42	1051.5	1093.5	0.001308	764.6
76	0.4444	62.254	44	1050.4	1094.4	0.001393	717.9
78	0.4749	62.235	46	1049.2	1095.2	0.001483	674.2
80	0.5070	62.218	48	1048.1	1096.1	0.001579	633.5
82	0.5411	62.198	50	1046.9	1096.9	0.001679	595.7
84	0.5771	62.178	52	1045.8	1097.8	0.001785	560.4
86	0.6152	62.158	54	1044.7	1098.7	0.001896	527.5
88	0.6556	62.137	56	1043.6	1099.6	0.002012	496.9
90	0.6983	62.113	58	1042.5	1100.5	0.002135	468.3
92	0.7434	62.092	60	1041.3	1101.3	0.002265	441.5
94	0.7910	62.069	62	1040.1	1102.1	0.002402	416.4
96	0.8410	62.045	64	1039	1103	0.002545	393
98	0.8939	62.021	66	1037.9	1103.9	0.002696	371
100	0.9495	61.995	68	1036.8	1104.8	0.002853	350.5
102	1.0080	61.970	70	1035.7	1105.7	0.003019	331.2
104	1.0696	61.944	72	1034.5	1106.5	0.003193	313.2
106	1.1347	61.917	74	1033.4	1107.4	0.003376	296.3
108	1.2033	61.890	75.9	1032.2	1108.2	0.003567	280.4
110	1.2752	61.862	77.9	1031	1109	0.003767	265.5
112	1.3509	61.833	79.9	1029.8	1109.8	0.003977	251.4
114	1.4304	61.804	81.9	1028.7	1110.6	0.004197	238.3
116	1.5137	61.775	83.9	1027.6	1111.5	0.004427	225.9
118	1.6012	61.745	85.9	1026.4	1112.3	0.004666	214.3
120	1.6929	61.714	87.9	1025.2	1113.1	0.004918	203.4
122	1.789	61.683	89.9	1024.2	1114.1	0.005180	193
124	1.890	61.652	91.9	1023.1	1115	0.005455	183.3
126	1.996	61.619	93.9	1021.9	1115.8	0.005742	174.2
128	2.107	61.587	95.9	1020.8	1116.7	0.006042	165.5
130	2.223	61.554	97.9	1019.6	1117.5	0.006353	157.4

t = temperature, in °F
 p = absolute pressure of the steam, in p.s.i.a.
 p' = gauge pressure of the steam, in p.s.i.g.
 d = density of water at t °F, in lb./cu.ft.
 q = sensible heat to raise 1 lb. of water from 32 to t °F, in B.T.U./lb.

r = latent heat of vaporisation of water at t °F, in B.T.U./lb.
 X = $q + r$ = total heat of 1 lb. of water at t °F
 $\bar{\omega}$ = density of vapour at t °F, in lb./cu.ft.
 v = specific volume of vapour at t °F, in cu.ft./lb.

TABLE 41.1 B (continued)

t	P	d	q	r	x	$\bar{\omega}$	v
132	2.345	61.520	99.9	1018.5	1118.4	0.006676	149.8
134	2.472	61.486	101.9	1017.4	1119.3	0.007016	142.5
136	2.605	61.452	103.9	1016.2	1120.1	0.007371	135.7
138	2.744	61.417	105.9	1015.1	1121	0.007741	129.2
140	2.889	61.382	107.9	1013.9	1121.8	0.008127	123.1
142	3.041	61.346	109.9	1012.8	1122.7	0.008526	117.3
144	3.200	61.309	111.9	1011.6	1123.5	0.008943	111.8
146	3.365	61.272	113.9	1010.4	1124.3	0.009375	106.7
148	3.538	61.235	115.9	1009.2	1125.1	0.009824	101.8
150	3.719	61.198	117.9	1008	1125.9	0.01029	97.14
152	3.907	61.160	119.9	1006.9	1126.7	0.01078	92.74
154	4.102	61.121	121.9	1005.7	1127.6	0.01129	88.57
156	4.306	61.082	123.9	1004.5	1128.4	0.01182	84.62
158	4.519	61.043	125.9	1003.3	1129.2	0.01236	80.88
160	4.741	61.003	127.9	1002.1	1130	0.01293	77.32
162	4.970	60.963	129.9	1000.9	1130.8	0.01352	73.97
164	5.213	60.922	131.9	999.7	1131.5	0.01413	70.77
166	5.463	60.881	133.9	998.4	1132.3	0.01477	67.73
168	5.723	60.840	135.9	997.2	1133.1	0.01542	64.83
170	5.994	60.798	137.9	996	1133.9	0.01611	62.08
172	6.274	60.755	139.9	994.8	1134.7	0.01681	59.47
174	6.566	60.712	141.9	993.6	1135.5	0.01755	56.99
176	6.868	60.669	143.9	992.3	1136.2	0.01831	54.62
178	7.184	60.626	145.9	991.1	1137	0.01909	52.37
180	7.512	60.582	147.9	989.9	1137.8	0.01990	50.24
182	7.852	60.538	149.9	988.6	1138.5	0.02074	48.21
184	8.204	60.493	151.9	987.4	1139.3	0.02161	46.27
186	8.569	60.448	153.9	986.1	1140.1	0.02252	44.41
188	8.948	60.403	155.9	984.9	1140.8	0.02345	42.65
190	9.341	60.357	158	983.7	1141.6	0.02441	40.96
192	9.747	60.311	160	982.4	1142.4	0.02541	39.36
194	10.168	60.264	162	981.2	1143.1	0.02644	37.82
196	10.606	60.217	164	979.9	1143.9	0.02750	36.36
198	11.059	60.170	166	978.7	1144.7	0.02860	34.97
200	11.528	60.123	168	977.5	1145.4	0.02973	33.64
202	12.013	60.075	170	976.2	1146.2	0.03090	32.36
204	12.514	60.026	172	975	1147	0.03211	31.14
206	13.034	59.977	174	973.7	1147.8	0.03335	29.99
208	13.569	59.928	176	972.5	1148.5	0.03463	28.88
210	14.124	59.879	178	971.3	1149.3	0.03595	27.82

TABLE 41.1 B (continued)

t	P	p'	q	r	X	$\bar{\omega}$	v
212	14.696	59.830	180.1	970	1150.1	0.03731	26.80
214	15.290	0.594	182.1	968.8	1150.9	0.03872	25.83
216	15.902	1.206	184.1	967.4	1151.5	0.04017	24.89
218	16.535	1.839	186.1	966	1152.1	0.04166	24
220	17.189	2.493	188.1	964.7	1152.8	0.04320	23.15
222	17.863	3.167	190.2	963.5	1153.7	0.04478	22.33
224	18.559	3.863	192.2	962.2	1154.4	0.04641	21.55
226	19.275	4.579	194.2	960.8	1155	0.04809	20.79
228	20.016	5.320	196.2	959.2	1155.7	0.04982	20.07
230	20.779	6.083	198.2	958.3	1156.5	0.05160	19.38
232	21.567	6.871	200.2	957.1	1157.3	0.05343	18.72
234	22.380	7.684	202.3	955.7	1158	0.05531	18.08
236	23.218	8.522	204.3	954.3	1158.6	0.05724	17.47
238	24.081	9.385	206.3	953	1159.3	0.05923	16.88
240	24.970	10.274	208.3	951.7	1160	0.06127	16.32
242	25.885	11.189	210.4	950.4	1160.8	0.06337	15.78
244	26.828	12.132	212.4	949.1	1161.5	0.06553	15.26
246	27.797	13.101	214.4	947.7	1162.1	0.06775	14.76
248	28.795	14.099	216.5	946.3	1162.8	0.07003	14.28
250	29.825	15.129	218.5	945	1163.5	0.07236	13.82
252	30.883	16.187	220.5	943.6	1164.1	0.07476	13.38
254	31.973	17.277	222.5	942.3	1164.8	0.07723	12.95
256	33.093	18.397	224.6	940.9	1165.5	0.07976	12.54
258	34.244	19.548	226.6	939.5	1166.1	0.08237	12.14
260	35.427	20.731	228.6	938.1	1166.7	0.08504	11.76
262	36.643	21.947	230.7	936.8	1167.5	0.08776	11.39
264	37.893	23.197	232.7	935.4	1168.1	0.09057	11.04
266	39.177	24.481	234.7	934	1168.7	0.09345	10.70
268	40.496	25.800	236.8	932.6	1169.4	0.09640	10.37
270	41.851	27.155	238.8	931.1	1169.9	0.09941	10.06
272	43.250	28.554	240.9	929.6	1170.5	0.1025	9.756
274	44.680	29.984	242.9	928.2	1171.1	0.1057	9.462
276	46.143	31.447	245	926.8	1171.8	0.1089	9.179
278	47.650	32.954	247.1	925.4	1172.5	0.1123	8.907
280	49.200	34.504	249.1	924.1	1173.2	0.1157	8.643
282	50.790	36.094	251.1	922.7	1173.8	0.1192	8.389
284	52.420	37.724	253.2	921.3	1174.5	0.1228	8.144
286	54.080	39.384	255.2	919.8	1175	0.1264	7.908
288	55.800	41.104	257.3	918.4	1175.7	0.1302	7.681
290	57.555	42.859	259.3	917	1176.3	0.1340	7.461
292	59.355	44.659	261.4	915.5	1176.9	0.1380	7.247
294	61.200	46.504	263.4	914.1	1177.5	0.1420	7.040
296	63.090	48.394	265.5	912.7	1178.2	0.1462	6.841
298	65.025	50.329	267.6	911.2	1178.8	0.1504	6.650
300	67.006	52.310	269.6	909.7	1179.3	0.1547	6.464

TABLE 41.1 B (continued)

P	p'	t	q	r	λ	\bar{a}	v
74.7	60	307.33	277.2	904.3	1181.5	0.1714	5.836
84.7	70	316.05	286.2	897.7	1183.9	0.1929	5.184
94.7	80	323.92	294.4	891.6	1186	0.2143	4.666
104.7	90	331.17	301.8	886.1	1187.9	0.2357	4.244
114.7	100	337.89	308.9	880.6	1189.5	0.2570	3.892
124.7	110	344.16	315.4	875.6	1191	0.2781	3.596
134.7	120	350.05	321.5	870.8	1192.3	0.2993	3.341
144.7	130	355.59	327.5	866.2	1193.7	0.3204	3.121
154.7	140	360.85	333.1	861.7	1194.8	0.3415	2.928
164.7	150	365.85	338.4	857.5	1195.9	0.3626	2.758
174.7	160	370.61	343.4	853.5	1196.9	0.3836	2.607
184.7	170	375.18	348.2	849.5	1197.7	0.4046	2.472
194.7	180	379.55	352.8	845.7	1198.5	0.4256	2.349
204.7	190	383.75	357.3	841.9	1199.2	0.4467	2.239
214.7	200	387.78	361.7	838.3	1200	0.4677	2.138
224.7	210	391.67	365.9	834.7	1200.6	0.4887	2.046
234.7	220	395.44	369.9	831.2	1201.1	0.5098	1.962
244.7	230	399.06	373.9	827.8	1201.7	0.5308	1.884
254.7	240	402.59	377.7	824.5	1202.2	0.5519	1.812
264.7	250	406	381.3	821.2	1202.5	0.5730	1.745
274.7	260	409.32	385	817.9	1202.9	0.5941	1.683
284.7	270	412.56	388.5	814.8	1203.3	0.6151	1.626
294.7	280	415.70	391.9	811.6	1203.5	0.6362	1.572
304.7	290	418.75	395.3	808.4	1203.7	0.6575	1.521
314.7	300	421.74	398.7	805.4	1204.1	0.6785	1.474
334.7	320	427.49	405	799.6	1204.6	0.7209	1.387
354.7	340	432.98	411	793.9	1204.9	0.7634	1.310
374.7	360	438.24	416.8	788.4	1205.2	0.8060	1.241
394.7	380	443.28	422.4	782.9	1205.3	0.8489	1.178
414.7	400	448.13	427.9	777.6	1205.5	0.8919	1.121
514.7	500	470	452.8	752.3	1205.1	1.109	0.9016
614.7	600	488.8	474.6	728.8	1203.4	1.332	0.7509
714.7	700	505.41	494.2	706.8	1201	1.559	0.6412
814.7	800	520.33	512.3	685.9	1198.2	1.793	0.5576
1,014.7	1,000	546.37	544.7	646.3	1191	2.281	0.4383
1,214.7	1,200	568.75	573.6	608.7	1182.3	2.801	0.3570
1,514.7	1,500	597.50	613	554	1167	3.660	0.2732
2,014.7	2,000	636.82	673	461	1134	5.377	0.1860
3,014.7	3,000	696.08	805.7	209.3	1015	11.930	0.0838
3,207.4	3,193	705.47	902.7	0	902.7	19.660	0.0509

TABLE 41.3

TOTAL HEAT OF SUPERHEATED STEAM
(Izart¹ - after Mollier)

This table gives the total heat A of eqn. (41.1) using the same nomenclature. p' = corresponding approximate gauge pressure, in kg/cm^2

A. METRIC UNITS

p	9	11	13	17	21	26	31	41	51	61
p'	8	10	12	16	20	25	30	40	50	60
$T^\circ\text{C}$										
200	677.5	674.9	671.8							
210	683	680.8	678.2	672.5						
220	688.4	686.3	684	679.2	673.7					
230	693.5	691.6	689.6	685.3	680.6	673.8				
240	698.7	697	695.2	691.3	687.2	681.2	674.6			
250	703.8	702.2	700.6	697	693.2	688.1	682.4			
260	709	707.5	706	702.7	699.2	694.6	689.6	677.8		
270	714	712.6	711.2	708.1	705	700.8	696.3	686.1	673.6	
280	719	717.7	716.3	713.5	710.6	706.8	702.8	693.6	682.8	670.1
290	724.1	722.8	721.5	718.9	716.2	712.7	709	700.7	691.3	680.3
300	729.2	727.9	726.7	724.3	721.8	718.4	715	707.5	699.1	689.5
310	734.3	733.1	731.9	729.6	727.2	724.1	721	714.1	706.5	698
320	739.3	738.2	737.1	734.8	732.6	729.7	726.8	720.4	713.5	705.8
330	744.3	743.2	742.2	740	737.9	735.2	732.5	726.6	720.2	713.3
340	749.4	748.4	747.4	745.4	743.4	740.8	738.2	732.7	726.7	720.5
350	754.4	753.5	752.5	750.6	748.7	746.3	743.7	738.5	733	727.2
360	759.4	758.5	757.6	755.8	754	751.7	749.3	744.3	739.2	733.8
370	764.5	763.7	762.8	761.1	759.3	757.1	754.9	750.1	745.4	740.4
380	769.6	768.7	767.9	766.2	764.6	762.5	760.3	755.9	751.5	746.8
390	774.6	773.8	773	771.4	769.8	767.8	765.7	761.6	757.4	753
400	779.6	778.9	778.1	776.6	775.1	773.1	771.2	767.3	763.2	759.1
410	784.8	784.1	783.3	781.9	780.4	778.6	776.7	772.9	769.1	765.2
420	789.9	789.2	788.5	787.1	785.7	783.9	782.1	778.5	774.8	771.1
430	795	794.3	793.6	792.3	790.9	789.2	787.5	784.1	780.5	777
440	800.1	799.4	798.8	797.5	796.2	794.6	792.9	789.7	786.2	783
450	805.2	804.5	803.9	802.7	801.4	799.9	798.3	795.2	792	788.8
460	810.2	809.6	809	807.8	806.6	805.1	803.6	800.6	797.5	794.5
470	815.3	814.8	814.2	813	811.9	810.4	809	806.1	803.2	800.2
480	820.5	820	819.4	818.3	817.2	815.8	814.4	811.6	808.8	806
490	825.7	825.2	824.6	823.6	822.5	821.2	819.8	817.1	814.5	811.7
500	838.8	830.3	829.7	828.8	827.7	826.4	825.2	822.6	820	817.4
510	836	835.5	835	834.1	833	831.8	830.6	828.1	825.5	823
520	841.2	840.7	840.2	839.3	838.3	837.1	835.9	833.5	831	828.6
530	846.4	846	845.5	844.6	843.7	842.6	841.4	839	836.6	834.4
540	851.6	851.2	850.8	849.9	849	847.9	846.8	844.5	842.2	840
550	856.8	856.4	856	855.1	854.3	853.2	852.1	849.9	847.8	845.6

B. BRITISH UNITS (A IN B.T.U./lb.)

p (p.s.i.a.)	P (p.s.i.g.)	$T^{\circ}F$						
		400	500	600	700	800	900	1000
114.7	100	1226.0	1277.0	1327.6	1378.0	1428.0	1479.6	1531.0
134.7	120	1223.0	1275.7	1326.3	1376.8	1427.6	1478.8	1530.6
154.7	140	1219.8	1273.6	1324.8	1375.8	1426.8	1478.1	1530.1
174.7	160	1216.2	1271.7	1323.3	1374.6	1425.8	1477.4	1529.6
194.7	180	1212.6	1269.6	1321.9	1373.6	1425.1	1476.7	1528.9
214.7	200	1208.7	1267.6	1320.4	1372.5	1424.3	1475.9	1528.4
264.7	250		1262.3	1316.9	1369.7	1422.0	1474.3	1526.9
314.7	300		1256.7	1313.1	1367.0	1419.8	1472.5	1525.6
364.7	350		1250.9	1309.3	1364.1	1417.6	1470.8	1524.1
414.7	400		1244.8	1305.5	1361.4	1415.4	1469.1	1522.7
464.7	450		1238.0	1301.4	1358.4	1413.2	1467.2	1521.3
514.7	500		1230.8	1297.4	1355.5	1411.0	1465.5	1520.1
614.7	600		1214.5	1288.8	1349.6	1406.6	1462.0	1517.0
814.7	800			1269.8	1337.3	1397.5	1454.9	1511.3

TABLE 41.4 A

SPECIFIC VOLUME OF SUPERHEATED STEAM (METRIC UNITS)

- p = absolute pressure of the steam, in kg/cm²
- p' = approximate gauge pressure of steam, in kg/cm²
- t = corresponding saturation temperature at p , in °C
- T = temperature of the superheated steam, in °C
- v = specific volume of steam at T °C and pressure p , in m³/kg

P	p'	$t^{\circ}C$	<i>Sp. vol. v in m³/kg at a temperature T (°C) of:</i>									
			t	150	200	250	500	350	400	450	500	
1	0	99.1	1.725	1.976	2.215	2.453	2.691	2.927	3.164	3.400	3.636	
2	1	119.6	0.902	0.980	1.102	1.222	1.342	1.461	1.580	1.698	1.816	
3	2	132.9	0.617	0.647	0.731	0.812	0.892	0.972	1.052	1.131	1.210	
4	3	142.9	0.471	0.481	0.545	0.607	0.668	0.728	0.788	0.847	0.907	
5	4	151.1	0.382		0.434	0.484	0.533	0.581	0.629	0.677	0.725	
6	5	158.1	0.321		0.359	0.402	0.443	0.484	0.524	0.564	0.604	
7	6	164.2	0.278		0.306	0.343	0.379	0.414	0.448	0.483	0.517	
8	7	169.6	0.245		0.267	0.299	0.330	0.361	0.392	0.422	0.452	
9	8	174.5	0.219		0.235	0.265	0.293	0.321	0.348	0.375	0.401	
10	9	179	0.198		0.210	0.237	0.263	0.288	0.313	0.337	0.361	
11	10	183.2	0.181		0.190	0.215	0.239	0.261	0.284	0.306	0.328	
12	11	187.1	0.166		0.173	0.196	0.218	0.239	0.260	0.280	0.300	
13	12	190.7	0.154		0.159	0.181	0.201	0.220	0.239	0.258	0.277	
14	13	194.1	0.144		0.146	0.167	0.186	0.204	0.222	0.240	0.257	
15	14	197.4	0.134		0.136	0.155	0.173	0.190	0.207	0.223	0.240	
16	15	200.4	0.126			0.145	0.162	0.178	0.194	0.209	0.225	
17	16	203.3	0.119			0.136	0.152	0.167	0.182	0.197	0.211	
18	17	206.1	0.113			0.128	0.143	0.158	0.172	0.186	0.199	
19	18	208.8	0.107			0.120	0.135	0.149	0.162	0.176	0.189	
20	19	211.4	0.102			0.114	0.128	0.141	0.154	0.167	0.179	
21	20	213.8	0.0968			0.108	0.122	0.134	0.147	0.159	0.170	
22	21	216.2	0.0925			0.103	0.116	0.128	0.140	0.151	0.162	
23	22	218.5	0.0886			0.0978	0.110	0.122	0.133	0.144	0.155	
24	23	220.7	0.0849			0.0932	0.105	0.117	0.128	0.138	0.149	
25	24	222.9	0.0816			0.0890	0.101	0.112	0.123	0.133	0.143	
26	25	225	0.0785			0.0852	0.0968	0.107	0.118	0.127	0.137	
27	26	227	0.0756			0.0817	0.0930	0.103	0.113	0.123	0.132	
28	27	229	0.0729			0.0783	0.0894	0.0994	0.109	0.118	0.127	
29	28	230.9	0.0704			0.0752	0.0860	0.0958	0.105	0.114	0.123	
30	29	232.8	0.0680			0.0723	0.0829	0.0924	0.101	0.110	0.118	
31	30	234.6	0.0658			0.0696	0.0800	0.0893	0.0979	0.106	0.115	
36	35	243	0.0566			0.0582	0.0678	0.0761	0.0837	0.0910	0.0982	
41	40	250.6	0.0495				0.0585	0.0661	0.0730	0.0795	0.0859	
51	50	263.9	0.0394				0.0454	0.0520	0.0578	0.0632	0.0685	
61	60	275.4	0.0325				0.0364	0.0425	0.0476	0.0523	0.0568	

TABLE 41.4 B

SPECIFIC VOLUME OF SUPERHEATED STEAM (BRITISH UNITS)

- p = absolute pressure of steam, in p.s.i.a.
 p' = gauge pressure, in p.s.i.g.
 t = temperature of saturated steam at pressure p , in °F
 T = temperature of the superheated steam, in °F
 v = specific volume of the steam at T °F, in cu.ft./lb.

P	p'	t °F	<i>Sp. vol. v in cu.ft./lb. at a temperature T (°F) of:</i>									
			t	300	400	500	600	700	800	900	W00	
14.7	0	212	26.8	30.5	34.7	38.8	42.9	46.9	51.0	55.1	59.1	
24.7	10	239.4	16.5	18.1	20.6	23.0	25.5	27.9	30.3	32.7	35.2	
34.7	20	258.8	12.0	12.8	14.6	16.3	18.1	19.8	21.6	23.3	25.0	
54.7	40	286.7	7.83	8.00	9.18	10.3	11.4	12.6	13.7	14.8	15.9	
74.7	60	307.3	5.84		6.67	7.51	8.35	9.17	9.98	10.8	11.6	
94.7	80	323.9	4.67		5.22	5.90	6.56	7.22	7.86	8.50	9.14	
114.7	100	337.9	3.89		4.28	4.85	5.40	5.94	6.48	7.01	7.54	
134.7	120	350	3.34		3.61	4.11	4.58	5.05	5.51	5.96	6.42	
154.7	140	360.9	2.93		3.12	3.56	3.98	4.39	4.79	5.19	5.58	
174.7	160	370.6	2.61		2.74	3.13	3.51	3.88	4.23	4.59	4.94	
194.7	180	379.5	2.35		2.42	2.80	3.14	3.47	3.79	4.11	4.43	
214.7	200	387.8	2.14		2.19	2.53	2.84	3.14	3.43	3.72	4.01	
234.7	220	395.4	1.96		1.98	2.30	2.59	2.87	3.14	3.40	3.66	
254.7	240	402.6	1.81			2.11	2.38	2.63	2.89	3.13	3.37	
274.7	260	409.3	1.68			1.94	2.20	2.44	2.67	2.90	3.13	
294.7	280	415.7	1.57			1.80	2.04	2.27	2.49	2.70	2.91	
314.7	300	421.7	1.47			1.68	1.90	2.12	2.32	2.52	2.73	
334.7	320	427.5	1.39			1.57	1.78	1.98	2.18	2.37	2.56	
354.7	340	433	1.31			1.47	1.68	1.87	2.06	2.23	2.41	
374.7	360	438.2	1.24			1.38	1.58	1.77	1.94	2.11	2.28	
394.7	380	443.3	1.18			1.30	1.50	1.67	1.84	2.00	2.16	
414.7	400	448.1	1.12			1.23	1.42	1.59	1.75	1.90	2.06	
464.7	450	459.5	1.000			1.08	1.25	1.41	1.55	1.69	1.83	
514.7	500	470	0.902			0.956	1.12	1.26	1.40	1.49	1.65	
614.7	600	488.8	0.751			0.772	0.921	1.05	1.16	1.27	1.37	
714.7	700	505.4	0.641				0.775	0.887	0.988	1.08	1.18	
814.7	800	520.3	0.558				0.664	0.768	0.859	0.945	1.03	

42. Steam Turbines

CONCISE THEORY OF THE STEAM TURBINE

Fundamental equation

The equation which constitutes the basis of all calculations in fluid mechanics is that of Bernoulli:

$$dz + v dp + d \left(\frac{V^2}{2g} \right) = 0 \quad (42.1)$$

It simply expresses the conservation of energy along a thin stream of fluid in motion:

- z = head of the fluid
- v = specific volume of the fluid
- p = pressure
- V = velocity
- g = acceleration due to gravity.

For gases, and in particular for steam, the integral of dz is always negligible:

(1) Because in the turbine everything takes place at the same height, which is that of the turbine,

(2) Because v , which is small in hydraulics (for water, $v = 0.001$ mVkg), is high for steam ($v = 1,650$ mVkg for saturated steam even at 100°C).

In limiting ourselves to steam, and taking no account of z , we obtain the formula of Barré de Saint-Venant:

$$A \frac{V^2}{2g} = - \int_{p_1}^{p_2} v dp \quad (42.2)$$

which applies to perfect fluids and does not take friction into account. The indices 1 and 2 apply, respectively, to the initial and final states of the steam in the change under consideration.

Relation between heat drop and steam velocity

Let us consider a transformation of steam from one state A to another state B , for instance: passage of the steam in (for the sake of simplification) a single-stage turbine. Let us draw a dynamic diagram (Fig. 42.1). The Saint-Venant formula (42.2) gives us:

$$\frac{V_B^2 - V_A^2}{2g} = - \int_{p_1}^{p_2} v dp = W \quad (42.3)$$

V_A = velocity of the steam in state A (before the transformation, for instance upstream of the wheel)

V_B = velocity of the steam in state B (after the transformation, for instance issuing from the wheel)

W - area $BbaA$ = work done by the steam in the transformation.

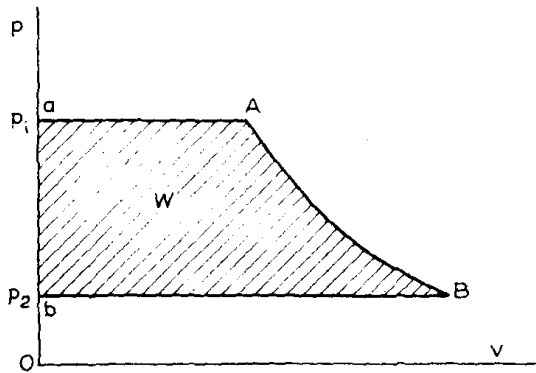


Fig. 42.1. Transformation of steam. $p-v$ diagram.

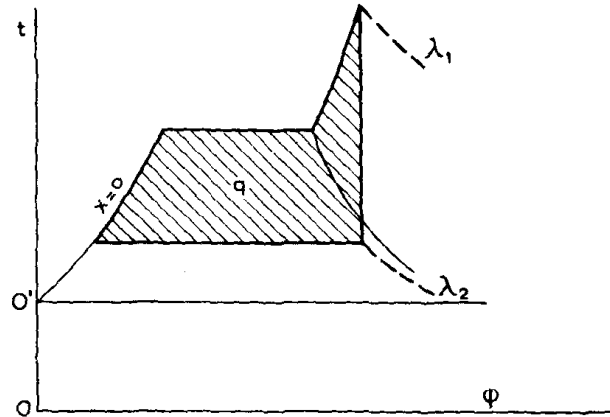


Fig. 42.2. Transformation of steam. Entropy diagram.

In the case of the turbine, starting from the nozzle-chest, we may always neglect V_A^2 (steam velocity in the nozzle-chest) in comparison with V_B^2 , and we obtain:

$$V^2 = 2gW \tag{42.4}$$

V = velocity of the steam producing work W .

On the entropic diagram, the line of zero volume is practically the line $x = 0$. The cross-hatched area (Fig. 42.2) represents the heat used q :

$$W = Eq \tag{42.5}$$

E = mechanical equivalent of the kcal = 427 kg m (1 B.T.U. = 778.3 ft.lb.)

$q = \lambda_1 - \lambda_2$, which can be read on the diagram.

We have then:

$$\begin{aligned} V &= \sqrt{2gEq} = \sqrt{2 \times 9.8 \times 427 (\lambda_1 - \lambda_2)} \\ &= 91.5 \sqrt{\lambda_1 - \lambda_2} \quad (224 \sqrt{\lambda_1 - \lambda_2}) \end{aligned} \tag{42.6}$$

This is a theoretical equation and relates to a perfect gas. As steam is not a perfect gas, we must write in practice:

$$V \approx 91.5 \sqrt{a (\lambda_1 - \lambda_2)} \quad (224 \sqrt{a (X_1 - X_2)}) \tag{42.7}$$

V = steam velocity, in m/s (ft.s)

X_1 = total heat of the steam, before the wheel, in kcal/kg (B.T.U./lb.)

X_2 = total heat of the steam, leaving the wheel, in kcal/kg (B.T.U./lb.)

with:

$a = 0.94$ for a first wheel, simple or multiple

$a = 0.98$ for the second and following wheels

$a = 0.97$ to calculate the speed at the throat of a nozzle.

In order to take into account the friction in a nozzle, the same formula is used, with $a = 0.85$.

This formula (42.7) is one of the most important in calculations relating to steam turbines.

Output of a nozzle

The Saint-Venant formula allows us to study the flow of steam through an elongated opening linking two compartments where pressures are respectively p_1 upstream and p_2 downstream.

The fine streams of flowing steam take the shape shown in Fig. 42.3, and form a constriction.

It may be shown that pressure p_0 at the constriction, for perfect gases, depends only on the upstream pressure p_1 .

$$p_0 = 0.52p_1 \tag{42.8}$$

But steam is not a perfect gas and in practice we have:

$$\begin{aligned} p_0 &= 0.58p_1 \text{ for saturated steam} \\ p_0 &= 0.54p_1 \text{ for superheated steam} \end{aligned} \tag{42.9}$$

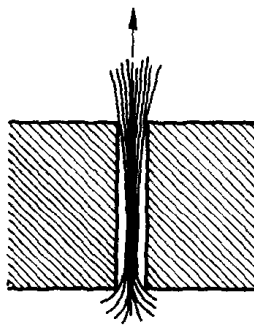


Fig. 42.3. Flow of steam.

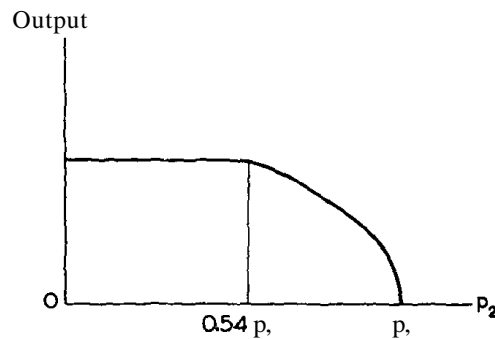


Fig. 42.4. Output of a nozzle.

We see that superheated steam is closer to being a perfect gas than is saturated steam.

If we draw a graph plotting the steam output of the nozzle as a function of the downstream pressure (Fig. 42.4), we shall see that the output remains constant while:

$$p_2 \leq 0.54p_1$$

then decreases rapidly and falls to 0 when $p_2 = p_1$.

It follows from this that, in a turbine, the steam output, and consequently the power, depends only on the characteristics of the upstream steam, p_1 and v_1 , and on the sections at the throats of the nozzles before the first wheel.

Shape of the nozzle

The form given to the nozzle is that which the flowing steam tends to take. Between the inlet and the throat, the form is roughly that of a cone with an apical angle of 7° . This angle is an optimum: a smaller angle would make a nozzle too elongated, thus increasing friction; a wider angle would not be followed by the steam.

The section at the throat approximates to a square. If we do not have exactly a square, the square should be found between the throat and the exhaust.

IMPULSE TURBINES AND REACTION TURBINES

The Bernoulli equation (42.1), discarding the negligible first term $d\mathbf{z}$, shows that the steam produced by the boilers has two power-generating properties: its pressure and its velocity.

Since the steam reaches the nozzle-chest with practically a negligible velocity, all its energy is in reality in its pressure (if it is superheated the factor v of the term $v dp$ obviously shows that the energy contained in a given weight of that steam at a given pressure increases with its superheat).

The energy contained in the steam under the form of pressure can be used in two ways:

- (a) By converting it entirely to velocity
- (b) By using part of it directly under the form of pressure, and by converting the rest into velocity.

Each of these two methods leads to a special type of turbine:

- (a) That in which all the energy is used under the form of velocity: this is the *impulse turbine*.
- (b) That in which the energy is used partly under the form of pressure and partly under the form of velocity: this is the *reaction turbine*.

The impulse turbine is so called because it is actuated by the impulse or impact due to the velocity of the steam. The reaction turbine gets its name from the fact that the expansion of the steam in the moving blades causes a *reaction* on them; this reaction contributes to their movement, as does the impact due to the steam velocity. In other words: while it is the steam pressure that is used in all turbines, some are driven by the impulse of the speed imparted to the steam by the pressure drop; these are impulse turbines. Others rotate under the effects of (a) the speed given to the steam by *part* of the pressure drop and (b) the reaction caused in the wheels by the expansion due to the fraction of the pressure drop *not* transformed into speed: these are the *impulse and reaction* (or action and reaction) turbines, more briefly termed reaction turbines.

This distinction is not at all purely academic: it imposes itself on the designer, from one end of the turbine to the other. In the impulse turbine, the total section of the passage through the fixed distributor blading will have to increase in a given distributor as well as increasing in the successive distributors. The exit of a distributor will have to be calculated

for the same pressure as the entry of the following distributor, however, while the steam will traverse the wheel placed in between these two distributors at a constant pressure.

Impulse turbines

Construction of the turbine. According to what we have just said, the wheels of an impulse turbine are not designed to withstand a difference in pressure. That is the reason why they are perforated, so that if, unexpectedly, there happened to be a reaction, the pressure could at once equalise on the two sides. However, the last wheels are not perforated, because:

- (a) The difference in pressure is small (*cf.* Table 42.2),
- (b) Reaction on the last wheels is not a drawback, but rather an advantage (*cf.* p. 1063),
- (c) Those wheels have longer blades and are submitted to a higher stress. Therefore it is better not to perforate them.

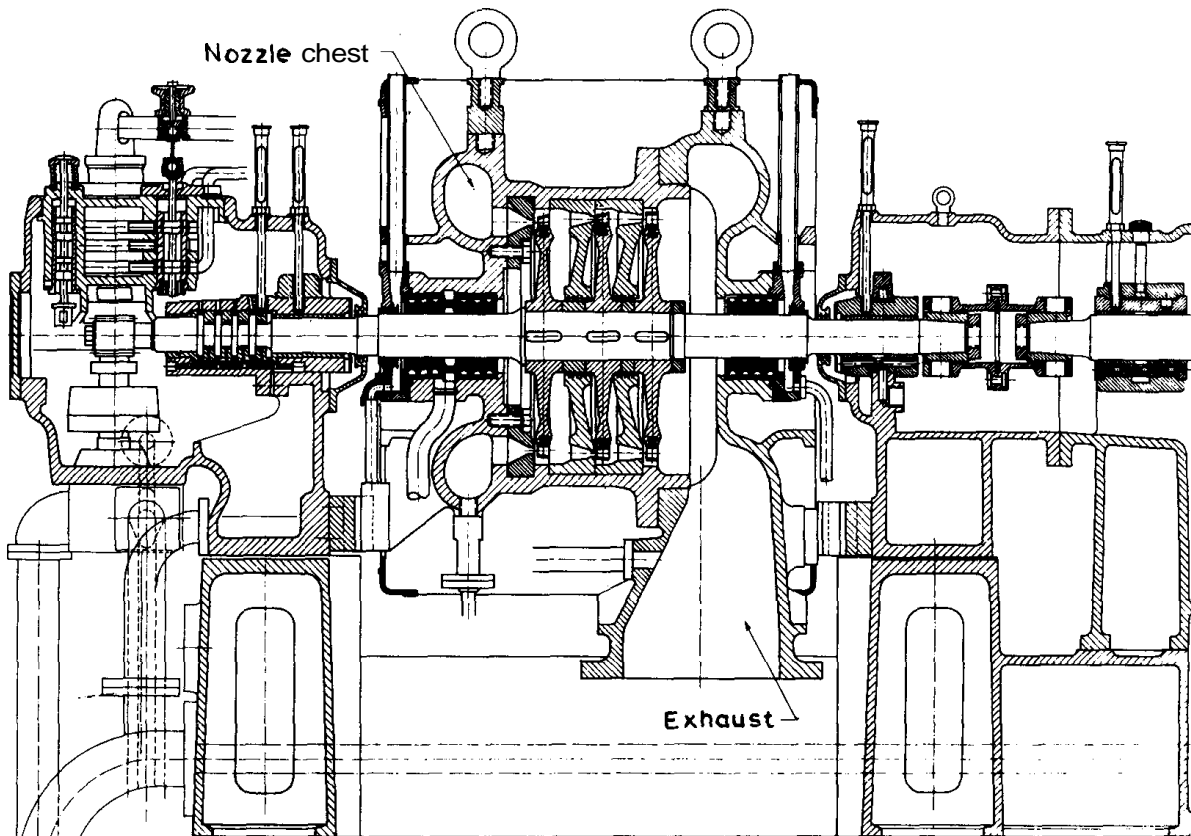


Fig. 42.5. Back-pressure steam turbine (Fives Cail - Babcock).

Figure 42.5 gives a sectional view of a turbine. The h.p. steam comes in at the admission nozzle-chest and passes through the nozzles onto the bladings of a first wheel. Issuing from that wheel, it is taken up by fixed blading, which gives it the desired direction and directs it on to a second wheel, and so on (Fig. 42.6). An important point in the construction of a turbine will then be the correct design of the blading; it has to receive or guide the steam

with the minimum of shocks, eddies, and friction. The material of the blading is steel with 12% chromium.

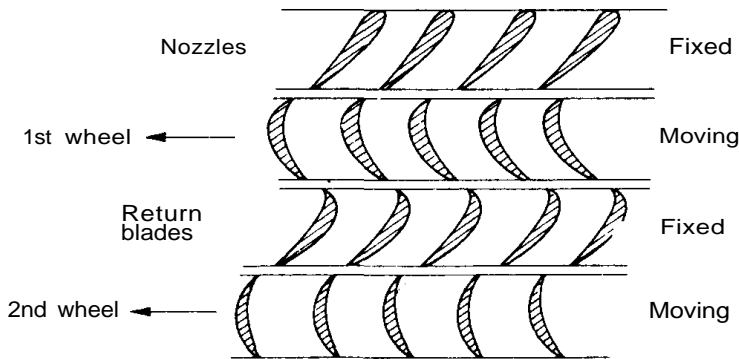


Fig. 42.6. Blading.

Efficiency of a wheel. Let

- U = mean peripheral speed of the blading due to the rotation
- V = absolute steam velocity
- W — relative steam velocity, relative to the blading
- α = angle between U and V
- β = angle between U and W .

Let us use index 1 for the entry to the wheel, and index 2 for the exit (Fig. 42.7). As there is no expansion and as we neglect friction: $W_2 = W_1$. Besides, $U_2 = U_1$ since we are considering a single wheel through which the steam passes in a direction parallel to the shaft.

Finally, we consider generally: $\beta_2 = 180 - \beta_1$.

Let us draw the figure again, transferring O_2 to O_1 , and W_2 to W_1 (Fig. 42.8).

We then have:

$$V_2^2 = V_1^2 + 4U^2 - 4UV_1 \cos \alpha_1 \tag{42.10}$$

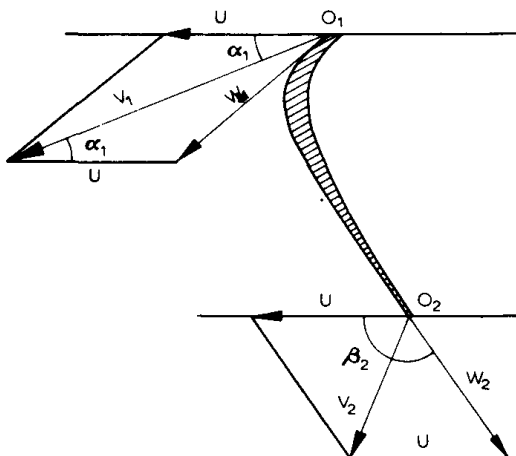


Fig. 42.7. Velocity of steam at entry and exit of a wheel.

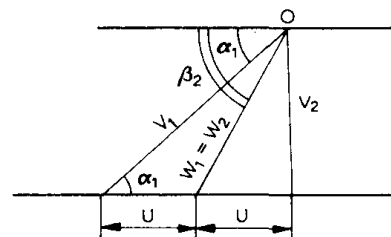


Fig. 42.8. Triangle of velocities.

Now, the wheel receives the energy: $(CV_1^2)/2g$ and releases without work: $CV_2^2/2g$.
 C = output of the steam from the turbine.

Thus the efficiency is:

$$e = \frac{V_1^2 - V_2^2}{V_1^2} \tag{42.11}$$

and we obtain, replacing the numerator by its value taken from (42.10):

$$e = \frac{4U(V_1 \cos \alpha_1 - U)}{V_1^2} \tag{42.12}$$

For a given speed V_1 and a given angle α_1 , let us try and find the value of U which will give us the maximum efficiency. We notice that the sum of the 2 variable factors in the numerator is a constant:

$$U + (V_1 \cos \alpha_1 - U) = V_1 \cos \alpha_1$$

therefore we have a maximum when the 2 terms are equal or:

$$U = V_1 \cos \alpha_1 - U$$

or

$$\frac{U}{V_1} = \cos \alpha_1 \tag{42.13}$$

Comments: (1) *Normal delivery.* We may write:

$$\cos \alpha_1 = \frac{2U}{V_1} \tag{42.14}$$

which shows that, in Fig. 42.8, V_2 is at right-angles to the movement of the wheels.

(2) *Maximum efficiency.* We get maximum efficiency for:

$$U = \frac{V_1 \cos \alpha_1}{2} \tag{42.15}$$

Its value is:

$$e_M = \frac{4 \cdot \frac{V_1^2 \cos^2 \alpha_1}{4}}{V_1^2} = \cos^2 \alpha_1 \tag{42.16}$$

We see then that the maximum efficiency corresponds to the smallest angle α_1 . But there is a limit to the diminution of the entrance angle, for, with $\alpha_1 = 0$, the steam would not enter the wheel. In general, α_1 varies between 12° and 20° for the first wheel. Quite often we have $\alpha_1 = 16^\circ$. It increases in the following wheels, up to 35° and even 40° for the last distributor.

(3) *The ξ rule.* Let us suppose:

$$\frac{U}{V_1} = \xi \quad (42.17)$$

Condition (42.13) will then be expressed by the following rule: the ratio ξ of the peripheral speed of the wheel to the entrance velocity of the steam to its blades has an optimum value $= (\cos \alpha_1)/2$.

We have seen that α_1 could not be smaller than 12° . Theoretically, then, the optimum ξ can not exceed 0.49. Besides, it is advantageous to increase the expansion, and, since a function varies little in the vicinity of its maximum, it is possible to get away slightly from the optimum ratio without appreciably changing the efficiency. In practice, therefore, we shall not exceed:

$$\xi = 0.45.$$

It is the rotational speed U which limits the designer, on account of the stress in the steel due to the centrifugal force. The smaller ξ , the greater is V , and consequently the heat drop q which is proportional to V^2 (cf. eqn. 42.6). We shall then come lower in the entropic diagram, to a lower pressure p_2 . A greater drop in each wheel will mean fewer wheels, with downstream temperatures and pressures decreasing more rapidly, therefore fewer metal expansions and leakages. For that reason, the theoretical efficiency is sometimes sacrificed and one comes down to $\xi = 0.30$. If it is not desired to let the efficiency suffer too much, the value is preferably kept in the vicinity of: $\xi = 0.40$. In the sugar industry, the turbines are back-pressure turbines in which the efficiency is not of primary importance, since the manufacture of sugar requires more exhaust steam than the turbine can produce. Therefore the turbines here will generally have a ξ varying from 0.30 to 0.40.

N.B. The Greek letter ξ is pronounced "xi".

Internal efficiency of the wheel. The efficiency we have just considered, expressed by formula (42.11), takes into account only the residual loss $V_2^2/2g$. It is also possible to take into account the losses undergone by the steam, through shocks, friction and turbulence, during its passage in the wheel. They amount to:

$$C \cdot \frac{W_1^2 - W_2^2}{2g} \quad (42.18)$$

C = steam output of the turbine.

The velocity U being the same at the entry and at the exit, we can draw the triangle of velocities at entry and the triangle at exit on the same vector $AB = U$ (Fig. 42.9). Those two triangles give us:

$$\left. \begin{aligned} W_1^2 &= V_1^2 + U^2 - 2UV_1 \cos \alpha_1 \\ W_2^2 &= V_2^2 + U^2 - 2UV_2 \cos \alpha_2 \end{aligned} \right\} \quad (42.19)$$

Hence:

$$V_1^2 - V_2^2 - (W_1^2 - W_2^2) = 2U(V_1 \cos \alpha_1 - V_2 \cos \alpha_2) \quad (42.20)$$

Multiplying by $C/2g$ we have:

$$C \cdot \frac{V_1^2}{2g} - C \cdot \frac{V_2^2}{2g} - C \cdot \frac{W_1^2 - W_2^2}{2g} = \frac{C}{2g} \cdot 2U(V_1 \cos \alpha_1 - V_2 \cos \alpha_2) \quad (42.21)$$

Now:

$$C \cdot \frac{V_1^2}{2g} = \text{power provided at the wheel}$$

$$C \cdot \frac{V_2^2}{2g} = \text{loss through residual speed}$$

$$C \cdot \frac{W_1^2 - W_2^2}{2g} = \text{losses through shocks and friction.}$$

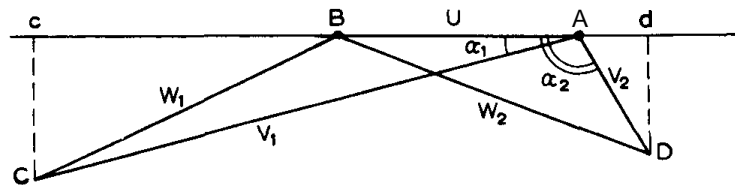


Fig. 42.9. Internal efficiency of the wheel.

The first term of eqn. (42.21) represents the power remaining after taking into account these two losses. Relating it to the power provided for, we shall have the "internal efficiency" of the wheel:

$$e_i = \frac{2U(V_1 \cos \alpha_1 - V_2 \cos \alpha_2)}{V_1^2} \quad (42.22)$$

It is also called: "hydraulic efficiency" or "thermodynamic efficiency".
On the diagram (Fig. 42.9), we can read it graphically:

$$e_i = \frac{2U \times \overline{cd}}{V_1^2} \quad (42.23)$$

Necessity of having several wheels. The normal heat drop in a turbine, between the boilers and the exhaust main, is about 100–150 kcal/kg (180-270 B.T.U./lb.) The most frequent value would be:

$$\begin{array}{r} \lambda_A (23 \text{ kg/cm}^2, 350^\circ\text{C}) = 747 \text{ kcal/kg} \\ \lambda_B (0.5 \text{ kg/cm}^2, x = 0.93) = 607 \text{ kcal/kg} \\ \hline \lambda_A - \lambda_B = 140 \text{ kcal/kg} \end{array}$$

$$\left(\begin{array}{r} \lambda_A (327 \text{ p.s.i.g.}, 662^\circ\text{F}) = 1,344 \text{ B.T.U./lb.} \\ \lambda_B (7 \text{ p.s.i.g.}, x = 0.93) = 1,092 \text{ B.T.U./lb.} \\ \hline \lambda_A - \lambda_B = 252 \text{ B.T.U./lb.} \end{array} \right)$$

If we wished to use this all on one wheel, we should reach the following steam velocity:

$$V = 91.5 \sqrt{0.94 \times 140} = 1,050 \text{ m/s} \quad (3,447 \text{ ft./s})$$

which would require a peripheral speed of:

$$V = \zeta V = 0.40 \times 1,050 = 420 \text{ m/s} \quad (1,379 \text{ ft./s})$$

We cannot hope, by a long way, however, to realise such a speed. The highest speeds reached in small turbines are about 300 m/s (1,000 ft./s), and generally steps are taken to keep to moderate speeds, 160-220 m/s (500-720 ft./s). Allen¹ generally adopts 190 m/s. For general figures, it may be commented in passing that the maximum speeds used in turbines of very high power are of the order of 570 m/s (1,870 ft./s).

In order to remain under the conditions of maximum efficiency, *i.e.* in order to observe the ζ rule, we may follow two methods.

1st Method: pressure compounding. If the turbine functions between the pressures p_A and p_B , the total pressure drop will be divided into x stages, each stage consisting of a ring of fixed distributor blading, where all the expansion will take place, and a wheel through which the steam will pass at a constant pressure.

How should these partial expansions be apportioned?

Rule. Efficiency is at a maximum when all the wheels have the same power.

Under those conditions, each wheel will have to absorb q/x kcal or B.T.U., and each set of distributor blading will deliver the steam at a velocity:

$$V_x = 91.5 \sqrt{\alpha \frac{q}{x}} \quad \left(224 \sqrt{\alpha \frac{q}{x}} \right) \quad (42.24)$$

and the optimum peripheral speed will be: $U_x = \xi V_x$.

Whence:

$$\frac{U_x}{U} = \frac{V_x}{V} = \frac{91.5 \sqrt{\alpha \frac{q}{x}}}{91.5 \sqrt{\alpha q}} = \frac{1}{\sqrt{x}} \tag{42.25}$$

With four wheels, it will therefore be possible to use a speed only half of that necessary for a single wheel.

Determination of the number of wheels. Let:

x = the total number of wheels in the turbine

q = the total heat drop: $\lambda_A - \lambda_B$

q_1 = the heat drop in the first wheel

q_n = the heat drop in each of the following wheels.

It is advantageous to have the same diameter and the same ξ for all the wheels of the turbine, and this is easy to realise in a back-pressure turbine suitable for the sugar factory. We shall then have:

$$V = \frac{U}{\xi} = \frac{\pi n}{60} \cdot \frac{D}{\xi} = \text{Constant} \tag{42.26}$$

n = rotation speed of the turbine, in r.p.m.

D = diameter of the wheels, at the middle of the blading, in m (ft.)

V = steam velocity, in m/s (ft.s).

Since (eqn. 42.7):

$$V = 91.5 \sqrt{0.94 q_1} = 91.5 \sqrt{0.98 q_n} \quad (224 \sqrt{0.98 q_n})$$

with:

$$q_1 = q - (x - 1)q_n$$

We then require:

$$0.94 [q - (x - 1)q_n] = 0.98 q_n$$

Hence:

$$q = (x + 0.04)q_n \tag{42.27}$$

Since:

$$U = \xi V = \xi 91.5 \sqrt{0.98 q_n} = \xi 91.5 \sqrt{0.98 \frac{q}{x + 0.04}}$$

Hence:

$$x + 0.04 = \frac{8,205 (\lambda_A - \lambda_B) \xi^2}{U^2} \quad \left(\frac{149,085 (\lambda_A - \lambda_B) \xi^2}{V^2} \right) \quad (42.28)$$

Rule. We shall give to x the values: 1, 2, 3, 4, 5, etc.; we shall calculate the corresponding values of ξ , and we shall choose the number of wheels x which will give the ξ value nearest to that desired.

2nd Method: velocity compounding. In order to retain the conditions of maximum efficiency, one can also utilise the pressure drop (or part of it) by sending the steam, under a constant pressure (impulse turbine), through two or more successive rings of mobile blading disposed on the same wheel, and separated by fixed rectifying blades (Figs. 42.10 and 16.2).

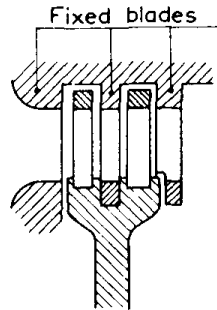


Fig. 42.10. Double wheel.

Characteristics of multiple wheels. A study of this solution, similar to the one we have just made of a single wheel, leads to the following conclusions:

(a) ξ . A multiple wheel with n rings has an optimum ξ of:

$$\xi_n = \frac{\cos \alpha_1}{2n} \quad (42.29)$$

Consequently, if the optimum ξ of a single wheel is 0.42, for instance, that of a double wheel will be 0.21 and that of a triple wheel 0.14:

$$\xi_n = \frac{\xi_1}{n} \quad (42.30)$$

(b) Heat drop: since it is the peripheral speed which sets the limit to the designer, and since that limit is almost the same for a multiple wheel and for a single wheel, we shall need to have (subscript 1 refers to the single wheel and subscript n to the multiple one):

$$U = \xi_1 V_1 = \xi_n V_n = \frac{\xi_1}{n} \cdot V_n \quad (42.31)$$

Hence:

$$V_{r_n} = n V_1 \tag{42.32}$$

or:

$$91.5 \sqrt{\alpha_{r_n}} = n 91.5 \sqrt{\alpha q_1}$$

Hence:

$$q_n = n^2 q_1 \tag{42.33}$$

We can see, then, that a double wheel will absorb 4 times as many heat units as a single wheel. In other words, a double wheel in a turbine will replace four single wheels.

Comment. Of course, this is no longer true unless we take corresponding values for ξ in both cases. It is easy to see, for instance, that a double wheel with $\xi = 0.2$ is only equivalent to 9/4 of a single wheel with $\xi = 0.3$. The efficiency of the latter is then far lower, however.

We can see in passing that it is possible to increase the power of a wheel by accepting a lower efficiency.

(c) Efficiency. The theoretical efficiency of a multiple wheel is the same as that of a single wheel:

$$\eta_n = \eta_1 = \cos^2 \alpha_1 \tag{42.34}$$

(d) Apportioning the power of the different stages. The triangles of velocities show that the powers transmitted by the different stages of a multiple wheel follow in relation to one another the sequence of the successive decreasing odd numbers:

$$\begin{aligned} \text{Double wheel:} & \quad 3 - 1 \\ \text{Triple wheel:} & \quad 5 - 3 - 1 \end{aligned}$$

Friction only aggravates this poor theoretical apportioning.

(e) Field of use of the double wheel. If we compare the total losses in a double wheel and in the four single wheels it replaces, we shall see that the power plays a part: the losses are the same for a given power P_0 . Below P_0 they are smaller for the double wheel; above P_0 , they are greater. Consequently the double wheel is interesting for small powers.

It is generally agreed that:

$$P_0 = 4p \left(\frac{n}{1,000} \right)^3 D^5 = \left(\frac{p \left(\frac{n}{1,000} \right)^3 D^5}{1,370} \right) \tag{42.35}$$

P_0 = limiting power for the use of the double wheel, in h.p.

p = admission pressure of the steam, in kg/cm² gauge (p.s.i.g.)

n = rotational speed of the turbine, in r.p.m.

D = mean diameter of the wheels, in m (ft.).

This formula shows us that if we are to have a double wheel, it will be placed first, since p falls sharply after the first wheel.

The field of the triple wheel is even more limited, and its use is rare in consequence.

(f) Advantages of the double wheel: There is a tendency to use the double wheel even a little outside its proper field, for a power slightly superior to P_0 , even though it means a fall in efficiency of about a $\frac{1}{2}$ or 1 unit. The double wheel offers some advantages:

(1) Considerable expansion in a single wheel, thus immediately and considerably decreasing the pressure at the admission end.

(2) Longer life of the h.p. packing in consequence of that drop in pressure.

(3) Parallel fall in temperature; hence the possibility of smaller clearances and less wear in the rotor.

(4) Rigidity of the wheel, owing to its double ring; hence less risk of warping through an abrupt opening of the steam valve.

(5) Shorter shaft (one wheel instead of four).

Reaction turbines

We have seen (p. 1050) that reaction turbines are turbines in which the expansion of the steam takes place in part in the fixed blading and in part in the mobile blading, thus giving to the latter a supplementary speed.

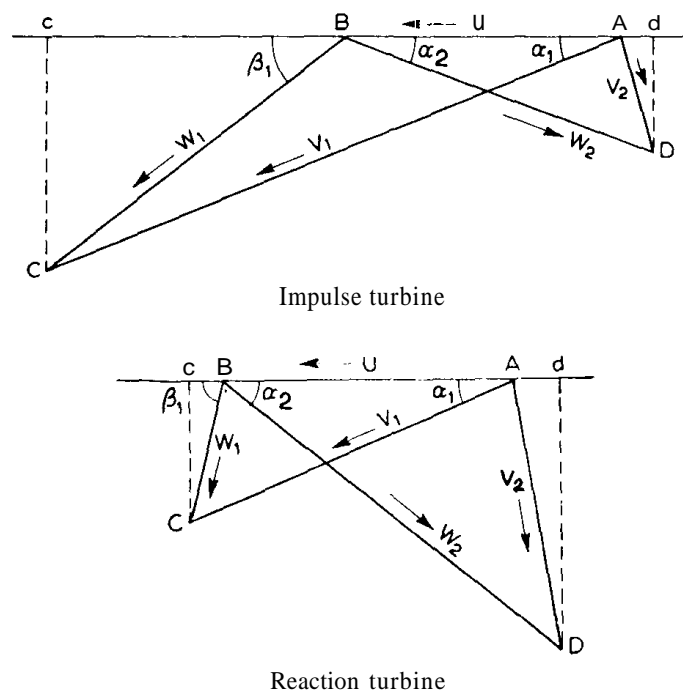


Fig. 42.11. Velocities for impulse and reaction turbines.

Triangle of velocities. The reaction causes rather large differences in the choice of speeds (Fig. 42.11). In the impulse turbines, W_2 is inferior to W_1 owing to the friction of the steam on the blading, and V_2 is much inferior to V_1 , since it is precisely the energy $(V_1^2 - V_2^2)/2g$ which is turned into movement in the wheel.

In the reaction turbine, since the steam expands in the wheel, W_2 is greater than W_1 , and V_2 is much greater than in impulse turbines. The angle β_1 is nearly 90° instead of $20 - 60^\circ$ in impulse turbines.

£ theory and practice show that the efficiency is maximum for $\xi = 0.85$ or thereabouts. If we attribute subscript 0 to the impulse, and the letter without a subscript to the reaction, we see that we have approximately:

$$\xi = 2\xi_0 \tag{42.36}$$

Steam velocity. Consequently we have:

$$\frac{U}{V} = 2 \frac{U_0}{V_0}$$

We know, however, that it is U that sets a limit to the designer, and that the reaction does not determine the strength required of the steel of the turbine. Therefore, $U = U_0$ and:

$$V = \frac{V_0}{2} \tag{42.37}$$

In a reaction turbine, the steam velocity is half that of the corresponding impulse turbine.

Number of wheels. In the reaction turbine, suppose we use half the heat drop under the form of speed. The other half is left for the reaction (this is the proportion generally adopted). If the number of wheels is x , the speed will be given by:

$$V = 91.5 \sqrt{\frac{q}{2x}}$$

and if we compare that turbine with an impulse turbine, functioning under the same heat drop q , we shall have according to the preceding rule (42.37):

$$91.5 \sqrt{\frac{q}{2x}} = \frac{91.5}{2} \sqrt{\frac{q}{x_0}}$$

Hence:

$$x = 2x_0 \tag{42.38}$$

The reaction turbine has twice as many wheels as the corresponding impulse turbine.

Shock and friction. Suppose η expresses the losses through shock and friction. They are given in each wheel by an expression of the form:

$$\eta = kV^2 \quad (42.39)$$

or, for the two turbines we are comparing:

$$17 = kxV^2 \quad \eta_0 = kx_0 V_0^2$$

Hence:

$$\eta = k2x_0 \cdot \frac{V_0^2}{4} = \frac{\eta_0}{2} \quad (42.40)$$

Reaction is thus a smoother and more delicate way of using the steam. This is its big advantage.

Leaks. With the impulse turbine there are leaks at the diaphragms only. With the reaction turbine, we have in addition the leaks on the periphery of the wheels, owing to the difference in pressure between the two sides of the wheel. This kind of leak is measured by r/h (Fig. 42.12).

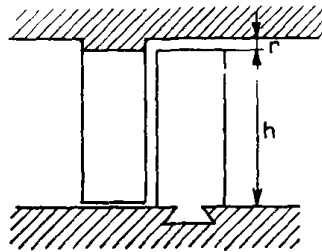


Fig. 42.12. Leaks.

Therefore it is much more important in the **h.p.** stages:

- (1) Because r is then larger owing to the big differences in temperature: from 1 to 1.5 mm.
- (2) Because h is small.

Axial thrust. The axial thrust is very small in an impulse turbine, but in a reaction turbine it becomes very large, thus compelling the designer to take special measures in order to absorb it.

Degree of injection. The reaction, in the first wheel, makes it necessary to have full annular injection, which is prohibitive where small powers are concerned, since it would lead to very short blades. These are never made less than 5 mm, and this is the reason why reaction is not used with small powers — in any case not as soon as the first wheel.

Conclusion. Summing up the advantages and drawbacks, we can see that it is advantageous to have impulse h.p. stages and reaction l.p. stages. Thus:

- (a) One can use partial injection.
- (b) Leaks can be avoided where r/h is greater.
- (c) The axial thrust is avoided in the wheels where the differences in pressure are greatest.
- (d) The reaction improves efficiency in the latter stages.

Ordinary sugar factory turbines are always impulse turbines, at least in the h.p. stages.

Steam consumption

Turbines are generally used to drive alternators in the turbo-alternator sets of the electric power-house; hence we shall consider steam consumption per kWh or per h.p.h. at the terminals of the switch board. It is given by:

$$Q = \frac{3600}{(X - \lambda')\eta e_m e_r e_g} \quad \text{or} \quad \frac{3600}{(\lambda - \lambda')\eta e_m e_r e_g} \tag{42.41}$$

or:

$$Q' = \frac{860}{(X - \lambda')\eta e_m e_r e_g} \quad \left(\frac{3,413}{(\lambda - \lambda')\eta e_m e_r e_g} \right) \tag{42.42}$$

Q = steam consumption of the turbo-alternator set, in kg/h.p.h. (lb./h.p.h.)

Q' = steam consumption of the turbo-alternator set, in kg/kWh (lb./kWh)

X = total heat of the steam at the nozzle-chest, in kcal/kg (B.T.U./lb.)

λ' = total heat of the steam at exhaust, in kcal/kg (B.T.U./lb.)

η = thermodynamic efficiency of the turbine

e_m = mechanical efficiency of the turbine

e_r = efficiency of the reduction gearing, if it is a geared turbine

e_g = efficiency of the alternator or the generator.

Value of e_m . Approximately 0.985.

Value of Q_s . Varies with power: 0.94 – 0.985.

Value of e_r . Varies from 0.97 to 0.985.

Value of η . The thermodynamic efficiency of the turbine depends on:

- (a) The mechanical standard of construction
- (b) Its power; the more powerful the turbine the higher the efficiency
- (c) The adiabatic heat drop; the higher the drop the better the efficiency.

We may use the figures of Table 42.1.

TABLE 42.1

MEAN VALUES OF THERMODYNAMIC EFFICIENCY

Impulse turbines	
Back-pressure turbines	
Turbines with reduction gear and double wheel	0.65
Back-pressure = 43 – 100 p.s.i.g.	0.65-0.70
Back-pressure = 7 – 43 p.s.i.g.	0.70-0.72
Condensing turbines	0.72-0.75
Condensing reaction turbines	0.75 – 0.80
Condensing turbines 3 - 8 MW	0.80
Condensing turbines 20 MW	0.82

In order to obtain the actual steam consumption, we shall have to add to the above value Q :

For losses through condensation: 3 – 5%
 For losses by leaks: 2-3%

The efficiency of the complete power plant is thus, taking the normal average values:

(a) Efficiency of the turbine: $\epsilon_m \epsilon_r \epsilon_g \eta = 0.985 \times 0.98 \times 0.95 \times 0.75 = 0.688$

(b) Losses: $\epsilon_p = 0.936$; $0.688 \times 0.936 = 0.644$

(c) Auxiliaries. If we take into account auxiliary equipment (feed pump, air heater, draught fan) which takes its power from the turbine to allow it to function, without direct use in the factory, we must also include a coefficient $\epsilon_a = 0.935$, and we obtain:

$$\epsilon_v = 0.644 \times 0.935 = 0.6$$

The useful energy is thus primarily only 60% of that contained in the steam. This applies for the turbines driving knives, shredder and mills. If we consider the electric motors of the factory we should also allow for efficiency of transmission lines and mean efficiency of electric motors, giving overall efficiency for electric drive as: $0.6 \times 0.97 \times 0.85 = 0.5$
 Thus the useful power of electric motors is only half of that derived from the steam.

Losses of pressure. We must not lose sight of the losses of pressure between the boiler and the turbine. Varying with the distance, the larger or smaller section of the pipes, and the more or less careful insulation, we shall have to deduct:

For pressure: 10 – 12% of the pressure at the boiler
 For temperature: 5 - 10% of the temperature at the boiler

Steam at 25 kg/cm², 360°C (350 p.s.i., 680°F) at the boilers will arrive at the turbine at 22-23 kg/cm², 340°C (315-320 p.s.i., 644°F). This difference takes into account the fluctuations of pressure at the boilers, which is rarely equal to the rated value, and reaches it only occasionally.

SPECIFICATIONS FOR A TURBINE

We shall proceed to establish the preliminary design of a turbine for a sugar factory, with back-pressure and gearing.

Data

Pressure at entry:	$p_A = 23 \text{ kg/cm}^2 \text{ gauge (325 p.s.i.g.)}$
Temperature of the superheated steam at the entry:	$t_A = 340^\circ\text{C (644}^\circ\text{F)}$
Back-pressure at exhaust:	$p_B = 0.5 \text{ kg/cm}^2 \text{ gauge (7 p.s.i.g.)}$
Power required at the alternator terminals:	$= 1,500 \text{ kW}$
Rotational speed of the turbine:	$n = 9,000 \text{ r.p.m.}$

Calculations. If there were no gearing, the rotational speed would be imposed by the frequency and the number of poles of the alternator. In this case, if we consider an alternator of 1,500 r.p.m., we shall need a speed reducer of 6 to 1.

Let us remember that $1,500 \text{ kW} = 2040 \text{ metric h.p.} = 2,010 \text{ British h.p.}$

Heat drop. The entropy diagram gives:

$$\begin{array}{r} \lambda_A = 742 \text{ kcal/kg} \\ \lambda_B = 607 \text{ kcal/kg} \\ \hline \lambda_A - \lambda_B = 135 \text{ kcal/kg} \end{array} \quad \left(\begin{array}{r} 1,335 \text{ B.T.U./lb.} \\ 1,092 \text{ B.T.U./lb.} \\ \hline 243 \text{ B.T.U./lb.} \end{array} \right)$$

Diameter. The diameter is first chosen according to the following two considerations:

(a) Keep moderate values for U : 160 m/s (500 ft./s) for a double wheel, 180-240 (590-780) for a single.

(b) In the case of high powers, we must have sufficient outlets in the last wheels, that is a moderate h/D at the blades. Here we have nothing to fear in that respect, however, since ours is a back-pressure turbine.

For the same reason, we shall be able to give to all the wheels the same diameter D , since:

$$U = \frac{\pi Dn}{60}$$

Let us take:

$$200 = \frac{3.14 \times D \times 9,000}{60}$$

Hence:

$$D = 0.425 \text{ m} \quad (1.4 \text{ ft.} \approx 17 \text{ in.})$$

Leading wheel. Let us check whether it would be advantageous to have a double wheel (eqn. 42.35):

$$P_0 = 4p \left(\frac{n}{1,000} \right) D^5 = (4 \times 23 \times 9^3 \times 0.425^5) = 930 \text{ h.p.} = 683 \text{ kW}$$

This means that we are outside the field of the double wheel, and it would not be suitable.

Number of wheels. Since we wish to have the same £ in all the wheels, we shall write (formula 42.28):

$$x + 0.04 = \frac{8,205 \times 135}{200^2} \xi^2 = 27.7 \xi^2$$

To each value of x there corresponds therefore a value of £. Let us look for the best solution among the different corresponding values.

For: $x = 3$	£ = 0.333
$x = 4$	£ = 0.384
$x = 5$	£ = 0.429
$x = 6$	£ = 0.470

If we were interested only in the efficiency, we should take $x = 6$, which would give us a £ value very close to 0.45. If, on the contrary, we required a simple and cheap turbine, we could come down to $x = 3$, which would give us a £ value close to 0.30. We shall reject these two extreme solutions, however, which would mean for us: the first, too expensive a turbine; and the second, a very low efficiency.

We are left with the two intermediate solutions. If we had, or planned to have, a thermo-compressor, and if we were seeking the utmost economy in bagasse, we should take $x = 5$ with a £ value greater than 0.40. We shall be content with $x = 4$ and with £ = 0.38, however, a value very suitable for a back-pressure turbine.

Scale of pressures. We have then £ = 0.384. Hence:

$$V = \frac{U}{\xi} = \frac{200}{0.384} = 524 \text{ m/s} \quad (1,719 \text{ ft./s})$$

On the other hand:

$$V = 91.5 \sqrt{0.94 q_1}$$

First wheel. Hence:

$$qt = \frac{524^2}{91.5^2 \times 0.94} = 34.8 \text{ kcal/kg} \quad (63 \text{ B.T.U./lb.})$$

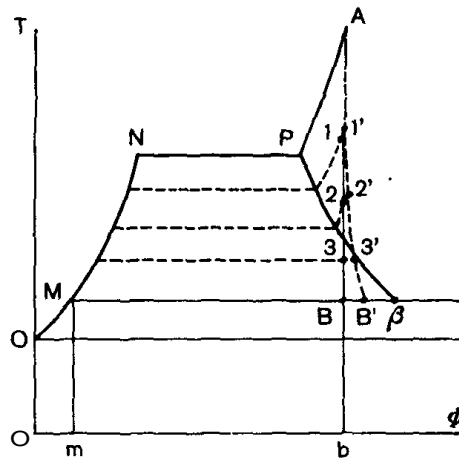


Fig. 42.13. Pressure stages.

On the entropic diagram, we look for point 1, on the vertical of A, such that:

$$\lambda_A - \lambda_1 = q_1$$

$$\lambda_1 = \lambda_A - q_1 = 742 - 34.8 = 707.2 \text{ kcal/kg} \quad (1,272 \text{ B.T.U./lb.})$$

and we read on the diagram that point 1 corresponds to: 13.6 kg/cm² abs., 263°C, $\omega_1 = 5.56 \text{ kg/m}^3$ (193 p.s.i.a., 505°F, $\omega_1 = 0.35 \text{ lb./cu.ft.}$)

Nozzles. If we have:

$$p_0 = 0.54p_1 = 0.54 \times 24 = 12.96 \text{ kg/cm}^2 \text{ abs.} \quad (183 \text{ p.s.i.a.})$$

we see that there is no constriction.

Other wheels. We have just found the heat drop in the first wheel, q_1 . We have now to release:

$$\lambda_1 - \lambda_B = 707.2 - 607 = 100.2 \text{ kcal/kg} \quad (180 \text{ B.T.U./lb.})$$

and to use three wheels. Consequently each of them will have to work under a heat drop of:

$$q_n = \frac{100.2}{3} = 33.4 \text{ kcal/kg} \quad (60.0 \text{ B.T.U./lb.})$$

and we can check that we have in fact:

$$\xi = \frac{U}{V} = \frac{200}{91.5 \sqrt{0.98 \times 33.4}} = 0.384$$

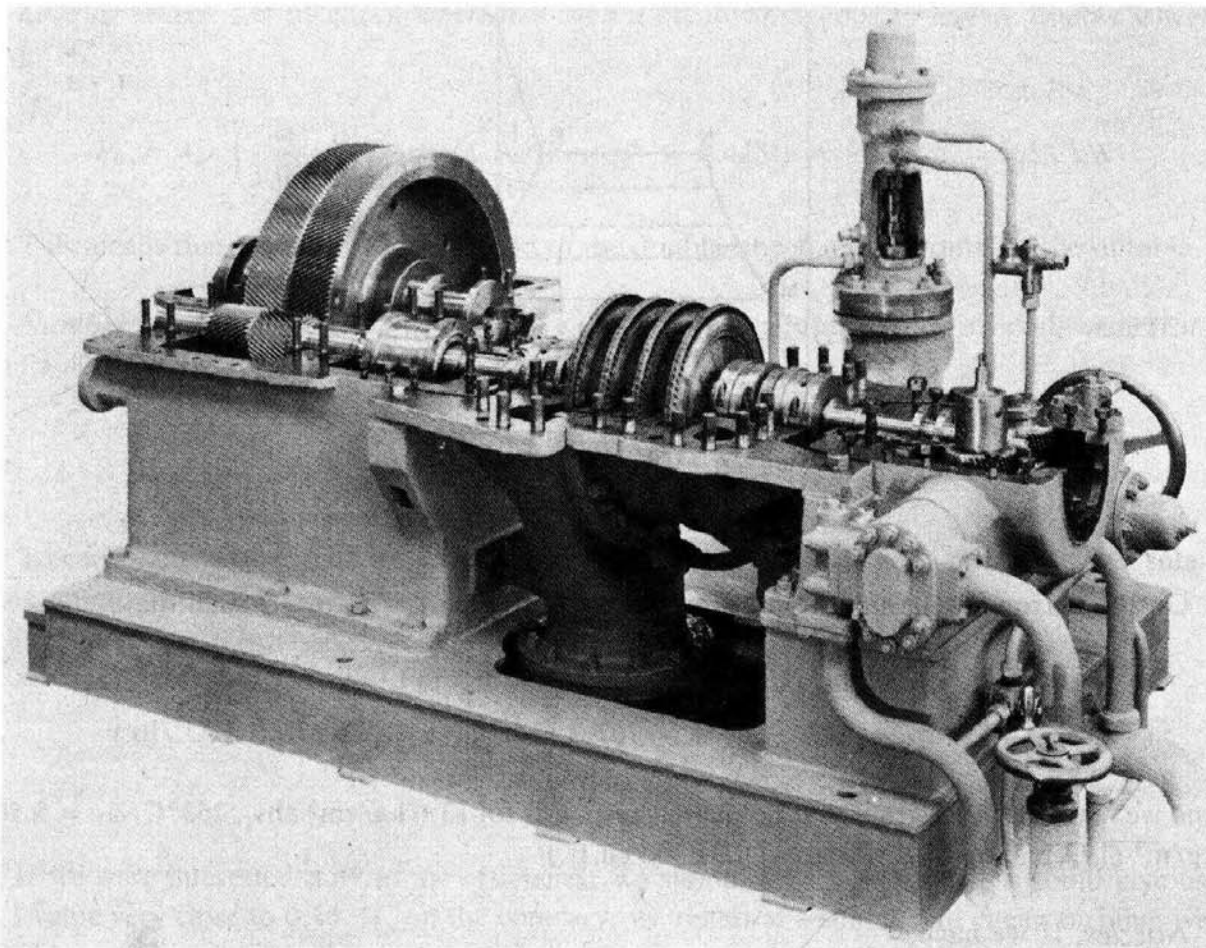


Fig. 42.14. Back-pressure steam turbine with four wheels and speed reducer, cover removed.

With the aid of the entropy diagram, we can then establish the scale of pressure stages given in Table 42.2.

The steam at the first four stages is superheated. In the last two it is wet and the dryness fraction is:

$$\text{In the 3rd wheel: } x = 0.98$$

$$\text{In the 4th wheel: } x = 0.94$$

Influence of losses. The losses in the nozzles and the losses through shock and friction in the wheels are turned into heat, thus increasing superheat or the dryness fraction of the steam. It follows that the representative points 1, 2, 3, B , actually come to $1'$, $2'$, $3'$, B' on the corresponding isobars (Fig. 42.13).

In practice we take the last point B' half-way between B and point β situated on the graph $x = 1$. Then we draw in the approximate line AB' .

Choice of turbine. The principal turbines supplied by Fives Cail - Babcock are the 40 CP and the 50 CP. We may adopt: for powers below 5000 kW the 40 CP, and for higher powers, the 50 CP.

TABLE 42.2

SCALE OF PRESSURES

	<i>Total heat λ</i>		<i>Temperature</i>		<i>Absolute pressure</i>		<i>Gauge pressure</i>		<i>Density</i>	
	<i>(kcal/kg)</i>	<i>(B.T.U./lb.)</i>	<i>(°C)</i>	<i>(°F)</i>	<i>(kg/cm²)</i>	<i>(p.s.i.a.)</i>	<i>(kg/cm²)</i>	<i>(p.s.i.g.)</i>	<i>(kg/cm²)</i>	<i>(lb./cu.ft.)</i>
Boiler	752	1,352	360	680	26	370	25	355	9.2	0.57
Admission	742	1,335	340	644	24	341	23	325	8.7	0.54
1st wheel	707	1,272	263	505	13.6	193	12.6	178	5.6	0.35
2nd wheel	674	1,211	188	369	7.1	100	6.1	85	3.4	0.21
3rd wheel	640	1,152	137	279	3.4	48	2.4	34	1.9	0.12
4th wheel	607	1,092	112	233	1.5	21	0.5	7	0.9	0.06

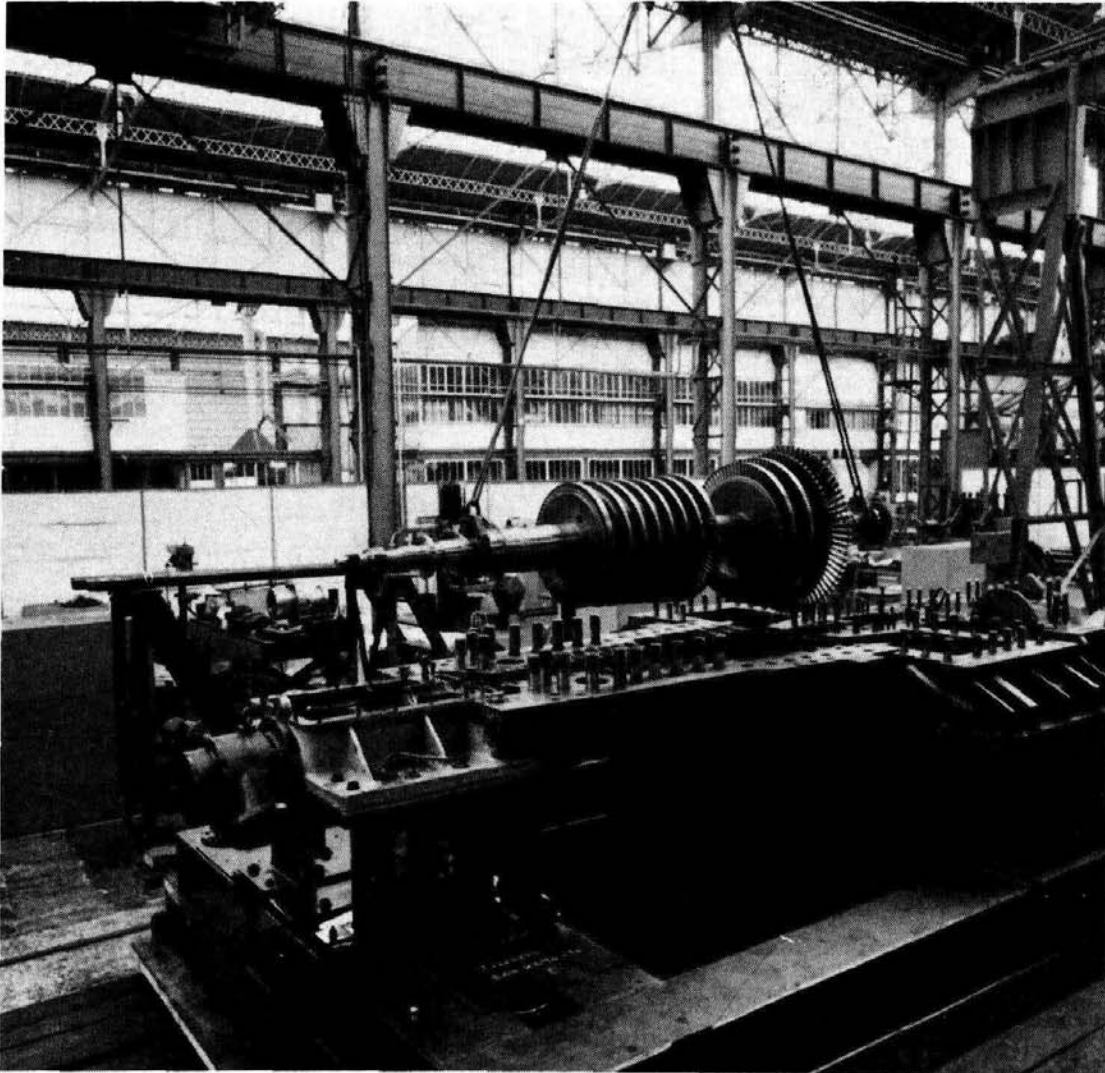


Fig. 42.15. Condensing steam turbine (40 bar, 400°C. extraction at 1.5 bar, Fives Cail – Babcock)

Minimum dryness fraction admissible

Experience has shown that in order to keep corrosion of the blades within reasonable limits, the wetness of the steam must not exceed 10% at exhaust from the last stage.

In other words, superheat will have to be chosen so that B' comes between the line of dryness 0.90 and the line $x = 1$.

Lyle² sets the extreme limit at 13% wetness; beyond that figure, the droplets of water cause serious erosion of the blades in the last stage of the turbine.

Valves

Lack of steam-tightness of the h.p. stop valve on the supply to the turbine can lead to condensation during shutdowns and consequent corrosion of the rotor.

To avoid this drawback, it is advisable to install two h.p. stop valves, placed one in front of the other with a drain open to atmosphere between the two, to evacuate condensate due to possible leakage of the upstream valve.

Accessories. For safety, a no-load speed should be limited to 10–15% above the maximum operating speed.

Speed. A mill turbine should not exceed 5,000 r.p.m. It is advisable to provide for a maximum speed of 4,500 r.p.m. For a given size of nozzles, and a fixed position of the regulator, the power drops with the speed. If it is for example, 1,000 kW at 4,500 r.p.m. it will be only 965 at 4,000, 915 at 3,500, 850 at 3,000, 765 at 2,500, and 650 at 2,000 r.p.m.

REFERENCES

- 1 ALLEN TURBINES, Sugar *Azucar*, 70 (1975) 25.
- 2 O. LYLE, *The Efficient use of Steam*, 11th edn., HMSO, 1974, p. 9.

43. *Electricity*

We have been able to give a short theory of steam turbines, summing up the principal notions relating to the working and design of those machines. It would have been desirable to do the same concerning alternators and electric motors.

Unfortunately, it is impossible to sum up in a few pages a theory, however short, of electricity and electrical machines; and those who wish to inform themselves completely on that subject must perforce be directed to special works.

We shall, therefore, be content here to recall the essential general notions and the factors which are most important from the point of view of sugar manufacture.

DIRECT CURRENT AND ALTERNATING CURRENT

Electricity is produced under two forms:

- (a) Direct current
- (b) Alternating current.

Direct current offers some advantages:

- (a) Simplicity
- (b) Powerful starting torque of the motors
- (c) Possibility of driving the motors at any given speed
- (d) Wide range of variation in the speed of each motor.

The last point is the real advantage of direct current. Steam permits of having a wide range of variation in speed in all the engines, simply by regulating the steam valve at the entry, or by adjusting the governor. With electricity we lose this big advantage; but it is alternating current which brings in the main restrictions from that point of view. Direct current still leaves a little more scope in the choice of motor speed, with the additional possibility of modifying and regulating that speed.

However, direct current is practical only for very small installations. The magnitude of the modern sugar factory demands alternating current, which offers the following advantages:

- (a) Easy transformation from high to low tension and conversely.
- (b) Relatively high efficiency in transmission and utilisation for units of large or moderate power.
- (c) Smaller size and lower cost of the motors.

THREE-PHASE ALTERNATING CURRENT

The only alternating current of any interest in sugar manufacture is three-phase supply.

Three-phase current is produced by an alternator which has three groups of windings, symmetrically disposed and displaced electrically by $2\pi/3$ in relation to each other. The windings of each of these three groups are linked in series, and it is the current they provide which feeds each of the three wires coming from the alternator. These three wires therefore carry currents displaced in relation to each other by $\frac{1}{3}$ of a period, and the same applies to their respective electromotive forces.

Let us consider these three groups of windings, or three corresponding turns in each of them. Let us call u the effective or r.m.s. (root mean square) potential difference created in each of the windings, and i the r.m.s. value of the current which flows through them. There are two ways of connecting these windings in order to transmit to the motors the current produced:

(a) *Star connection* (Fig. 43.1). The r.m.s. potential difference between any two of the three lines is then:

$$U = u \sqrt{3} \tag{43.1}$$

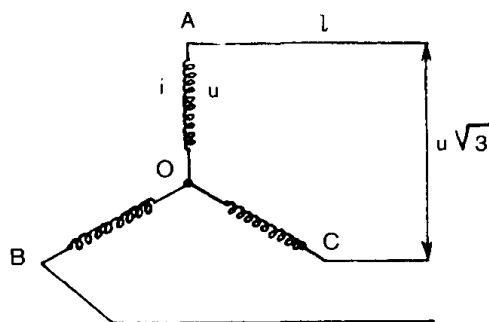


Fig. 43.1. Star connection.

and the r.m.s. current flowing through them:

$$I = i \tag{43.2}$$

(b) *Delta connection* (Fig. 43.2). It may easily be shown that in this case:

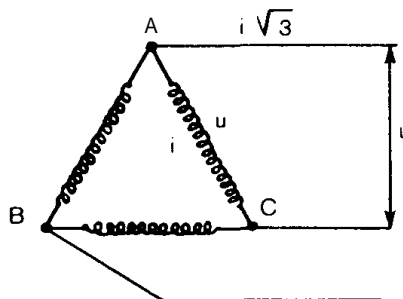


Fig. 43.2. Delta connection.

$$U = u \tag{43.3}$$

$$f = \frac{1}{T} \quad (43.4)$$

Power factor. By definition, alternating currents are currents which vary in value periodically, changing direction from a positive maximum to a negative maximum of the same magnitude (Fig. 43.3). The difference in potential follows the same law of variation, which approximates more or less a sinusoidal form.

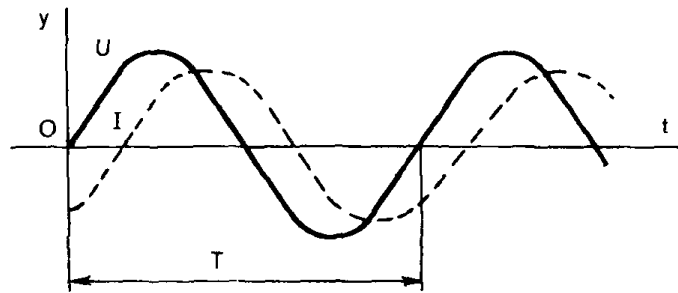


Fig. 43.3. Alternating current.

The duration T of a complete cycle is called the period, and the number of periods per second is called the frequency f of the current:

$$f = \frac{1}{T} \quad (43.5)$$

The variation of the current I or that of the potential difference U can also be represented by a vector revolving round a given point at the rate of f revolutions per second (Fig. 43.4), the instantaneous value of the current or of the potential difference under consideration being then represented by the projection Om of the vector on the x -axis.

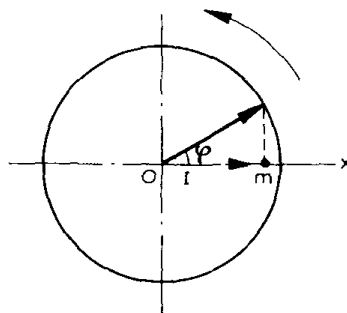


Fig. 43.4. Schematic representation of alternating current.

Owing to the self-induction of the windings, there is always a certain lag Φ (Fig. 43.4), between the current and the potential difference. If those two vectors were in phase, the power would be given by their product UI . When there is a lag Φ between them, the power is given by:

$$UI \cos \Phi \quad (43.6)$$

$\cos \Phi$ is called the "power factor". If $\Phi = 90^\circ$, $\cos \Phi = 0$, and the power produced is nil.

The power factor plays an important role in electrified factories and we shall consider it further on (p. 1093).

GENERATION OF ELECTRICITY

How can we produce on the spot the electrical energy required by the factory? Since the only actual source of energy is the bagasse, the factory has to produce the electricity in a power station driven by steam produced from bagasse in the boilers.

The back-pressure steam turbine is of great interest in the sugar factory. In producing steam at 25 kg/cm^2 (356 p.s.i.) and 360°C (680°F), for instance, we shall use about 750 kcal (1,350 B.T.U.) per kg (lb.) of steam, but that kg (lb.) of steam, when passing through a steam turbine, will give us all the power we need in the factory and to spare. And, at the exhaust of the turbine, we shall find our kg (lb.) of steam available for all the heat requirements of the whole process, and still containing more than 600 kcal (1,080 B.T.U.), that is, more than 1 of the heat we have given to it.

Thus we see how low the cost is of all the power necessary in a factory, to drive its engines and its mills, all the pumps and all other machinery; less than 25% of the heat it has to produce or of the fuel it has to burn for the purpose of heating and concentrating the juices and syrups.

That fraction is so small that it is considerably less than the difference between the consumption of steam for heating in a badly organised and badly equipped factory, and that of a well-designed one which uses its heat logically and economically. We can say that power costs nothing, or next to nothing, in a sugar factory. It is obtained almost free, while producing steam for heating purposes which is indispensable.

This shows that, even if we have at our disposal a public supply giving electricity at a very low rate; even when we have, in the vicinity of the factory, a source of water power which could be easily harnessed, we should still find it advantageous to produce the power necessary to run the factory from the bagasse.

Only some use for the bagasse giving it a value much above its value as a fuel (production of bagasse board, paper pulp, cellulose products, etc.) could lead to the use of an external source of energy for production of electrical power; or indeed, the need for power external to the factory, exceeding the margin of energy available from the bagasse, such as pumping for large irrigation demands, for example. Even there, however, since heating steam is in any case indispensable, it is highly probable that, even in these two cases, it would be of interest to produce this steam at the temperature and pressure required to obtain simultaneously the power requirements for the factory and the heating steam which is equally necessary. Whatever the source of heat utilised in the boilers, it is advisable that this source of heat should at the same time furnish the heat for power generation and that for heating.

Steam turbines of small power output, up to about 1,500 kW, are generally of geared type: they run at 8,000 – 9,000 r.p.m. and, through reduction gearing, drive an alternator running at 1,500 r.p.m. For higher powers, above 1,800 or 2,000 kW, turbines generally run at 3,000 r.p.m. with direct drive to a two-pole alternator.

Relation between frequency and speed. With an a.c. machine we have:

$$f = \frac{pn}{60} \quad (43.7)$$

f = frequency of the current, in cycles/s

p = number of pole pairs

n = rotational speed, in r.p.m.

Hence:

$$n = \frac{60f}{p} \quad (43.8)$$

and if: $f = 50$ cycles per second:

$$n = \frac{60 \times 50}{p} = \frac{3,000}{p} \quad (43.9)$$

This equation is rigorously true for alternators and synchronous motors. For non-synchronous motors it is correct if allowance is made for slip:

$$g = \frac{n - n'}{n} \quad (43.10)$$

g = slip

n = synchronous speed, given by eqn. (43.9)

n' = actual speed of the motor.

Consequently the speed of alternators and synchronous motors, or the synchronous speed of induction motors, has to be a sub-multiple of 3,000 when using a current of frequency 50, or of 3,600 for a frequency of 60 (Table 43.1).

TABLE 43.1

SYNCHRONOUS SPEEDS

	p	1	2	3	4	5	6	7	8
$f = 50$	n	3,000	1,500	1,000	750	600	500	428	375
	n'	2,900	1,450	960	720	580	480	415	360
$f = 60$	n	3,600	1,800	1,200	900	720	600	514	450
	n'	3,500	1,750	1,160	870	700	580	500	435

p = number of pole pairs of alternator or motor.

n = synchronous speed, in r.p.m.

n' = actual speed of an asynchronous motor, for a normal slip of 3 — 4%.

50 cycles per second is the standard frequency in Europe. In America, the standard frequency is 60 cycles.

The normal slip for an asynchronous motor is 3–5%; it can be appreciably higher and motors are manufactured providing for slips as high as 50%, without stalling.

Units. Let us recall in passing two units frequently used in electricity:

$$\begin{aligned} 1 \text{ kg m} &= 9.8 \text{ J} \\ 1 \text{ h.p.} &= 75 \text{ kg m/s} = 735 \text{ J/s} = 0.735 \text{ kW} \\ 1 \text{ ft.lb.} &= 1.356 \text{ J} \\ 1 \text{ h.p.} &= 550 \times 1.356 = 746 \text{ J/s} = 746 \text{ W} \end{aligned}$$

Torque of electric motors. Let us recall that:

$$W = C' \Omega \quad (43.11)$$

W = power of a motor, in watts

C' = corresponding torque, in joules

Ω = angular velocity of the motor, in radians/s.

Or, reverting to usual units:

$$C = \frac{W}{g\Omega} \quad \left(0.738 \frac{W}{a} \right) \quad (43.12)$$

C = torque, in m kg (ft.lb.)

But (*cf.* eqn. 43.7):

$$\Omega = \frac{\omega}{p} = \frac{2\pi f}{p} = \frac{2\pi n}{60} \quad (43.13)$$

ω = angular frequency = $2\pi f$ (by definition).

Hence:

$$C = \frac{60W}{2\pi ng} \quad \left(\frac{0.738 W \times 60}{2\pi n} \right)$$

or, expressing power in kilowatts:

$$C = 975 \frac{K}{n} \quad \left(7,050 \frac{K}{n} \right) \quad (43.14)$$

C = torque, in m kg (ft.lb.)

K = power, in kW = $W/1,000$

n = rotational speed, in r.p.m.

This eqn. (43.14) is used mainly for an alternator or a synchronous motor. Taking the efficiency into account, we have, for instance, for an alternator:

$$C = 975 \frac{K}{\eta} \quad \left(7,050 \frac{K}{\eta} \right) \quad (43.15)$$

C = torque required for driving the alternator, in m kg (ft.lb.)

K = power developed at the terminals of the alternator, in kW

n = rotational speed of the alternator, in r.p.m.

η = efficiency of the alternator (usually 0.90 – 0.95).

Power

The number of volt-amperes absorbed by a motor using three-phase a.c. supply is (*cf.* eqns. 43.1 – 43.4):

$$W = UI\sqrt{3} \quad (43.16)$$

W = volt-amperes absorbed

U = r.m.s. voltage between phases, in volts

I = r.m.s. current in each phase, in amperes.

If we require this figure in kVA, we shall have to write:

$$K' = \frac{UI\sqrt{3}}{1,000} \quad (43.17)$$

K' = input in kVA.

The corresponding power is

$$W = UI\sqrt{3} \cos \Phi \quad (43.18)$$

W = power used, in watts

Φ = lag between the electromotive force and the current.

$\cos \Phi$, as already stated, is called "power factor", and usually varies between 0.5 and 0.8.

Or, in order to have the power in kilowatts:

$$K = \frac{UI\sqrt{3} \cos \Phi}{1,000} \quad (43.19)$$

K = power consumption, in kilowatts.

It is of interest to know the relation between the power absorbed by a motor and the intensity of the current it uses. Since normally all the motors in a factory work on a single voltage, it is interesting to calculate once and for all the constant term:

$$\frac{U\sqrt{3}}{1,000} \quad (43.20)$$

in the formula (43.19). Knowing $\cos \Phi$ we shall have directly the coefficient:

$$m' = \frac{U \sqrt{3}}{1,000} \cdot \cos \Phi \tag{43.21}$$

and simply by multiplying by that value m' the number of amperes read on the motor ammeter, we obtain the power absorbed by the motor expressed in kilowatts.

If we prefer to have it in h.p. we shall calculate:

$$m = \frac{U \sqrt{3}}{735} \cos \Phi \quad \left(\frac{U \sqrt{3}}{746} \cos \Phi \right) \tag{43.22}$$

and we shall then have:

$$P = mI \tag{43.23}$$

P = power absorbed by the motor, in h.p.

m = coefficient given by formula (43.22) or Table 43.2

I = amperes absorbed, read on the motor ammeter.

It will be noticed that, for 550 volts, and $\cos \Phi = 0.78$, the number of amperes read on the ammeter corresponds exactly to the number of h.p. absorbed.

In the same way, at 220 volts, the value of m is roughly equal to half $\cos \Phi$.

Comment, The power P' developed on the shaft of the motor is obtained by multiplying the above power absorbed by the efficiency ρ of the motor.

$$P' = \rho P \tag{43.24}$$

TABLE 43.2

RATIO $m = P/I$ OF POWER IN H.P. TO CURRENT ABSORBED IN A MOTOR USING THREE-PHASE ALTERNATING CURRENT (British h.p.; for metric h.p. multiply by 1.0147)

Cos Φ	Voltage at the terminals (V)				
	220	380	440	500	550
0.8	0.408	0.706	0.816	0.929	1.022
0.7	0.357	0.618	0.714	0.813	0.894
0.6	0.306	0.529	0.612	0.697	0.766
0.5	0.255	0.441	0.510	0.580	0.638

In the standard type of alternator it is the armature which is fixed: it is then called the stator, and surrounds the field or inducing circuit, which is then called the rotor.

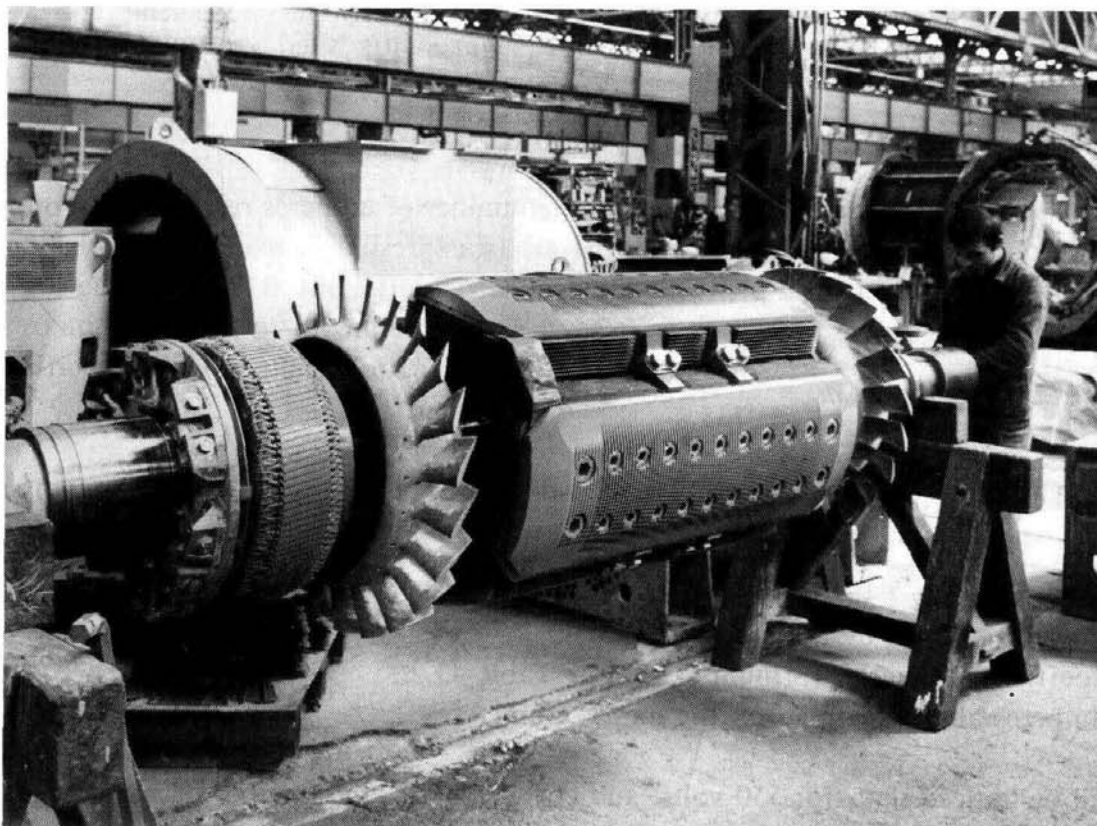


Fig. 43.5. Rotor of salient-pole alternator (4-pole) (Fives Cail – Babcock)

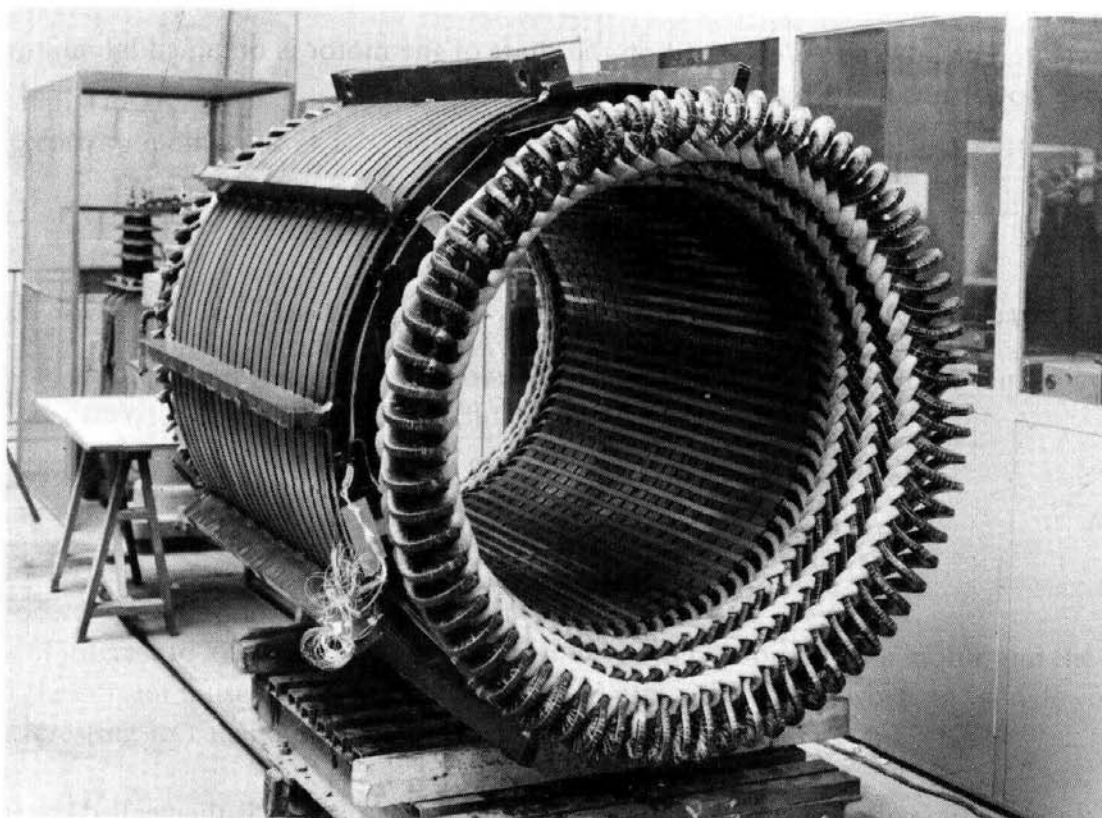


Fig. 43.6. Stator of alternator (Fives Cail – Babcock).

There are two types of rotor:

(a) Salient-pole rotors (Fig. 43.5)

(b) Smooth rotors, which are found mainly where high powers are concerned.

The current required for excitation is supplied to the field circuits by a small dynamo producing direct current, fixed to the end of the rotor shaft (Fig. 43.7) and called an "exciter". The power output of the alternator is regulated by adjusting the excitation current.

Power of individual alternators

When the capacity of the factory permits, it is of advantage to divide the total power requirement between several turbo-alternators so that the individual capacity of each set does not exceed 1800 kW, above which it becomes difficult to obtain suitable equipment, particularly safe circuit-breakers.

Efficiency

The efficiency of typical alternators in the sugar factory, which generally range from 300 to 12,000 kW, is of the order of 93.5 – 98.5%.

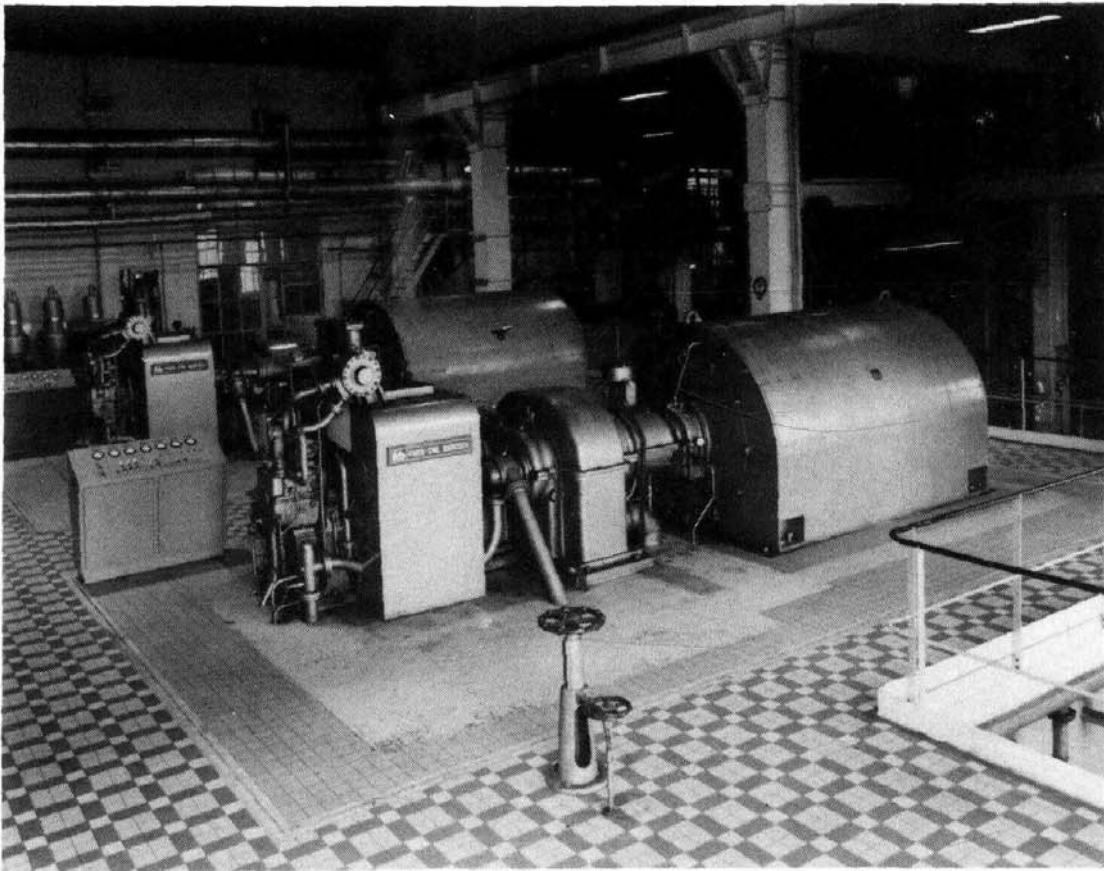


Fig. 43.7. Back-pressure turbo-generator, 9,500 kW, 7,500/1,500 pm. Adm., 40 bar eff., 400°C.
Exhaust, 2.5 bar eff.

ELECTRIC MOTORS

Different types of electric motor are distinguished by:

- (a) Their system of protection and the cooling method which it involves
- (b) Their windings and the resulting electrical characteristics.

Classification of motors according to system of protection

This classification combines protection of personnel against contact with moving parts or live conductors, and protection of the machine against external material: solid bodies, dust, water or gases. The open type, which involved risks to personnel, is no longer constructed, and today we distinguish the following principal types:

(a) **Drip-proof type** (Fig. 43.8): drops of water and objects falling vertically cannot penetrate into the motor. Certain parts which are alive or moving are exposed.

(b) **Protected type** (Fig. 43.10): the motor is protected against projection of water or solid objects which could hit the machine at a direction making an angle with the vertical up to 60° .

These two types are often provided with grilles (Fig. 43.8), *i.e.* their ventilation openings are covered with open grille-work to prevent accidental contact or penetration of solid bodies above a certain dimension.

(c) **The semi-enclosed type** (Figs. 43.9, 11 and 12): while the machine is not completely sealed, exchange of air between the interior of the motor and the exterior is prevented except

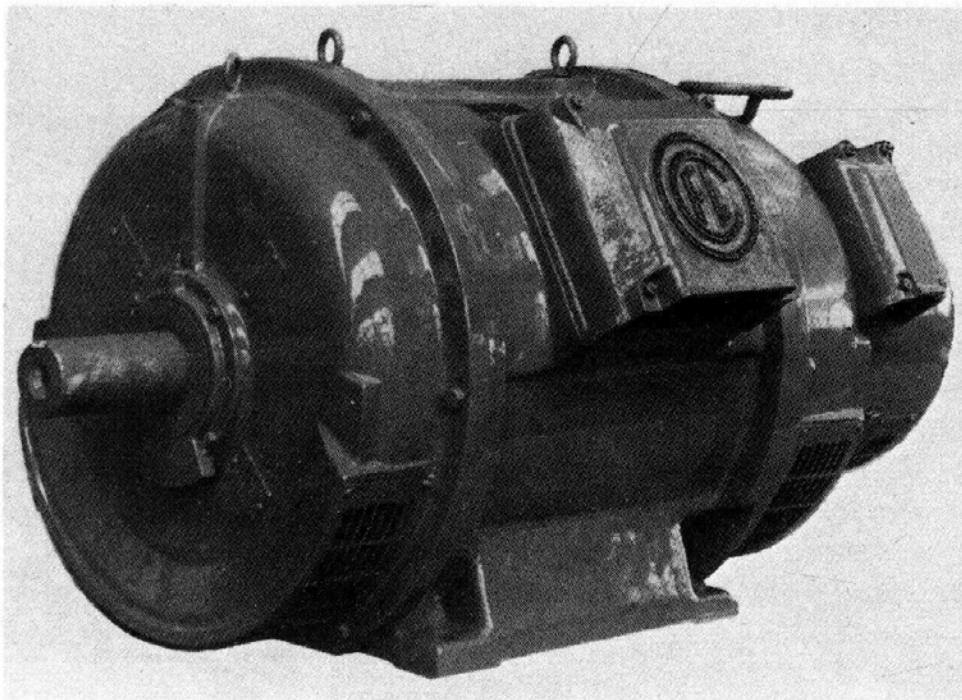


Fig. 43.8. Drip-proof motor (Fives Cail – Babcock).

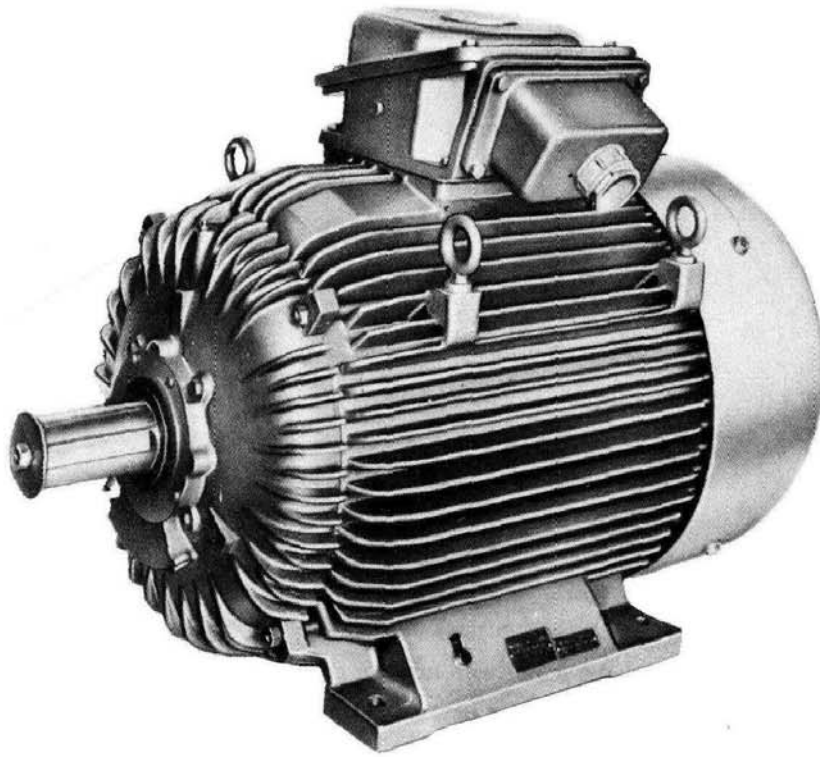


Fig. 43.9. Enclosed induction H.T. motor (Jeumont - Schneider).

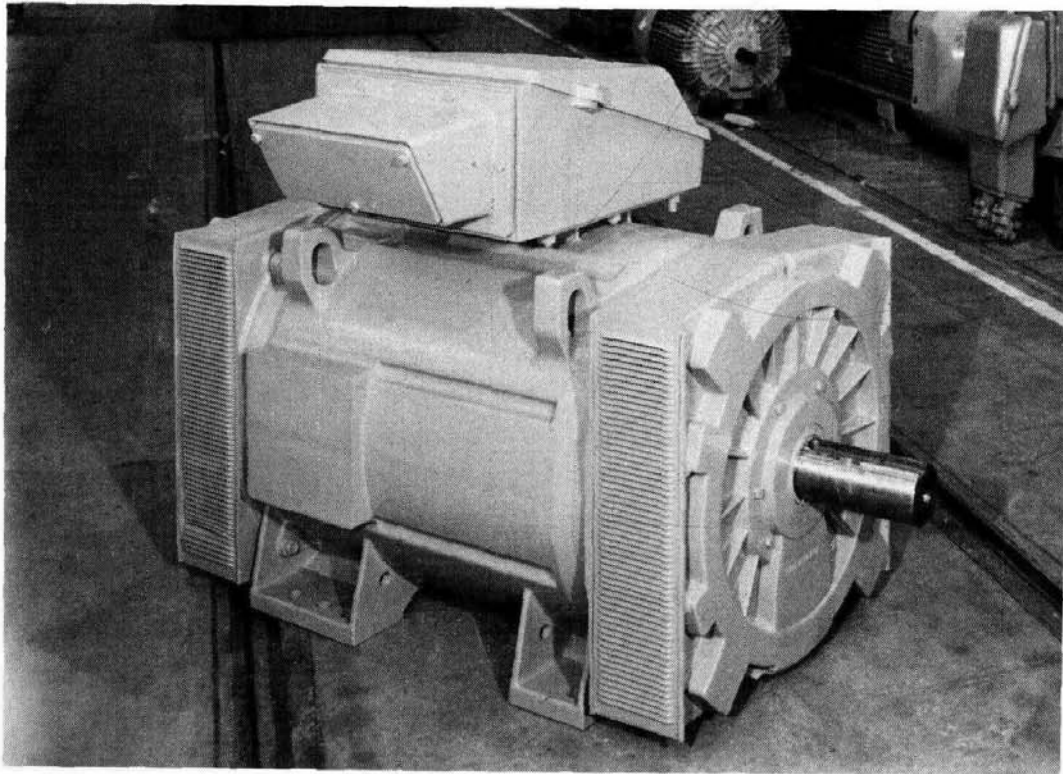


Fig. 43.10. Protected induction motor (Jeumont - Schneider).

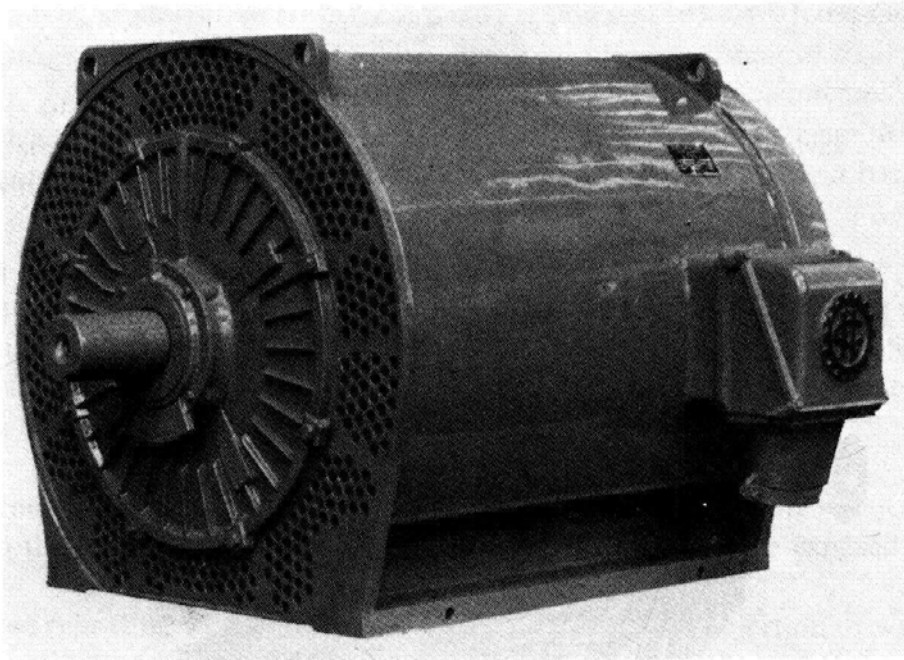


Fig. 43.11. Enclosed motor with tube for cooling (Fives Cail-Babcock).

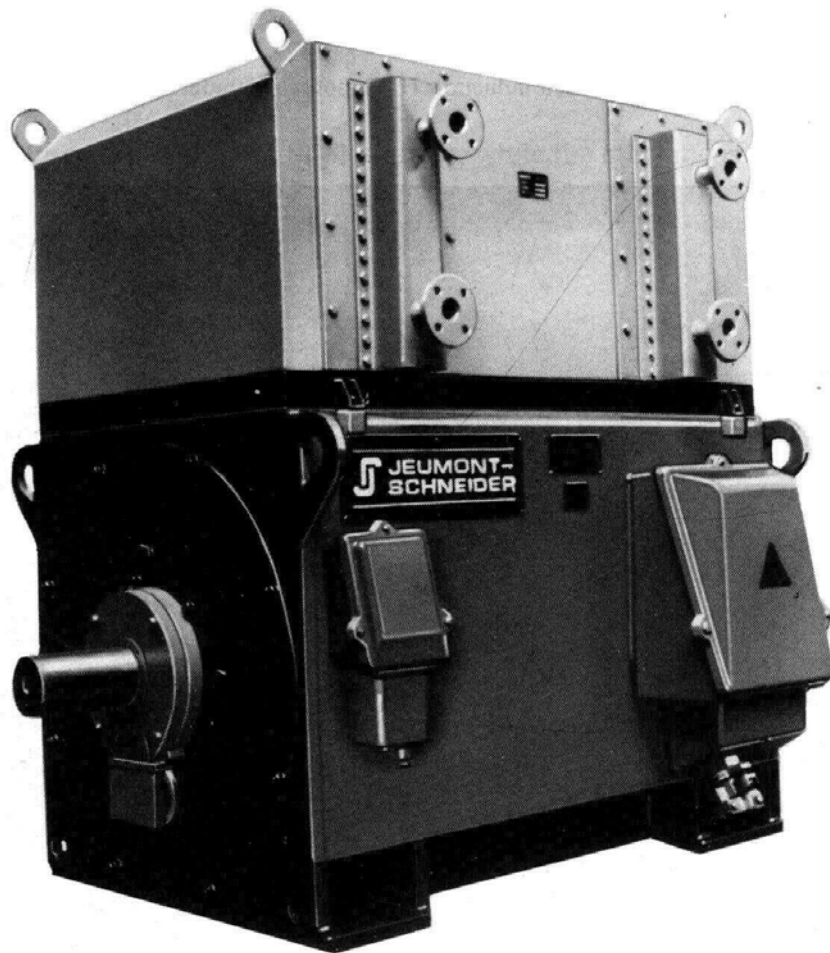


Fig. 43.12. Enclosed motor with air or water cooling (Jeumont-Schneider).

at restricted openings. Large particles and drops of water cannot penetrate; no contact is possible with live parts or moving parts.

(d) Totally-enclosed type: the motor is constructed so as to prevent penetration of gas or steam under specified conditions of pressure and temperature, or penetration of water when submerged to a specified depth.

(e) Explosion-proof type: the machine is constructed so as to operate in an ambient atmosphere which may become explosive. An explosion occurring inside the casing cannot be propagated to the outside air (Fig. 43.13).

Monnier estimates that with similar magnetic and electric circuits, fully enclosed motors show, compared with the earlier type of open motors, a reduction in power output given by Table 43.3.

TABLE 43.3

REDUCTION IN POWER COMPARED WITH AN OPEN MOTOR

Enclosed ventilated motor	12%
Enclosed motor with double ventilation	20%
Totally enclosed motor without ventilation	50%

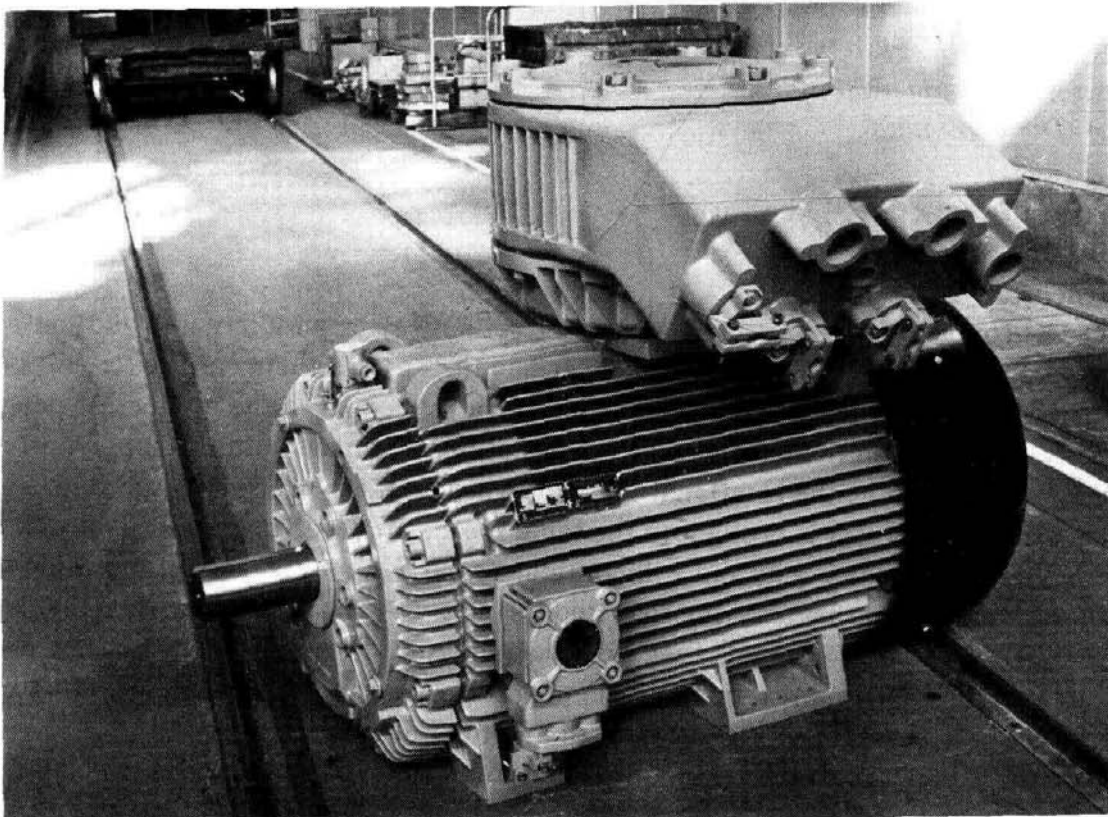


Fig. 43.13 Explosion-proof motor (Jeumont - Scheider).

The enclosed motor with double ventilation is fitted with two fans, one moving the air inside the motor, the other on the outside along the fins of the casing.

Normally, according to requirements, the types used in sugar factories will be the semi-protected, protected or drip-proof types wherever the atmosphere is clean and the risk of access of water is slight. In countries with high rainfall, and especially in countries subject to cyclones, it is advisable to choose preferably the protected and drip-proof types, owing to the danger of wetting due to drops falling from the roof or from the skylights, or driving rain coming through doors and windows. If the air is loaded with dust, ashes or fine bagasse, the totally enclosed motor is necessary, in spite of the cost. This is the case with the motors driving the lateral tables, the auxiliary carrier, the cane crane, the bagasse conveyors, and all the motor pumps placed in pits or in a humid place and exposed to splashes or drips of liquids.

Classification of motors according to their electrical characteristics

According to whether the speed of the rotor is rigorously tied to the frequency, we distinguish:

(1) *Synchronous motors*, the speed of which is given exactly by line n of Table 43.1.

(2) *Asynchronous motors*, which give a torque only when their speed is lower than the synchronous speed. These are subdivided into two main groups:

(a) *Induction motors*, including:

Induction motors with wound rotor and slip rings

Squirrel-cage induction motors.

(b) *The commutator motor*.

We shall study briefly the various types.

1. Synchronous motors

The synchronous motor is built in exactly the same way as an alternator and may be considered as an alternator used as a motor. It has then a direct current exciter (Fig. 43.14).

The inducing rotor has a number of poles obviously equal to the number of poles of the induced stator. When the terminals of the armature are linked with the supply, a revolving field is produced in it, the rotational speed of which is the synchronous speed given by formula (43.8). If the rotor is brought to the same speed, the revolving field will attract the corresponding pole following it, and will continue to carry it along. The torque, which was nil when the lag between the revolving field and the pole was zero, increases when that lag varies from 0 to half a polar interval. It is then maximal and decreases when the lag varies from half to one polar interval. In the last case, operation is unstable, since an increase of the torque leads to an increase in the lag, and the motor falls out of step.

If we draw the graph representing the relation between the current I in the stator and the excitation current i (Fig. 43.15), we obtain a V-curve. The current I is then at a minimal for a certain value of i ; below this value the current lags behind the electromotive force (e.m.f.);

at higher values it is ahead of it. Hence an interesting property of the synchronous motor: if over-excited, it returns reactive energy to the network instead of absorbing it.

The synchronous motor, then, has the very great advantage of improving the power factor of the factory. From the mechanical point of view it has the advantage of permitting a considerably larger air-gap than the non-synchronous motors.

However, it is very seldom used, because:

(1) It is difficult to handle.

(2) It needs an exciter.

(3) It is difficult to start, and does so only if the resisting torque is very small, *i.e.* at most 15 — 30% of the normal torque, according to the starting method used. Otherwise it has to be started with the help of a small auxiliary non-synchronous motor.

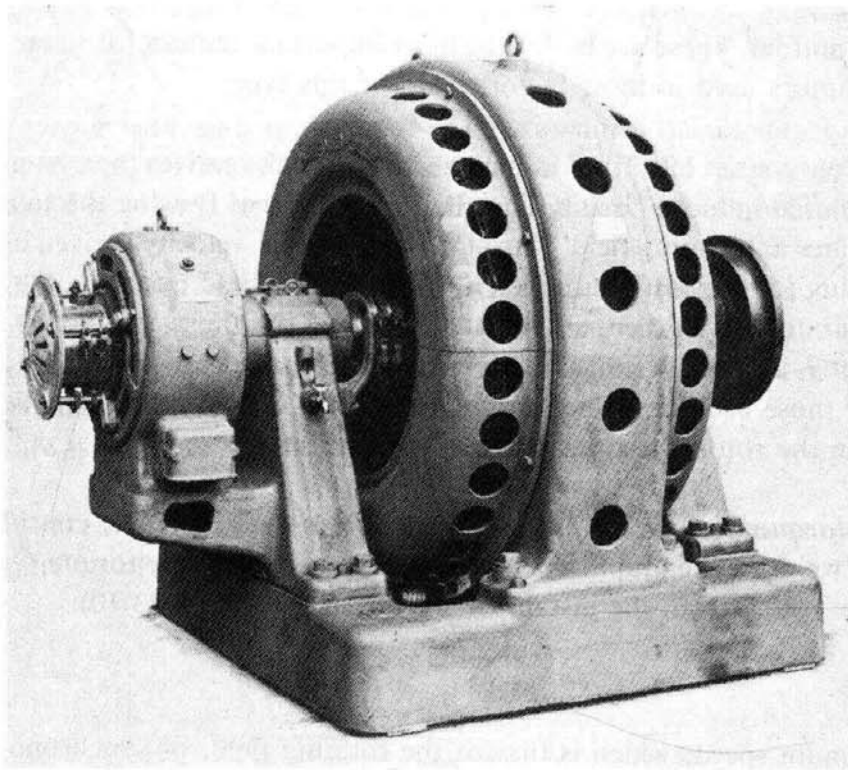


Fig. 43.14. Synchronous motor.

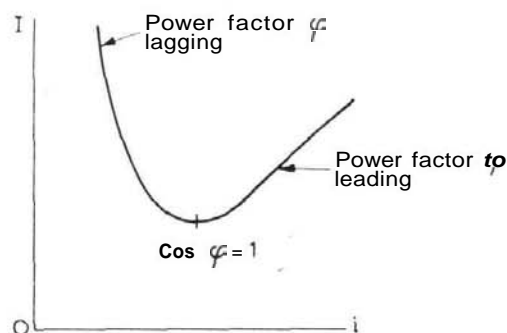


Fig. 43.15. Synchronous motor. Variation of the stator current I as a function of the excitation current i .

(4) Even from the point of view of the improvement of $\cos \Phi$, it can nowadays be replaced by synchronised or compensated non-synchronous motors, which are more convenient.

(5) It allows no variation of speed (this characteristic can, however, offer an advantage in the cases where a constant speed is needed).

(6) It risks falling out of step when the overload is heavy. However, it generally carries an overload corresponding roughly to a total load double the normal load.

A synchronous motor could be used for an air pump or a fan; however, it is not widely used in factories. It could also be used simply as a condenser, by letting it run without load, in order to improve the power factor. But there are other ways of obtaining this result (*cf.* p. 1094).

2. Asynchronous motors

A. Induction motors. These are by far the most important motors for sugar factories, and practically all motors used in most factories are of this type.

Operation. They consist of a fixed inducing stator, which receives the current, and a rotor which forms a closed induced circuit. The alternating current flowing through the windings of the stator creates a "rotating field", turning at an angular velocity Ω given by eqn. (43.13), and corresponding to the rotational speed n . This rotating field intersects the rotor conductors and in its turn causes in them an induced e.m.f. E_2 . As those conductors form a closed circuit, a current I_2 flows in them, the intensity of which depends on the reactance and the resistance R_2 of those windings. The torque produced by this electro-magnetic action of the rotating field on the rotor is proportional to the current I_2 .

Variation of torque with speed. Let us suppose R_2 is the resistance, considered as fixed, of the rotor. If we draw the graph representing the variation of the torque C as function of the speed of the rotor, from the instant of starting (*cf.* formula 43.10):

$$n' = 0 \quad g = 1$$

up to the maximum speed, which is that of the rotating field, or synchronous speed:

$$n' = n \quad g = 0$$

we obtain a graph of the form represented by Fig. 43.16. The torque starts from a certain value C_0 when the rotor is stationary, at starting, reaches a maximum C_M , and tends towards zero as the speed approaches the synchronous speed. At that speed, the rotating field, having the same speed as the rotor, is stationary in relation to it, and is no longer cut by the conductors. Therefore there is no variation of the flux cut, nor any induced e.m.f.

Stability. When the load increases, the motor slows down. It can be seen at once that the motor will be able to function only on part MNA of the curve, at which any reduction in speed gives an increase in the torque. On part MC_0 , on the contrary, a slowing down causes a decrease in torque, and the motor falls out of step immediately.

As it is imperative to keep a safety margin in order to avoid stalling, the motors are constructed so as to cope with the normally accepted overloads. This generally leads to the adoption of a normal torque C_N equal to about half the maximal torque C_M or slightly more: ($C_N = 50 - 60\%$ of C_M). The corresponding slip g_N will be about one-third of the slip at stalling g_M .

Slip. It is seen that the slip increases almost proportionally to the torque from zero to the maximal torque C_M .

The slip at no load is of the order of 0.001; for large motors, 0.0008. The motor thus runs practically at synchronous speed.

At full load the slip is of the order of 2% for large motors of 200 h.p. and above, or in general 1.5 – 3%.

The drop in speed between 0 and 10% slip is, at constant load, proportional to the rotor resistance².

Variation of torque with rotor resistance. We have assumed the resistance of the rotor fixed. If, by some device, we make it vary, and then draw on the same graph all the curves similar to that of Fig. 43.16, we shall find a family of curves as shown on Fig. 43.17.

We see that:

- (1) The smaller the resistance R_2 , the lower is the starting torque C_0 .
- (2) The maximum torque C_M is independent of the resistance R_2 of the rotor.
- (3) The maximum torque C_M corresponds to a speed n' increasing (or to a slip g decreasing) when resistance R_2 decreases.
- (4) We can always choose R_2 so that the maximum torque C_M is reached for a given speed n'_x .

In particular, R_2 can be given the value $R_2(4)$ for which the maximum torque is reached at the very beginning of the starting ($n' = 0$).

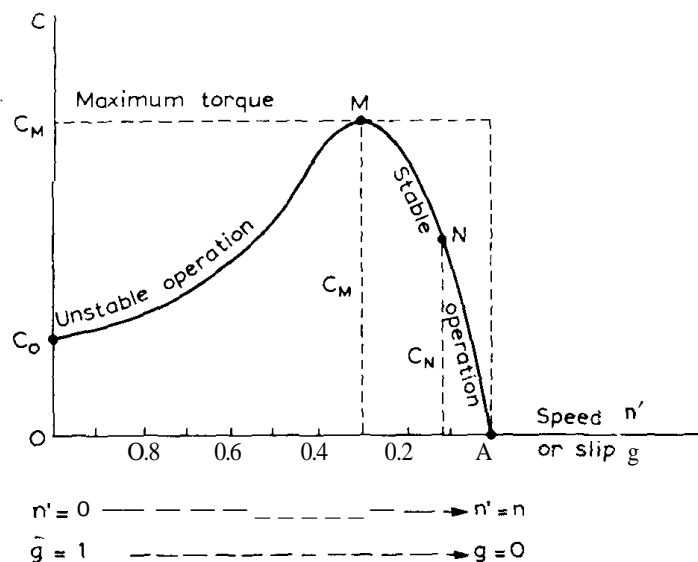


Fig. 43.16. Induction motors. Variation of torque as a function of speed.

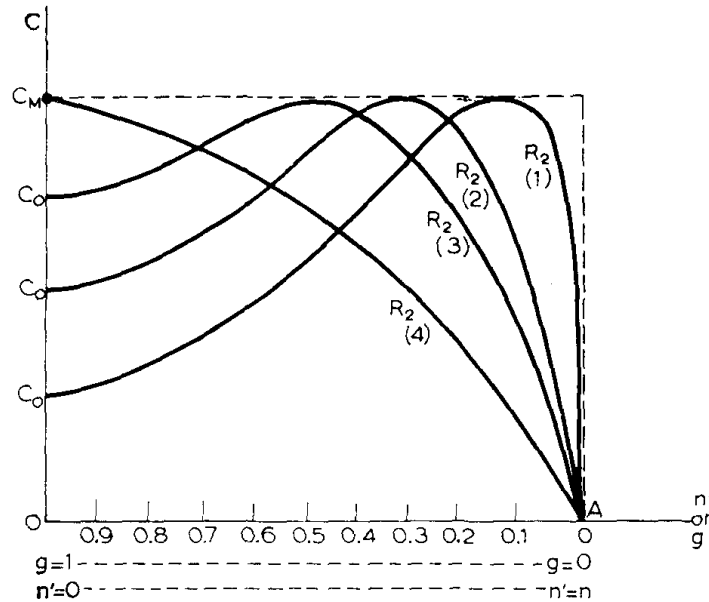


Fig. 43.17. Induction motors. Variation of torque as a function of speed and rotor resistance R_2 .

Types of induction motor. This study enables us to see the difference between the two types of induction motor. According to whether or not it is possible to vary the resistance R_2 of the rotor, we distinguish between:

- (a) The *squirrel-cage* induction motor, in which the rotor resistance R_2 is fixed.
- (b) The *induction motor with wound rotor and brushes*, in which the resistance R_2 of the rotor may be varied.

(a) *Squirrel-cage motor.* The rotor consists simply of a series of copper rods, linked together at both ends by copper rings (Fig. 43.18).

As we can see, this motor is very simple, very strong and very cheap. It does not require any rheostat, and starts without any extra operation, simply by switching on. Its efficiency and its $\cos \Phi$ values are slightly superior to those of the motor with wound rotor.

But it also has drawbacks:

(7) It makes it necessary to choose, once and for all, the resistance R_2 of the cage and, consequently, the curve of the torque variation. If one needs a strong starting torque, an R_2 (3) curve (Fig. 43.17) corresponding to a high resistance will be considered. In general, this is the case with high-grade centrifugals which have to be started rapidly and frequently. If starting can be effected at no load or with a reduced load, a curve of type R_2 (1) or R_2 (2) will be chosen.

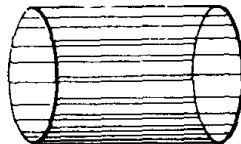


Fig. 43.18. Squirrel-cage rotor.

(2) Curve R_2 (4) cannot be chosen, for conditions at normal running would be very bad. Consequently the resistance R_2 chosen is always too low for starting, and a very heavy current I_2 has to be used at that moment. I_1 in the stator varies more or less as I_2 , for I_2 is the main component of I . Consequently squirrel-cage motors require in the stator a very strong starting current, which becomes prohibitive where high powers are concerned.

Those drawbacks have been remedied in two ways as follows:

(1) By star-delta starting: the stator is connected in delta for the normal running and in star for starting.

(2) By using the double-cage Boucherot motor: the rotor is formed by two superposed cages, the outside one made of small rods, having a high resistance and low reactance; the inside one, made of thick rods, having a low resistance and high reactance. The outside cage gives a torque of the C_1 type (Fig. 43.19), maximal and preponderant at the moment of starting. The inside cage gives a torque of the C_2 type, which is preponderant when it nears the synchronous speed. It is as if the torque followed the variation $C = C_1 + C_2$.

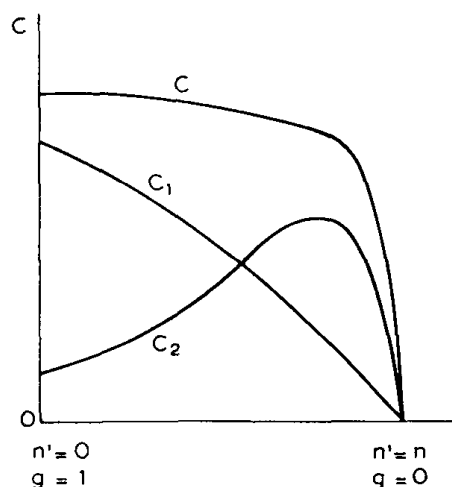


Fig. 43.19. Torque of Boucherot motor.

TABLE 43.4

SQUIRREL-CAGE MOTORS. STARTING TORQUE AND CURRENT AS % OF VALUES AT FULL LOAD

Motor	Method of starting	Torque	Current
Single cage	{ Direct on line	100 – 150	500-800
	{ Star-delta	40	150
Double-cage Boucherot	{ Direct on line	160 – 250	200 – 400
	{ Star-delta	60 – 70	110-150

However, those two remedies are only palliatives, as Table 43.4 will show us: it gives the torque and starting current, in % of their value under full load.

Usage. The squirrel-cage motor will be used for preference whenever possible, owing to its low cost and its simplicity, and because it requires little upkeep and gives little trouble.

But its characteristics limit its use to the following cases:

- (a) When the speed required has to be constant
- (b) When the starting torque is less than the full load torque.

Because of its influence on the factory network, at the moment of starting, it will intentionally be used for low rather than for high powers. In general, the squirrel-cage motor is not used above 30 or 40 h.p.

It is used mainly for centrifugals and for small pumps, and sometimes for cane carrier drives.

(b) *Motor with wound rotor and slip rings.* This is by far the most common type used in sugar factories. The rotor consists of windings connected to three rings placed on the shaft of the motor, and corresponding to the three phases. By means of three brushes, these rings can be connected with three variable resistances, constituting a triple rheostat.

When the resistance increases, the speed decreases proportionately. The voltage drop occurring in the resistance should be compensated by the rotor, which involves an increase in slip, or in other words a drop in speed.

The maximum value of the resistances is chosen in such a way that it corresponds to curve R_2 (4) in Fig. 43.17. By applying the maximum resistance it is possible to start with the maximum torque, with a relatively low current. Then the resistances are decreased gradually by changing from one contact stud to another on the rheostat, which corresponds to passing from one curve to another in the series R_2 (4), R_2 (3), R_2 (2) ... When the motor is nearing the synchronous speed, the resistances are suppressed by lifting the brushes, which leaves the rotor short circuited on its own resistance. Previously, the motor was stopped by lifting the brushes. At present the brushes are left permanently in contact, and a time switch is used to cut out the resistances successively from the circuit, over a period of 5 — 15 seconds. The motor with wound rotor and slip rings therefore is not so simple; it requires a starting rheostat and a time switch. Its efficiency and its $\cos \Phi$ are slightly lower than those of the squirrel-cage motor, but it permits of starting with a high torque, without absorbing excessive current. Coil resistances were previously used; at present resistances of electrolyte type are used, the electrolyte being, for example, soda solution.

Torque and current at starting. It is possible to obtain a starting torque more than double the full-load torque, if desired. Whatever the starting torque used, if it is m times the full-load torque the corresponding starting current will be $1.25m$ times the full-load current, and acceleration will be very smooth if the resistance in the rotor circuit is progressively cut down.

Usage. The motor with wound rotor and slip rings is used:

(a) When the speed requires slight adjustments below the synchronous speed. The slip is then made to vary by using the rheostat. The adjusting margin is in the range of 10% for ordinary motors, but it can reach 30% and even 50% in some specially designed motors. However, that margin is obtained at the expense of efficiency, since the energy is lost in heat in the rheostat.

(b) When the starting torque is equal or superior to the full load torque.

The wound rotor motor is used for the mills, the vacuum pumps, other larger pumps and, in short, for almost all the large motors in the factory.

Efficiency of induction motors. The efficiency of induction motors improves slightly with:

Speed
Power
Load

(a) Speed. The influence of speed is rather small. Still, there is an advantage in choosing motors with the highest speed possible; but the difference in efficiency between, say, 750 r.p.m. and 1,500 or 3,000 r.p.m. is little more than 1%.

(b) Power. On the contrary, the effect of the power is very important. The optimum efficiency is in the range given by Table 43.5.

TABLE 43.5

EFFICIENCY OF INDUCTION MOTORS		
Power (h.p.)		Efficiency (approx.) %
3-	5	77-83
5-	10	79-85
10-	20	82-87
20-	50	84-90
50-	100	88-91
100-	1,000	90-92

Powerfactor. One of the biggest drawbacks of non-synchronous motors is their low power factor. Like the efficiency, the power factor of induction motors varies with speed, power and load.

(a) Speed. The influence of speed on $\cos \Phi$ is much greater than on the efficiency (*cf.* Table 43.6).

TABLE 43.6

POWER FACTORS OF INDUCTION MOTORS (optimum $\cos \Phi$ at full load)					
Power (h.p.)		Speed at full load (r.p.m.)			
		3,000	1,500	1,000	750
3-	5	0.88	0.82	0.75-0.78	0.73-0.76
5-	10	0.89	0.83-0.84	0.78-0.81	0.76-0.80
10-	20	0.90	0.84-0.88	0.81-0.84	0.80-0.83
20-	50	0.90-0.91	0.88-0.89	0.84-0.87	0.83-0.86
50-	100	0.91	0.89-0.90	0.87-0.88	0.86-0.87
100-	1,000	0.92	0.90-0.91	0.88-0.90	0.87-0.88

(b) Power has an influence on $\cos \Phi$ similar to its effect of efficiency.

(c) Load. The power factor always decreases at the same time as the load, slowly at first, then rapidly. It falls to about 0.2 when the motor runs at no load. Hence it is reasonable to oppose the tendency of the manufacturers to build motors which are too powerful and offer too wide a safety margin. If they had it their own way, a factory would work with an

average $\cos \Phi$ of 0.5. It must be remembered that the shorter the duration of the overloads, the greater they can be, and the power of each motor installed has to be carefully determined.

Average power factor. In spite of the precautions which can be taken, a factory often functions with an average power factor, shown by the switchboard at the electric power house, much lower than the theoretical value 0.8, which is so often quoted but rarely recorded in practice.

This is due to the motors working under variable load, which work far below their nominal power for an important part of the time, and in particular:

- (a) The centrifugals
- (b) Pumps regulated by a valve on the discharge.

The average power factor in a factory in which no special system is used to improve it is generally in the range 0.6—0.7. Sometimes it comes down to 0.5, but rarely goes above 0.75, unless special measures are taken.

Drawbacks of a low power factor. A low power factor presents drawbacks:

- (a) Unsatisfactory utilisation of the alternator. The output of the alternator which produces electricity for the factory, in the power house, is limited by the current I which flows through it. But the useful power provided is only proportional to $I \cos \Phi$. Consequently if $\cos \Phi = 0.4$, the power the alternator will be able to produce will be only half that which it could produce with a $\cos \Phi = 0.8$. If that has been foreseen, it has been deemed necessary to buy an alternator bigger than it should be. If not, the expected power will not be available.
- (b) Increase in the losses through the Joule effect in the lines; these losses are proportional to I^2 .
- (c) Increased voltage drop in the alternator and the lines.
- (d) Decrease in the efficiency of the alternator; this can be more than 1%.

Improvement of the power factor. In order to improve power factor we have at our disposal the following means:

- (a) Better adaptation of the power of the motors to the work they have to perform.
- (b) Introduction of condensers, either at the switchboard, or preferably at the motors responsible for the low power factor, and in particular the centrifugal motors, so as to relieve the corresponding line connecting the switchboard to the centrifugals, which otherwise would remain overloaded. In this way the power factor of that sector can be brought up to about 0.95. The drawback of these condensers, however, is that they are expensive and that they create risks of boosting the voltage when the motors are being switched on or off.
- (c) Use of an over-excited synchronous motor (*cf.* Fig. 43.15); but the weak torque at starting makes the use of such motors difficult.
- (d) Use of synchronised or compensated asynchronous motors: these are now hardly used in industry.

B. Commutator motor. This is the second type of non-synchronous motor. Although it is much less widely used than the induction motor, it presents a certain advantage and has found some applications.

In these motors the rotor, like the armature of a dynamo, is connected with a commutator

by brushes, the number of which is equal to the product of the number of phases by the number of pole pairs. Rotor and stator are supplied with current, either in series or shunt connection, from the a.c. supply. By modifying the position of the brushes, the lag between the e.m.f. of the stator and that of the rotor may be altered at will, hence a great number of possible combinations.

Advantages. From the point of view of the range of speed variation, the commutator motor offers possibilities much superior to those offered by induction motors. Hence:

TABLE 43.7

RANGE OF SPEED VARIATION OF COMMUTATOR MOTORS

Series motor:	from 0.4 to 1.3 times synchronous speed, say 1 : 3 approx.
Shunt motor:	from 0.2 to 1.5 times synchronous speed, say 1 : 8 approx.

- (a) Wide range of speed variation⁴, see Table 43.7.
- (b) Easy and very smooth starting.
- (c) Small variation of speed with load, for the shunt motor.
- (d) Possibility of choosing a given speed for any given load.
- (e) High efficiency, maintained over a large part of the scale of speeds.
- (f) Very high power factor for speeds around the synchronous speed, sometimes reaching 1 and even leading at speeds higher than synchronous.

It is the commutation which limits the use of the commutator motor: good at low speeds, it becomes bad above the synchronous speed.

Usage. The commutator motor is the obvious choice every time a wide range of speed variation is required, and especially when that variation is expected without the high losses involved in the rheostat of an induction motor with a wide range of slip.

It is sometimes used for pumps, and especially for fans with adjustable speed.

Combinations of a.c. and d.c. motors. In the sugar factory, these combinations are employed only in mill drives. They are discussed in Chapter 15.

Protection of electric motors. A prudent practice, which we recommend, consists in never installing any motor of more than 7.5 h.p. without providing a device comprising starter together with overload and low voltage protection.

Overload relays. However, we have to watch continually, and fight against, certain tendencies of the employees in the factory: very often the person in charge of an electric motor, because he is annoyed at the circuit breaker opening frequently, will modify the adjustment so that it does not open; but instead of this, the reason for its frequent opening should be sought and corrected, while taking careful precautions that the protection which it should give is not lost.

It is generally agreed that maximum current relays should function for a current three times that corresponding to the normal continuous load of the motor.



Fig. 43.20. Group of two condensing turbo-generators (40 bar - 400°C. Extraction at 1.5 bar).

Thermal overload relays. Thermal overload relays, on the other hand, which operate by heating, have to be adjusted for a current only 10-20% above that corresponding to the maximum power which the motor can provide for one hour of sustained running.

Maintenance of electric motors

As soon as the campaign is finished, the small electric motors have to be removed, marked for identification, cleaned and stored in a special room, sheltered from moisture and from cyclones. For large motors, a hood of light sheet iron should be built which will cover the motor completely. After cleaning the motor, it will be covered with a sheet of impervious material such as plastic or polythene fitted closely over it, and, thus wrapped, will be placed under the metal hood.

Before the crushing season begins, the motors will be carefully dried, for example by switching on under the cover a large electric lamp which gives out a certain amount of heat, and leaving it there for a week. The motors will then be run at no load for half a day or a day, or longer if necessary.

If a motor has been splashed with juice, the windings will be washed with a brush and a cloth, until all viscous deposits have been removed. They will then be wiped and dried carefully.

Cleaning during the slack season should include blowing out the dust from the spaces between the windings. This operation is readily performed with the aid of an air compressor or a simple tyre pump.

If the motors are wet following a cyclone, they have to be dried out as soon as possible

in a drying room, which can be a makeshift cabinet easily constructed. The temperature should not exceed 90°C (194°F) so as to avoid damage to the insulation.

CONDUCTORS

Different types of conductor

These may be classified under two main headings.

A. Control cables. These are small wires serving for control, regulation and metering, particularly for automatic equipment. They form the nerve system of the factory. These conductors are generally of copper; their cross-section generally ranges between 0.8 and 10 mm². They are used with insulation designated as:

(a) V 1000 RO 2V, for L.T. (up to 1000 V), International standard CEI 502, consisting of:
A core of copper or aluminium, 3 conductors

Number of wires 6 to 37

Insulating coating in PRC (chemical polyethylene reticule) around each conductor

Elastoplastic coating of the bundle of 3 conductors, or synthetic strip

Exterior coating of black p.v.c. (polyvinyl chloride)

Usual cross-section of each conductor = 1.5, 2.5, 4, 6, 10, 16 or 25 mm²

This cable, if laid underground, should have a protective metal sheet.

(b) U 1000 RGPV, for L.T. (to 1000 V); similar protection as above with additionally:
Water-tight sheath of lead

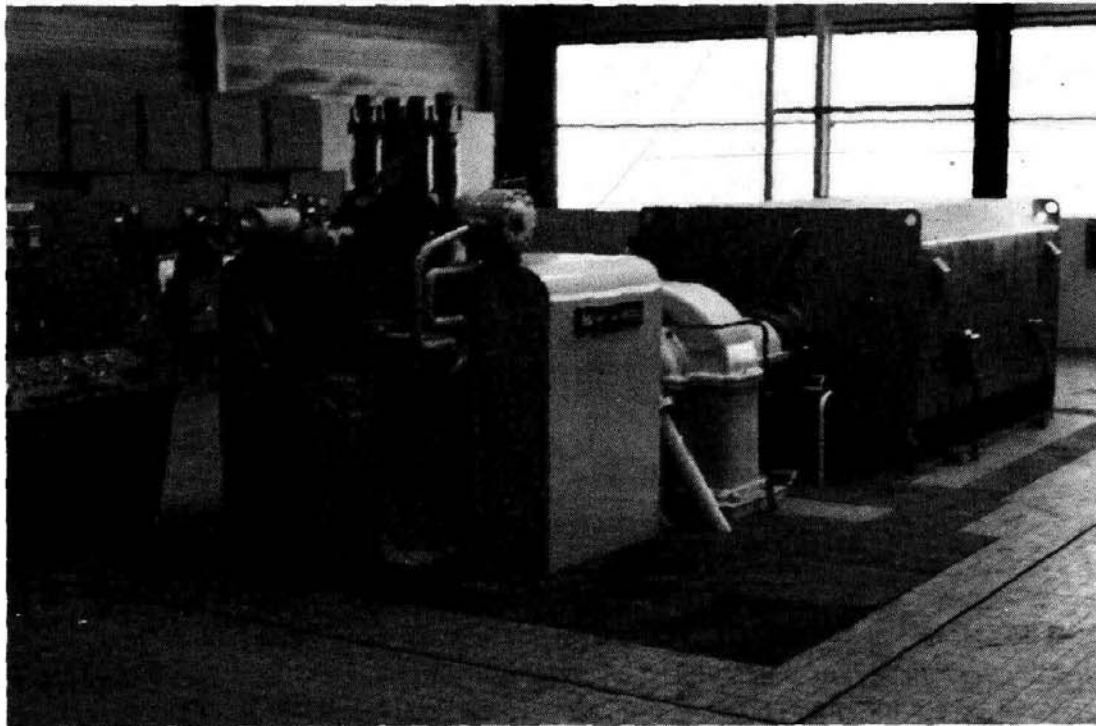


Fig. 43.2). Turbo-generator. 12,800 kW. 7,500/1,500 r.p.m., 38 bar, 375°C. exhaust 2 bar, alternator 15,000 V (Fives Cail – Babcock).

Armour consisting of paper coated with tar and steel sheath
External layer of black p.v.c.

This cable ensures protection of joints in the most severe conditions.

(c) There are other types of cable, armoured or not, the specifications of which have been established by the makers.

B. Power cables. These are the cables serving the motors. Their conductors may be of copper or of aluminium. For carrying equal current, the section of aluminium should be 50% greater than that of a copper cable; taking this fact into account, aluminium cables are lighter and cheaper, but more susceptible to corrosion. The normal series in French industry is designated by:

(7) The letter U, conforms to the standard of Union Technique de l'Electricité (U.T.E.);

(2) A number designating the nominal maximum voltage permissible;

(3) Letters indicating the nature of the constituents, such as:

B = vulcanised butyl rubber

C = rubber (caoutchouc)

E = polythene

H = hypalon

J = impregnated paper

K = silicone rubber

L = ethyl propyl rubber

N = polychloroprene, or equivalent

R = polythene chemically reticulated

V = polyvinylchloride (p.v.c.).

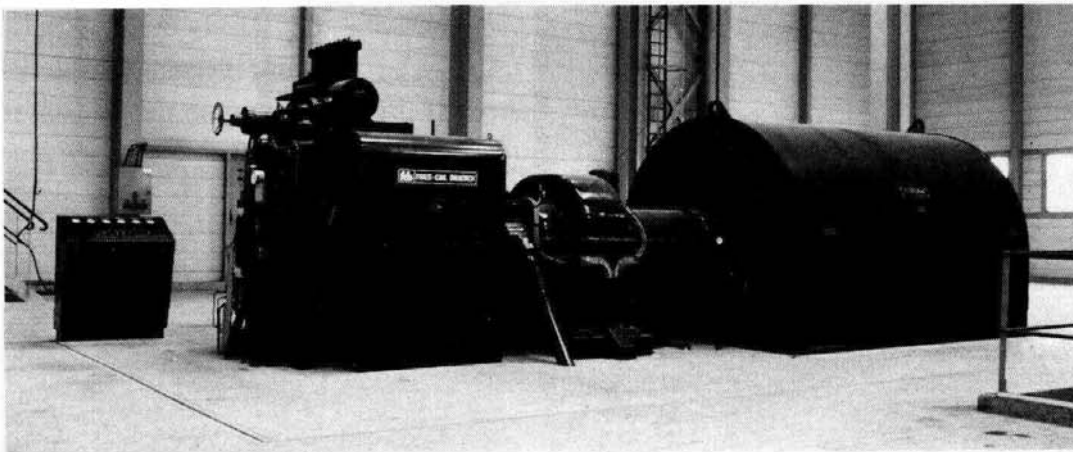


Fig. 43.22. Turbo-generator. 12,800 kW, 7,000/1,500 rpm, 43 bar eff.; 415°C. exhaust 2.5 bar eff., alternator 15,000 V (Fives Cail – Babcock)

For L.T. ($U \leq 1,000$ V), for example, the cable used is U 1000 RO 2 V, which may or may not be armoured; for particularly severe conditions: U 1000 RGPV. For medium tension ($U = 1,000 - 30,000$ V), the cable used may be EdF HN 33 S 23, which may be placed underground, or HN 33 S 22, which should not be directly underground.

The use of aluminium is developing rapidly; it tends to replace copper.

Section of the conductors

The cross-section of the electric wires and cables carrying the current in the factory has to be proportional to the current carried.

It is heating in the wires that limits the carrying capacity; and it must be remembered that a wire becomes heated through its whole mass, whereas it cools down through its surface. Since the ratio of the outer surface of a given length of wire to the section of that wire decreases when the diameter increases, the density of current it can carry decreases when the diameter increases.

The current density is the ratio of the current flowing in the conductor to the section of that conductor:

$$g = \frac{I}{s} \tag{43.25}$$

It is evaluated in amperes per square millimetre.

The current density that can be carried depends on:

(7) The limit of heating permissible. A temperature of 65°C is generally accepted for impregnated paper, 70°C for polyethylene, 75°C for p.v.c., 90°C for p.r.c., or 140°C for silicone rubber.

(2) The ambient temperature, which can leave a larger or smaller margin before reaching the limit temperature. In tropical factories estimates may generally be based on 40°C, except sometimes near a boiler or a hot surface.

Permissible current density. Table 43.8 gives, in amperes, the current density permissible for an L.T. cable suspended freely in the open air. It depends on the insulation and on the protection of the cable. It is stated in amp/mm² of cross-section of each of the three conductors, and is thus to be multiplied by that cross-section.

TABLE 43.8

PERMISSIBLE CURRENT DENSITY

<i>Section (mm²)</i>	<i>Rubber (amp/mm²)</i>	<i>U 1000 RO2V</i>	<i>U 1000 RGPFV</i>
1.5	12	16	15
4	7	10	11
10	5	7	8
25	3.5	5	5.7
50	2.8	4	4.2
120	2	2.8	3.1
240	1.5	2.1	2.4
300	1.4	2	
400	1.2	1.6	

This maximum current density applies for cables in the open air at 30°C. For a different ambient temperature, the figures should be multiplied by the following values of coefficient *k*:

At	30	35	40	45	50	55	60°C
$k =$	1	0.96	0.91	0.87	0.82	0.76	0.71

Similarly, for a cable in the sun:

With clear insulation:	$k = 0.9$
With black insulation:	$k = 0.8$

For adjacent location of cables:

In a group of:		2	3	4 or 5	6-8	9 and above
Horizontal layer:	$k =$	0.85	0.78	0.75	0.74	0.70
Vertical layer:	$k =$	0.80	0.73	0.70	0.68	0.66

L.T. cables, located together in conduit, according to the number:

Number of cables:	1	2	3	4	5	6	7	>7
$k =$	1	0.89	0.80	0.75	0.71	0.66	0.60	0.59

Underground:

In soil at:	20	25	30°C
$k =$	1	0.96	0.93

POWER HOUSE

Electric power required

For mean powers installed, Tromp³ quotes for several power stations in Cuba the figures in Table 43.9.

TABLE 43.9

AVERAGE TOTAL INSTALLED POWER FOR AN ELECTRIFIED FACTORY

Totally electrified	30 kW/t.c.h. = 40 h.p./t.c.h.
Electrified except for mills	15 kW/t.c.h. = 20 h.p./t.c.h.

We recall the values quoted in Table 41.29 (but these concern the mean power used, and not *the* installed power):

	<i>Mills</i>	<i>Other machines</i>	<i>Total</i>
Factories totally electrified	12-20	12-20	20-40 h.p./t.c.h.
Factories with steam-driven mills		12-20	12-20 h.p./t.c.h.

These figures are now much exceeded.

In his figures, Tromp counted for lighting: 0.75 kW/t.c.h. = 1 h.p./t.c.h.

Slack season power. During the slack season, requirements for workshop, distillery, lighting. etc., could be estimated at:

$$1 \text{ h.p./t.c.h}$$

Of course, that value is only a rough indication, since the crushing rate of the factory in t.c.h. can give only a rough idea of the slack season load. Requirements for irrigation pumping may of course greatly exceed this figure.

Transformation of power

We have several times drawn attention to the double transformation of power which takes place in the power station of a factory.

Owing to the losses involved in each transformation; we can say that, *in order to obtain 1 h.p. at the shaft of an electric motor, we must have close to 1 kW at the terminals of the alternator.*

Since the efficiency of an electric motor with \ load varies, according to its speed and power, between 73 and 88%, we may count 80% as an average. Besides, the efficiency of the transmission line is about 0.95 or 0.97. One h.p. applied to the shaft corresponds then to:

$$1 \text{ h.p.} \times \frac{1}{0.80 \times 0.96} \times 0.746 = 0.96 \approx 1 \text{ kW}$$

at the terminals of the alternator.

REFERENCES

- 1 P. MONNIER, *L'Électricité dans l'Usine*, Presses Universitaires de France, Paris, 1934, p. 91.
- 2 D. L. HUGHES, *Proc. 36th Conf. SASTA*, 1962, p. 69.
- 3 L. A. TROMP, *Machinery and Equipment of the Cane Sugar Factory*, Norman Rodger, London, 1936, p. 576.
- 4 P. MONNIER, *op cit.*, pp. 189- 191.

44. Pumps

CENTRIFUGAL PUMPS

The vast majority of pumps in the sugar factory are centrifugal pumps, with direct coupled electric motors (Fig. 44.1).

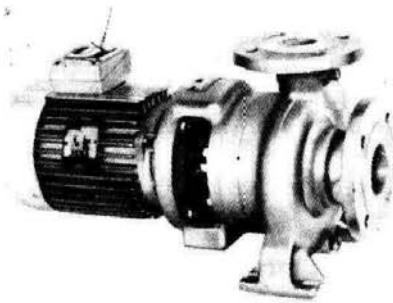


Fig. 44.1. Single stage electric pump "Men-bloc" (Jeaumont - Schneider).

Speed

The main drawback of such pumps is that the speed of the electric motor is fixed once and for all, and cannot be adjusted: 2,900, 1,450, 960 . . . r.p.m. (50 cycles alternating current).

Adjustment of the output

The adjustment of the output has then to be done with a valve, and that valve must always be placed on the delivery side and not on the suction side. Otherwise cavitation would take place leading to rapid wear of the impeller.

The power absorbed by a pump thus adjusted actually decreases with the output, although of course it does not decrease in proportion. But the $\cos \Phi$ of the motor also decreases and could fall very low.

However, adjustment by means of the delivery valve does not always allow the output of the pump to be regulated to suit the continual and sometimes substantial variations required in output.

In order to do away with the continual attention and supervision required of such a valve, it is of advantage to fit each pump with a device consisting of a balanced valve operated by a float, placed in an intermediate tank, and regulating the output of the pump (Fig. 44.2). The output is thus rendered proportional to the quantity of liquid arriving in the intermediate tank *B*. If the level in that tank is above a certain mark *N*, the float closes the valve completely, and all the liquid is forced away through the delivery pipe *T*. If the level comes down below *N*₂, the float opens the valve fully and all the liquid flows in a closed circuit between

the tank *B*, the pump and the valve. In normal operation the valve recirculates a certain proportion of liquid.

This device is very useful, in order to secure satisfactory functioning of a centrifugal pump on mixed juice, for example, to avoid fluctuations in the the feeding of the units placed downstream in relation to the pump.

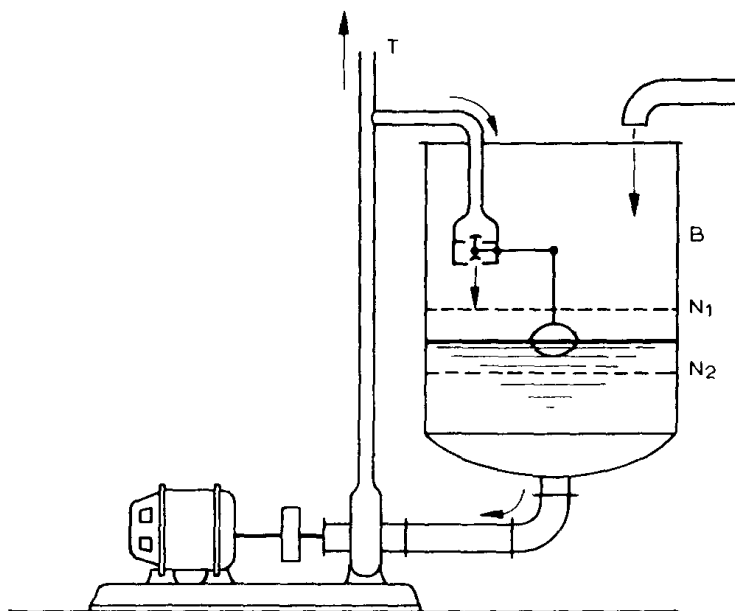


Fig. 44.2. Adjustment of the output of a pump by float-controlled valve.

Choosing the output of the pump

When drawing up a project of an installation which includes centrifugal pumps, the designers tend to provide for pumps the nominal flow of which is 25 or 30% greater than the average output needed.

It is essential to provide a margin, so as to face sudden variations in output, air leakages, minor accidents in operation, etc., but an excessively wide margin sabotages the $\cos \Phi$ of the factory. We consider a margin of 15 – 20% as a maximum is sufficient to avoid all troubles. For instance, in a factory where the volume of juice is 100,000 l/h (12,000 gal./h), we shall consider pumps designed for 120,000 l/h (14,000 gal./h). There is no reason to allow a wider margin; then the impellers have to be changed or repaired when they are worn out. Besides, if the capacity of the factory increases gradually, as is often the case, and certain pumps are found to be barely adequate, it is generally easy to find another use for them, in a different part of the factory.

This remark, of course, does not apply to the boiler feed pumps, for which prudence demands a substantial margin in the output (*cf.* p. 998).

Power

The power absorbed by centrifugal pumps is given by the formula:

$$P = \frac{Q(H + h)}{102} \quad \left(\frac{Q(H + h)}{720} \right) \quad (44.1)$$

P = power absorbed by the pump, in kW

Q = output of water or juice, in kg/s (lb.s)

H = liquid head, total of suction and delivery, in m (ft.)

h = liquid head corresponding to losses of head in the piping, in m (ft.)

Q = volumetric efficiency of the pump.

For the efficiency ρ of the pump, we shall take:

Small pumps: e — 0.50

Medium size pumps: g — 0.60

Large pumps: g — 0.70

Tromp¹ gives the more precise values of Table 44.1.

TABLE 44.1

MECHANICAL EFFICIENCY OF CENTRIFUGAL PUMPS

Output		Efficiency
(l/min)	(Imp. gal./min)	
300- 1,000	75- 250	55-65
1,000- 5,000	250- 1,000	65-70
5,000- 15,000	1,000- 3,000	70-73
15,000- 25,000	3,000- 6,000	73-75
25,000- 50,000	6,000- 10,000	75-80

Multi-stage pumps obviously have a lower efficiency. We must take 0.35–0.45 for ordinary pumps, and somewhat more for special pumps.

Multi-stage pumps

The ordinary single impeller or single-stage pumps can raise water only up to approximately:

$H = 50$ m (165 ft.) when they have no fixed vanes

$H = 100$ m (330 ft.) when fitted with fixed vanes.

Consequently, when the head is greater than 100 m or 10 kg/cm², multi-stage pumps are required. These are simply single-stage pumps arranged in series on the same shaft, one delivering the liquid to the other. In sugar factories they are to be found mainly:

(a) As feed pumps for high pressure boilers.

(b) For pumping juice through a battery of heaters.

(c) As high pressure pumps for hydraulic drive to centrifugals, where this is still used.

(d) As water pumps for delivery to an elevated reservoir.

Suction

Centrifugal pumps are obviously limited where suction height is concerned.

A head of 7m (23 ft.) is the limit, and that only in the most favourable cases.

Hot water further lowers that limit. We take from the **Bréguet** catalogue the graph of Fig. 44.3, giving the maximum suction height admissible as function of the temperature of the liquid. We must notice that from a temperature of 68°C (155°F), the so-called "suction" tank will have to be located above the pump.

The suction height is counted as from the shaft of the pump to the level of the liquid in the suction tank.

Above 3 m (10 ft.) of lift, it will be necessary to fit a foot-valve.

Besides, there is always an advantage in installing centrifugal pumps so that they have a positive suction head, since they do not self-prime automatically as do piston pumps. When working under suction they always risk losing their water if stoppages occur, and re-priming is sometimes difficult.

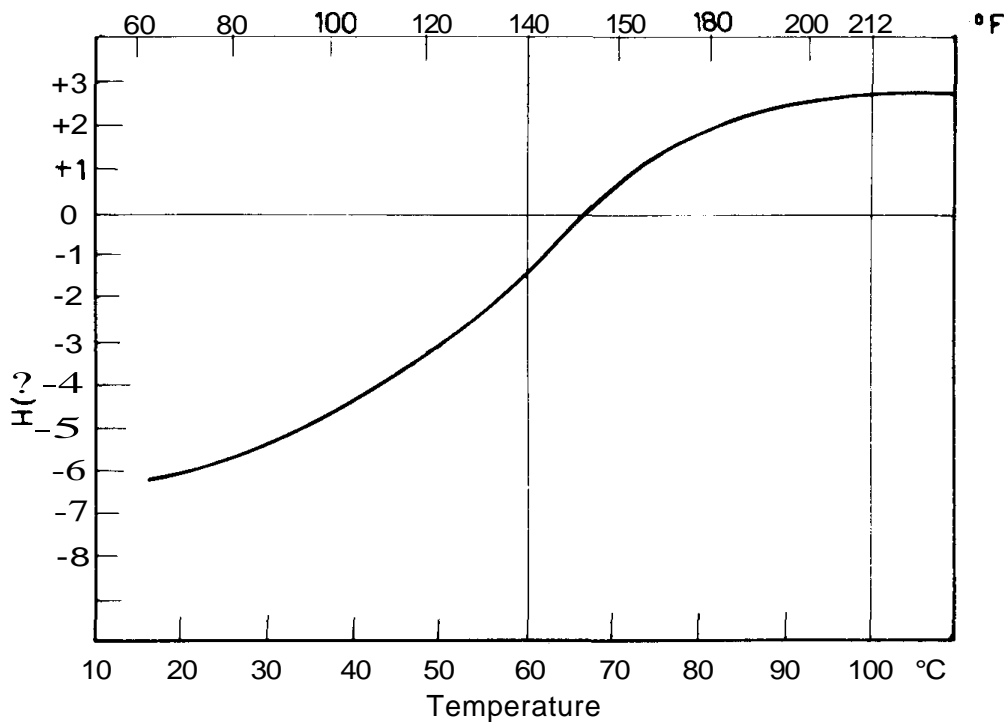


Fig. 44.3. Centrifugal pumps. Maximum suction height (or minimum positive head) according to temperature of liquid.

Net positive suction head

Problems posed by the suction in a pump are resolved by the calculation of the NPSH, that is, the net absolute pressure at the point of minimum pressure in the pump. If this head is negative, it will cause cavitation, rapid wear of the impeller and failure of pumping. The calculated head is then compared with the minimum head required for correct functioning of the pump, and if the difference or the net head is positive, there will be no cavitation. It is advisable to allow a margin of safety. It is convenient to express this pressure in terms of the height of a column of the liquid to be pumped. It may also be expressed in kg/cm² or any other unit of pressure.

The net absolute head at the pump consists of:

(a) The pressure P exerted on the liquid at the tank from which it is pumped. If this is open to the atmosphere, P is the atmospheric pressure.

(b) The vapour tension 0 . This is a negative element which can decrease the available pressure and which must be deducted from the preceding figure.

(c) The suction lift h , that is the difference in level between the axis of the pump and the surface of the liquid in the tank feeding it. This is also a deduction, unless the pump is under a positive head, in which case this height will be added to the pressure P .

(d) The loss of head J corresponding to the output of the pump, also a deduction.

These four elements are to be taken into account, following the choice effected, either in height of liquid or in other pressure units.

Example. Consider a factory at sea level, which requires a pump to pump syrup from ground level to the second floor of the factory, the difference in level between the suction tank and the delivery tank being 8 m (this however does not influence the NPSH). The syrup is at 60°C and its density is 1.32. The pump is 0.9 m above the surface of the syrup in the tank, the suction pipe is 125 mm (5 in.) in diameter; its length is 6 m (20 ft.), and the quantity to be pumped is 60 t/h.

NPSH available: (a) pressure at the lower tank; it is open, so this is atmospheric pressure; taking that as 770 mm of mercury:

$$P = \frac{770 \times 13.6}{1.32 \times 1,000} = 7.93 \text{ m} \quad (26 \text{ ft.})$$

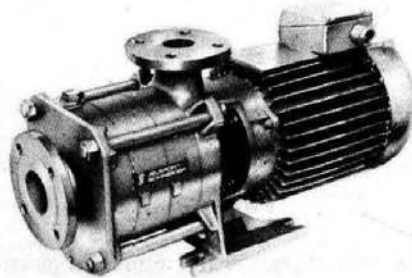


Fig. 44.4. Multi-stage electric pump FM (Jeumont - Schneider).

(b) Vapour tension: at 60°C (Fig. 44.3). it is 2.3 m of water, or in syrup of density 1.32:

$$0 = \frac{2.3}{1.32} = 1.74 \text{ m} \quad (5.72 \text{ ft.})$$

(c) Suction head: this is precisely the difference in level: pump minus tank:

$$h = 0.90 \text{ m} \quad (3 \text{ ft.})$$

(d) Loss of head: the flow of syrup per second is, by weight:

$$q = \frac{60,000}{3600} = 16.7 \text{ kg/s}$$

and, by volume:

$$q' = \frac{16.7}{1.32} = 12.6 \text{ l/s} \quad (0.45 \text{ cu.ft./s})$$

The velocity u of the liquid in the suction pipe:

$$u = \frac{12.6}{0.7854 \times 1.25^2} = 10.29 \text{ dm/s} = 1.029 \text{ m/s} \quad (3.32 \text{ ft./s})$$

The kinematic viscosity ν (Table 45.1 or more complete table):

$$\nu = 12 \times 10^{-6} \text{ m}^2/\text{s} \quad (129 \times 10^{-6} \text{ sq.ft./s})$$

The Reynolds number is thus:

$$R = \frac{uD}{\nu} = \frac{1.029 \times 0.125}{12 \times 10^{-6}} = 10,720 \quad (\text{dimensionless})$$

Formula (45.11) gives:

$$X = \frac{1}{\sqrt{100 \times 10,720}} = 0.031$$

Hence the loss of head:

$$h = 0.031 \times \frac{1.029^2}{19.6} \times \frac{6}{0.125} = 0.08 \text{ m} \quad (0.26 \text{ ft.})$$

We have finally:

$$\text{NPSH available: } 7.93 - 1.74 - 0.90 - 0.08 = 5.21 \text{ m} \quad (17.02 \text{ ft.})$$

The suppliers generally indicate in their catalogues the NPSH necessary for proper operation of the pump, as a function of the capacity. It is necessary to compare the available NPSH with the required figure, to verify that it is higher and that the safety margin is sufficient.

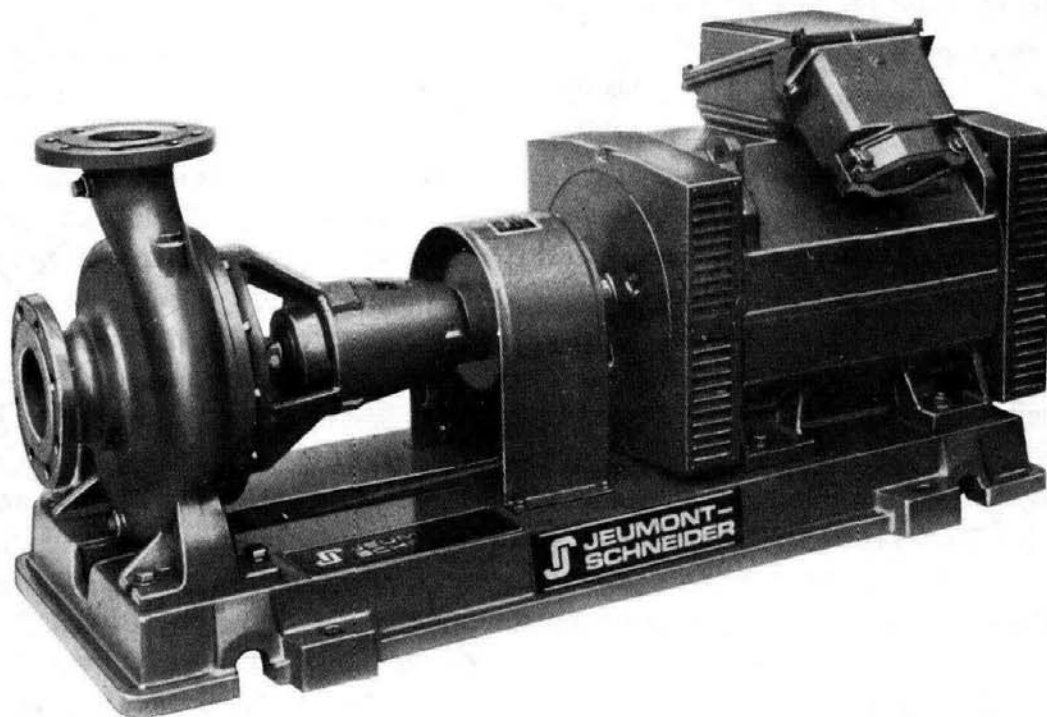


Fig. 44.5. Single-stage "Men" pump with protected motor (Jeumont-Schneider).

Unchokable pumps

These are centrifugal pumps specially designed so as to pass pieces of bagasse the size of those which drop from the mills, between the feed plate or trash plate and the rollers.

They are sometimes called also "dredging pumps" or "chokeless" pumps.

They have necessarily a very low efficiency, and cannot be regulated by a valve without risk of obstruction. Consequently, they should be fitted with a variable speed motor, or an airleak to the suction, to regulate the output.

Coulter, an engineer at Naalehu (Hawaii) advises the use of pumps with a capacity 25% greater than the maximum quantity of juice they may have to handle, and with a power 50% greater than the maximum theoretically needed.

Storrar² considers that no pump should be classed as unchokable unless its suction and delivery openings are at least 7.5 cm (3 in.) in diameter. It follows that, in practice, unchokable pumps cannot be used in factories of less than 40 t.c.h., especially those using less than 15% of imbibition. We would even advise raising this limit to 80- 100 t.c.h. and 20%.

ROTARY PUMPS

Use

Molasses can be handled by gear-pumps (Fig. 46.6), which are very simple and rugged and are now constructed with direct coupled motor drive (Fig. 44.7). They do not require much

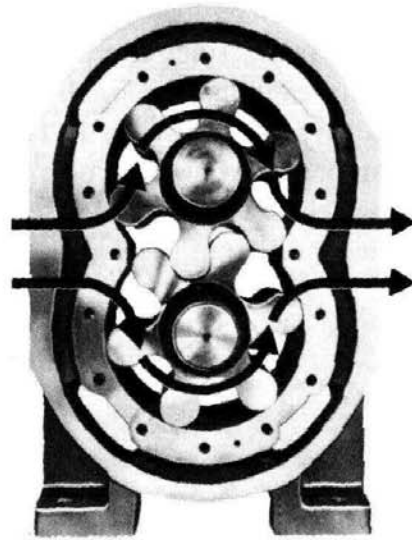


Fig. 44.6. Principle of a gear pump for massecuite and magmas (Broquet).

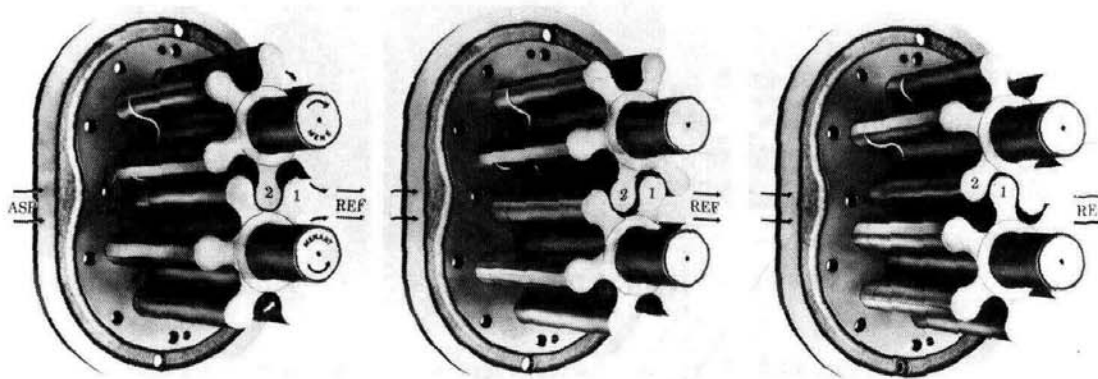


Fig. 44.7. Diagram showing why crystals cannot be crushed in Broquet pump.

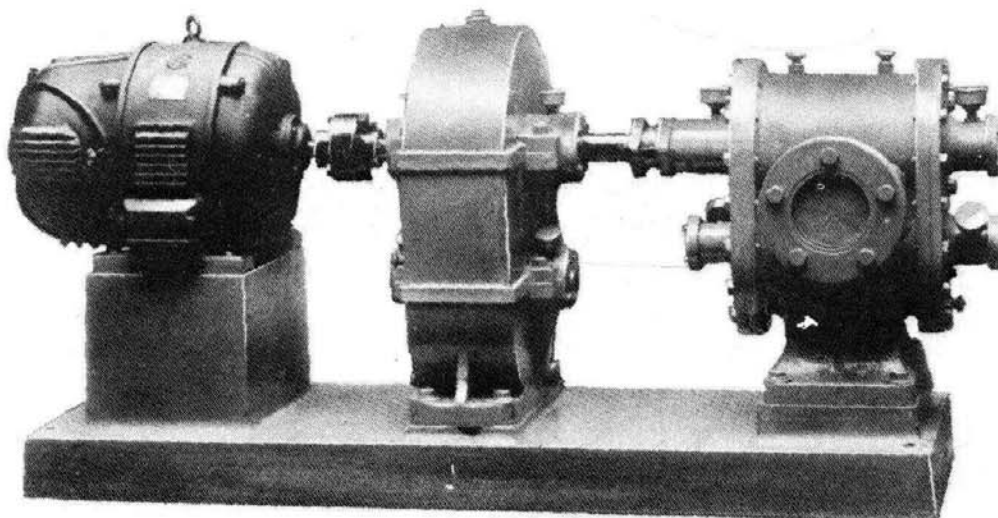


Fig. 44.8. Motor-driven gear pump for molasses and massecuite (Broquet).

attention and we shall not study them here.

Massecuite is more difficult to handle, since crystal damage must be avoided. It may, however, be satisfactorily handled in gear pumps especially manufactured (Figs. 44.8 and 44.9) as well as in pumps of the "Rota" type (Figs. 44.10, 44.11 and 44.12).

It is a positive displacement pump, consisting of a rotor of elliptical section rotating inside the casing of the pump, and on which rests a spring-loaded scraper (Figs. 44.10 and 44.11). Its operation can be readily understood from the figure.

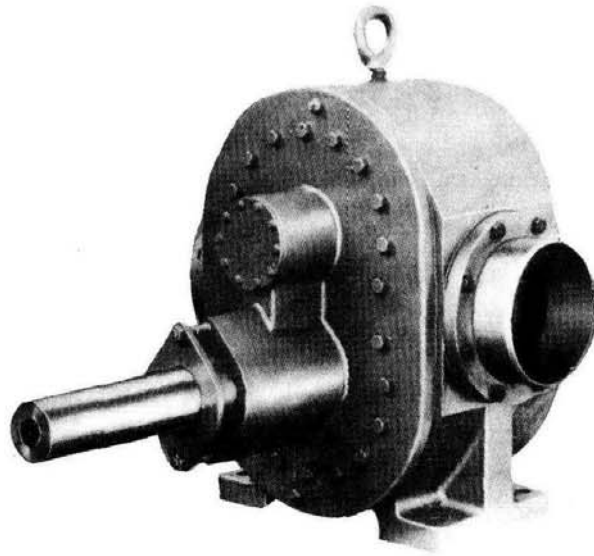


Fig. 44.9. Pump for massecuite and magmas (Broquet).

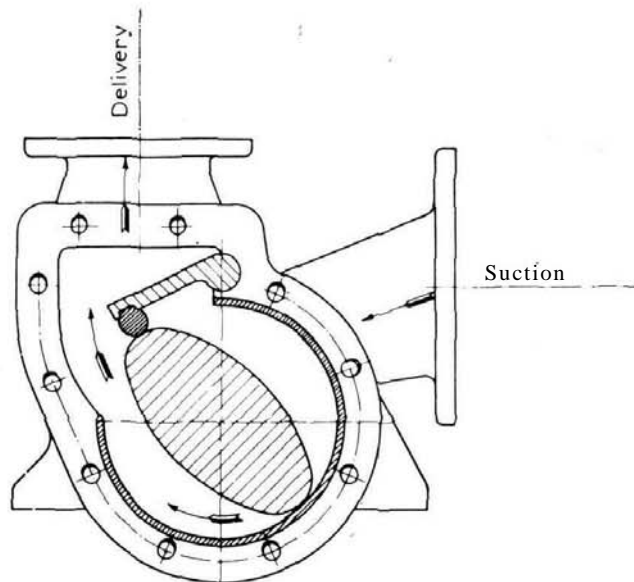


Fig. 44.10. Rota pump. Section.

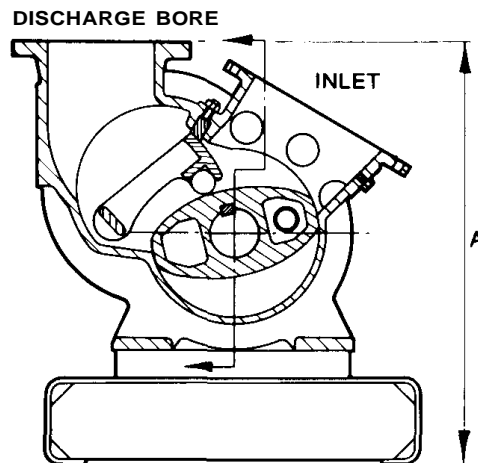


Fig. 44.11. Pump for masecuite (Fletcher and Stewart).

Speed

These pumps are usually supplied either under the form of a complete set with motor, or with a reducing gear and driving pulley.

Speed reduction between pulley and pump is preferably done by Vee belts, or by gear with a leather pinion. In the case of the motor-driven pump there was previously a first reduction by worm-gear, which lowered the efficiency considerably, but much simplified the speed reduction. Today such sets are sold with a coaxial speed reducer, the efficiency of which is much higher.

The speed of the rotor varies from 120 to 150 r.p.m. for molasses and from 30 to 60 r.p.m. for masecuite.

Output

We have:

$$Q = 60 \left(\frac{\pi D^2}{4} - s \right) L n e_v \tag{44.2}$$

Q = output of the pump, in l/h (cu.ft./h)

D = diameter of the cylindrical body, in dm (ft.)

s = section of the rotor, perpendicular to the shaft and expressed in dm^2 (sq.ft.)

L = width of the barrel, parallel to the shaft and expressed in dm (ft.)

n = speed of the rotor, in r.p.m.

e_v = volumetric efficiency.

We shall take for:

Molasses: $e_v = 0.70 - 0.85$ according to viscosity

Masecutes: $e_v = 0.60 - 0.75$ according to viscosity

Power

The power absorbed by the pump is given by:

$$P = \frac{V(H+h)d}{100 \epsilon_m \epsilon_r \epsilon_M} \quad \left(\frac{1}{138 \epsilon_m \epsilon_r \epsilon_M} \right) \quad (44.3)$$

P = power absorbed by the motor of the pump, in kW

V = volume of the material pumped, in l/s (cu.ft./s)

d = density of the material pumped, in kg/dm³ (lb./cu.ft.)

H = total height between suction and delivery, in m (ft.)

h = loss of head, in m (ft.) of water; for very viscous liquids this term can be very large

ϵ_m = overall efficiency of the pump itself

ϵ_r = efficiency of the speed reducer and of the transmission

ϵ_M = efficiency of the electric motor.

In general:

ϵ_m is in the range of $\left\{ \begin{array}{l} 0.70 - 0.80 \text{ (molasses)} \\ 0.60 - 0.70 \text{ (massecuite)} \end{array} \right.$

ϵ_r is in the range of $\left\{ \begin{array}{l} 0.90 - 0.95 \text{ for two-stage gearing} \\ 0.50 - 0.60 \text{ for one worm reduction} \end{array} \right.$

ϵ_M is in the range of: 0.80-0.90.

Suction and delivery

Primed with molasses, which fills the clearances well, these pumps can lift to 2-3 m (6-10 ft.), corresponding to 3-4 m of water. They can deliver up to 20-30 m height, say 3-5 kg/cm² (60-100 ft. or 45-70 p.s.i.).

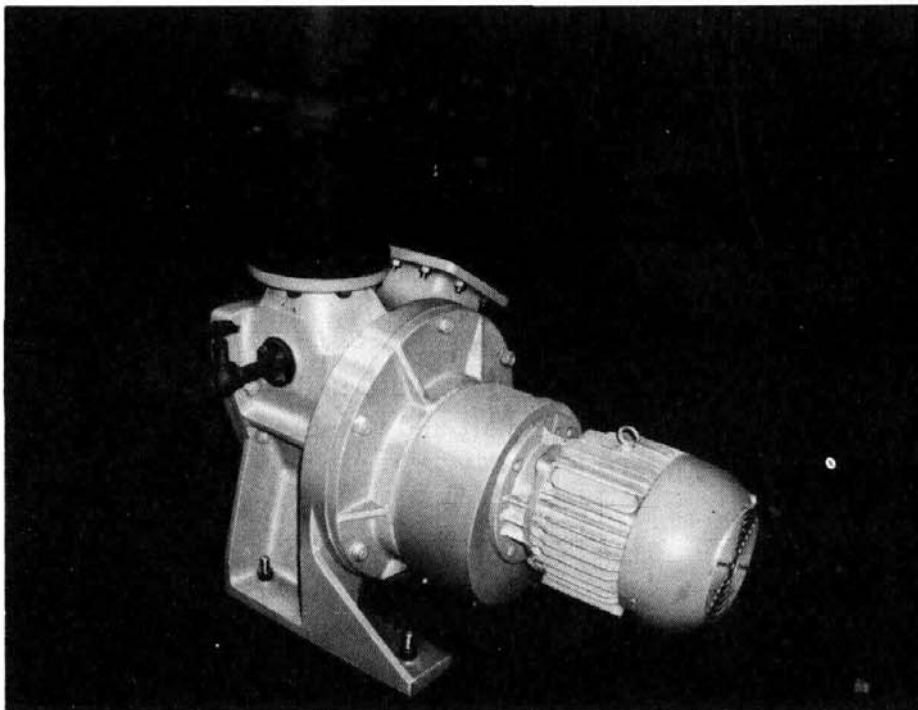


Fig. 44.12. Rota pump (Smith Mirrlees).

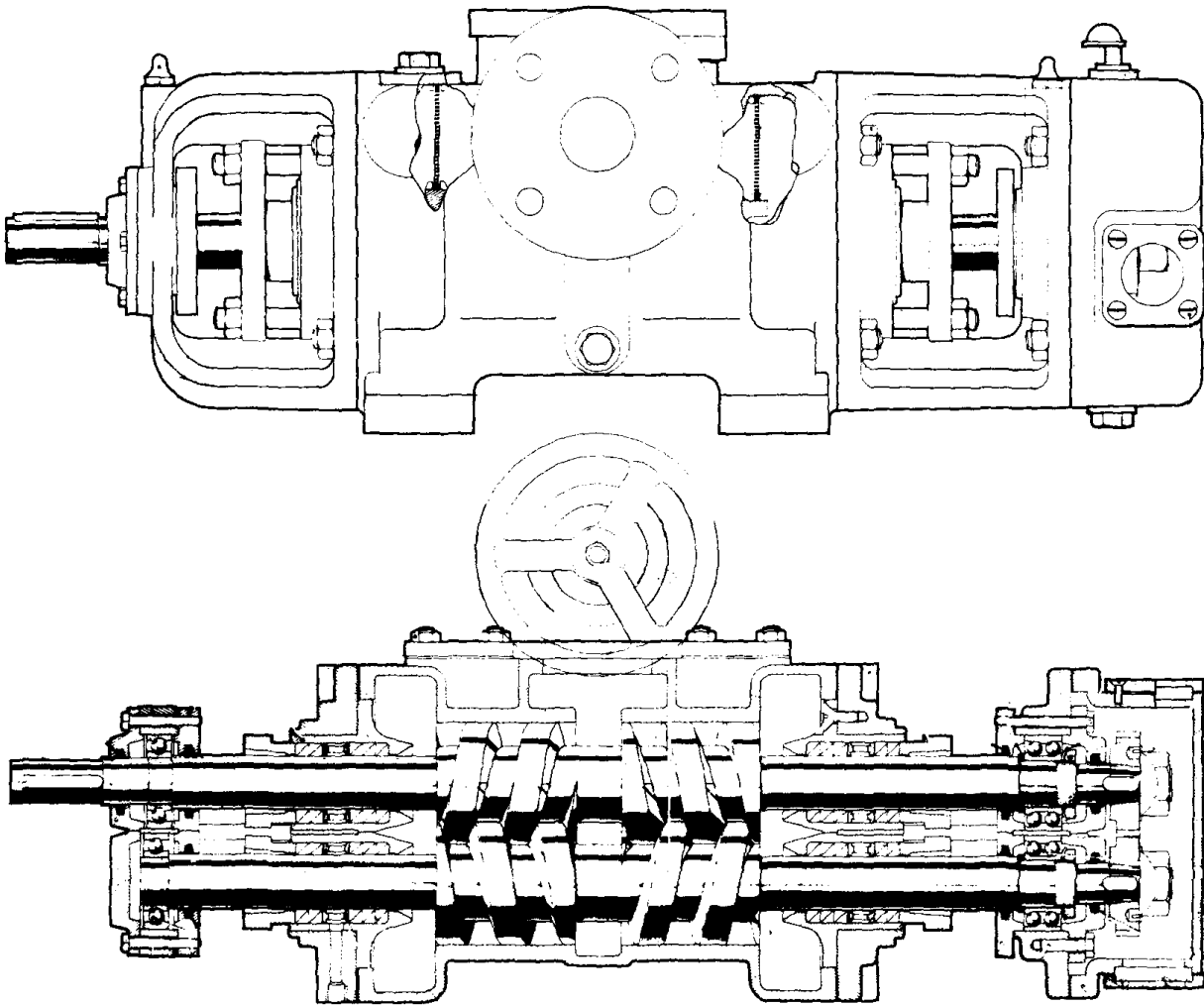


Fig. 44.13. Guinard screw pump, Houttuin type.

Application

They are rather expensive and it is advisable to use them only for massecuites, for which they are very useful, particularly as they do not damage the crystals. Once installed, they require little attention and upkeep. The part which breaks most frequently is the spring which loads the scraper. It is advisable to carry an adequate stock of spares, and to demand a metal of very good quality. In some cases the spring can be replaced by a counter weight, but this expedient is not always successful.

SCREW PUMPS

These are pumps rather analogous to gear pumps, but in which the volume displaced during a rotation corresponds to the free space between two threads of a screw. The type most used in French factories is the Guinard double screw pump (Fig. 44.13): the engagement of one screw with the other assures separation of the individual volumes between successive turns of the screw and consequently prevents leakage, that is it ensures irreversibility of the flow.

These pumps are utilised for molasses; their efficiency is of the order of 0.70.

A pump of similar design, well known in British countries, is the "Mono".

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- 1 L. A. TROMP, *Machinery and Equipment of the Cane Sugar Factory*, Norman Rodger, London, 1963, p. 506.
- 2 T. STORRAR, *Proc. 9th Congr. ISSCT*, 1956, p. 121.

45. Piping and Fluid Flow

The flow of fluids in pipes is expressed by the same laws, whether liquid or gas is concerned. This may appear strange at first sight, but it is due to the important role played by viscosity of the fluid; and since viscosity is defined in terms of resistance to flow, it is due to its effect that the formulae become unified.

Since most problems of gas flow in a sugar factory may be easily resolved by the adoption of recommended velocities, we shall be concerned here mainly with flow of liquids. However, once the viscosity is known, problems can be solved in either case by completely analogous methods.

In the factory, we are concerned mainly with the following fluids: water, juice, syrup, molasses (intermediate and final), steam, carbon dioxide, air and flue gases.

GENERAL FORMULAE

A. Equation of flow

In any pipe transporting a liquid, we have:

$$Q = \frac{\pi D^2}{4} \cdot U \quad (45.1)$$

Q = volume flow, in m^3/s (cu.ft./s)

D = pipe diameter, in m (ft.)

U = velocity of liquid, in m/s (ft./s).

B. Pressure drop

All flow involves friction, and hence loss of energy, described in the present connection as "pressure drop". We have:

$$Ap = \lambda \omega \cdot \frac{l^2}{2g} \cdot \frac{U}{D} \quad (45.2)$$

Ap = pressure drop of the liquid along the pipe, in kg/m^2 (lb./sq.ft.)

X = coefficient expressing resistance to flow (dimensionless)

ω = density of the liquid, in kg/m^3 (lb./cu.ft.)

U = mean velocity of flow of the liquid, in m/s (ft./s)

g = $9.8 \text{ m}/\text{s}^2$ ($32.16 \text{ ft.}/\text{s}^2$)

L = length of pipe, in m (ft.)

D = diameter of pipe, in m (ft.).

It is sometimes convenient to express a pressure drop in head of liquid. We have then:

$$J = \frac{\Delta P}{\omega} = \lambda \cdot \frac{U^2}{2g} \cdot \frac{L}{D} \quad (45.3)$$

J = pressure drop (or loss of head) along the pipe, in m (ft.) head of liquid.

In the same way, the head loss may be expressed per unit length of pipe:

$$j = \frac{J}{L} = \frac{\lambda}{D} \cdot \frac{U^2}{2g} \quad (45.4)$$

j = unit loss of head, in m per m length pipe (ft. per ft.).

(For water, 1 ft. head = 62.4 lb./sq.ft.)

When the flow takes place by gravity, without obstruction or restriction at entry, exit, or along the pipe, the loss of head J is equal to the difference of level:

$$J = Z \quad (45.5)$$

The critical point consists in calculating X , which for laminar flow is a function of the Reynolds number only, but with turbulent flow is also influenced by the roughness of the interior walls of the pipe.

C. Reynolds number

This is a dimensionless coefficient which characterises the nature of the flow, whether laminar or turbulent:

$$R = \frac{UD}{\nu} \quad (45.6)$$

R = Reynolds number

U = fluid velocity, in m/s (ft./s)

D = pipe diameter, in m (ft.)

ν = kinematic viscosity of the fluid = $\eta g / \omega$, in m/s (sq.ft./s)

η = absolute viscosity of the fluid, in (lb.s/sq.ft.)

ω = density of the fluid, in kg/m³ (lb./cu.ft.)

When $R < 2,320$, the flow is laminar, *i.e.* the elements of fluid move parallel to the centreline of the pipe, and the variation of velocity across an axial section of the pipe follows a parabolic law, the maximal velocity being that at the centre.

When $R > 3,000$, the flow is turbulent, or eddying.

When R lies between these two values, the flow is unstable and may be either laminar or turbulent. The values 2,320 and 3,000 are called the upper and lower critical values of the Reynolds number.

D. Viscosity

The kinematic viscosity ν is generally expressed in stokes. In calculations for piping, the M.K.S. system (metre, kilogram-force, second), or the British f.p.s system are generally used; the conversion is readily made:

$$\begin{aligned} 1 \nu (\text{m}^2/\text{s}) &= 10,000 \text{ stokes (cm}^2/\text{s)} \\ 1 \text{ f.p.s. unit (sq.ft./s)} &= 929 \text{ stokes} \\ 1 \nu (\text{mVs}) &= 10.764 \text{ f.p.s. units} \end{aligned} \quad (45.7)$$

The kinematic viscosity is determined by means of a gravity flow viscosimeter (since it is the density of the liquid which then provides the pressure difference).

The absolute viscosity η is generally expressed in poises in the metric system, and conversion to British f.p.s. units is equally convenient:

$$\begin{aligned} 1 \eta (\text{kg s/m}^2) &= 98 \text{ poises} \\ 1 \text{ f.p.s. unit (lb.s/sq.ft.)} &= 14.89 \text{ poises} \end{aligned} \quad (45.8)$$

The absolute viscosity is determined by a viscosimeter operating on an applied force instead of weight of the liquid, the "Höppler" for example. Table 45.1 gives viscosities for water and pure sucrose solutions.

TABLE 45.1

ABSOLUTE VISCOSITY η AND KINEMATIC VISCOSITY ν OF PURE SUCROSE SOLUTIONS

% Sucrose	Temperature (°C)						
	20	30	40	50	60	70	80
	<i>Absolute viscosity $\eta \times 10^4$ (lb.s/sq.ft. $\times 10^4$ or kg s/m²)[*]</i>						
0	1.02	0.82	0.67	0.56	0.48	0.42	0.37
20	5.98	1.52	1.21	0.99	0.83	0.69	0.60
30	3.25	2.42	1.87	1.50	1.22	1.02	0.87
40	6.29	4.46	3.30	2.54	2.01	1.63	1.35
50	15.73	10.31	7.13	5.17	3.88	3.00	2.39
60	59.6	34.5	21.4	14.3	9.85	7.12	5.30
70	490.9	225.9	116.2	65.6	39.8	25.5	17.1
75	2,373	910	397	197	107	62.6	39
	<i>Kinematic viscosity $\nu \times 10^6$ (sq.ft./s $\times 10^6$ or m²/s $\times 10^6$)^{**}</i>						
0	1	0.80	0.66	0.56	0.48	0.42	0.37
20	1.80	1.39	1.10	0.91	0.76	0.64	0.57
30	2.83	2.11	1.64	1.32	1.08	0.91	0.78
40	5.24	3.73	2.78	2.14	1.70	1.39	1.15
50	12.55	8.25	5.73	4.17	3.15	2.44	1.96
60	45.5	26.4	16.5	11.0	7.64	5.55	4.15
70	357.5	165.2	85.3	48.4	29.4	19.0	12.8
75	1,688	644	285	142	77.4	45.5	28.5

* For centipoises, multiply η by 0.98×10^4 .

** For centistokes, multiply ν by 10^6 (hence figures in table are centistokes).

A much more detailed table is available in other publications^{1, 2}.

An impure solution of sucrose has a viscosity much higher than that of a pure solution of the same brix, and very variable according to the nature of the impurities. In the absence of better data, an approximate value for the viscosity of a solution of purity p may be deduced from that of a pure solution of the same brix by taking:

$$\text{Viscosity of the impure solution} = \frac{\text{viscosity of the pure solution}}{p} \times 100 \quad (45.9)$$

It is obviously more prudent and more dependable to measure the viscosity in the laboratory.

Example. Brix of juice 20°, purity 85, at a temperature of 140°F:

$$\nu = \frac{0.76 \times 10^{-6}}{0.85} = 0.89 \times 10^{-6}$$

Calculation of kinematic viscosity. The second part of Table 45.1 does not allow of sufficiently precise interpolation. However, it is possible by means of the graph of Figure 45.1 and the tables of Cane Sugar Handbook² to calculate the kinematic viscosity of a sucrose solution when the temperature and concentration are known.

For example, take a syrup at 60°C and 60° Brix. The specific gravity of this product is given, for 20°C, by Table 16 of ref. 2: $d = 1.28908$, and the specific gravity at 60° by Table 5 of the same Manual. Since the volume at 60° is 1.0172 of its volume at 20°, the specific gravity is:

$$\omega = \frac{1.28908}{1.0172} = 1.2673, \text{ hence density is } 1267.3 \text{ kg/m}^3$$

The graph of Figure 45.1 gives its absolute viscosity:

$$\eta = 9.4 \text{ centipoises} = \frac{9.4}{0.98 \times 10^4} = 0.00096 \text{ kg s/m}^2$$

Its kinematic viscosity is then:

$$\nu = \frac{\eta \cdot g}{\omega} = \frac{0.00096 \times 9.81}{1267.3} = 0.00000743 \text{ m}^2/\text{s}$$

Having assumed Brix equal to sucrose content, we do not find exactly the same values as those of Table 45.1.

E. Resistance coefficient X

This coefficient depends on the nature of the flow, in other words on the Reynolds number R .

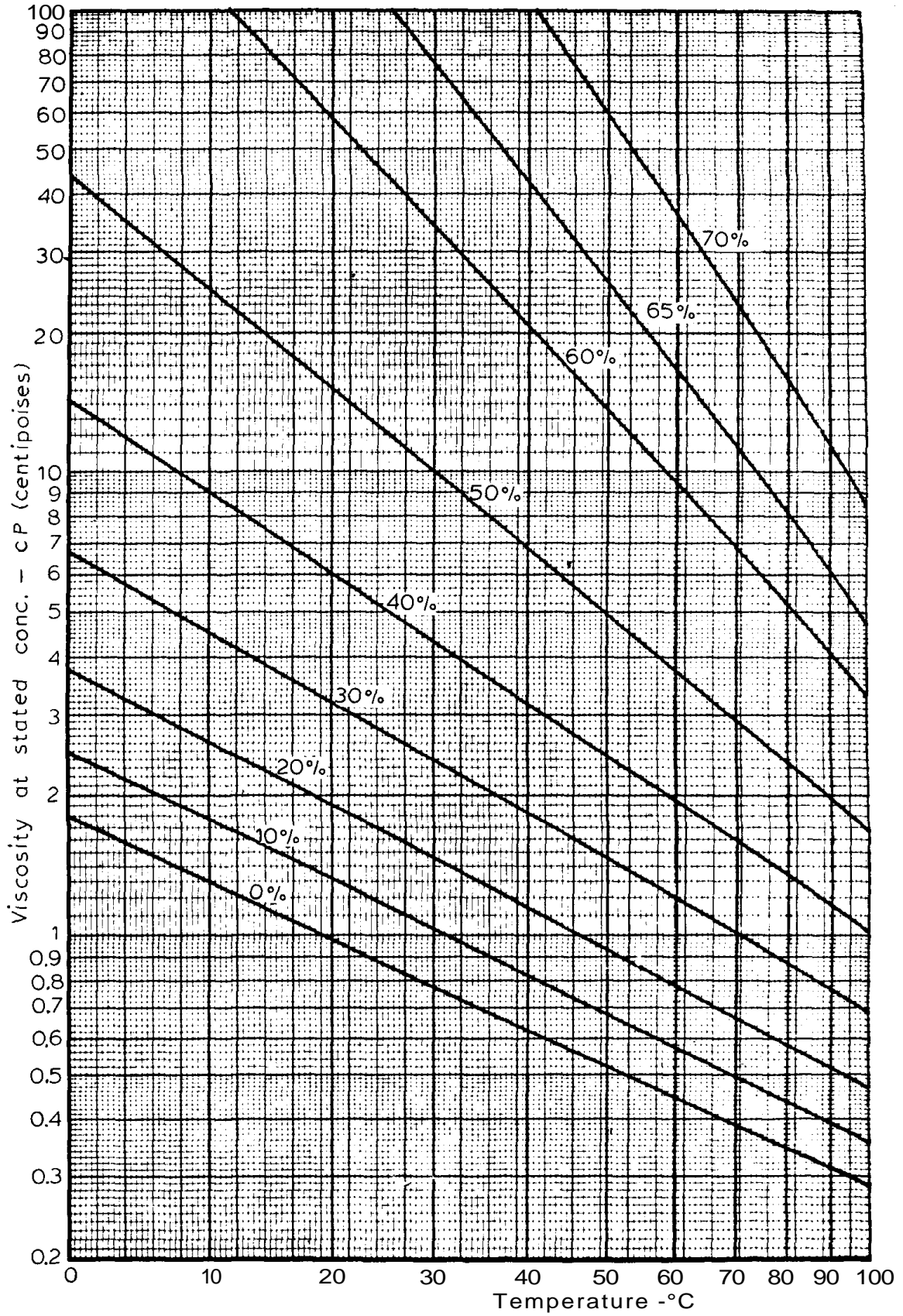


Fig. 45.1. Kinematic viscosity of sugar solutions as a function of temperature and concentration.

(a) **Laminar flow.** We shall take:

$$X = \frac{64}{R} \quad (45.10)$$

(b) **Turbulent flow.** With turbulent flow, X depends also on the state of the interior surface of the pipe.

Smooth pipes. For $R < 10^5$:

$$X = \frac{1}{\sqrt[4]{100R}} \quad (45.11)$$

For $R > 10^5$:

$$X = 0.0032 + \frac{0.221}{R^{0.237}} \quad (45.12)$$

Rough pipes. When the Reynolds number is very high ($10^5 - 10^6$, according to roughness), we may take:

$$X = 0.01 \left(\frac{A}{D} \right)^{0.314} \quad (45.13)$$

D = interior diameter of the pipe, in m (ft.)

A = coefficient having the dimension of length, given by:

Pipe of new metal, relatively smooth:	$A = 1.5$ m	5 ft.
Pipe of new cast iron or steel:	$A = 2.5$ m	8 ft.
Pipe of old and rusted iron:	$A = 5$ m	16.5 ft.

For intermediate cases and lower Reynolds numbers, X varies over a relatively lower range and assumes a value intermediate between those applying to smooth and rough pipes.

Other sources of pressure drop

Further pressure drop is caused by bends (particularly elbows), valves, and changes of cross-section, and these must be taken into account. They may be expressed as an equivalent additional length of pipe. Precise values are given for each case by Oniga³, in a very complete treatise on fluid flow.

Example 1. Checking a piping system

Data: D and J .

Unknowns: U and Q .

Consider the case of a pipe 120 mm diam. and 100 m in length conveying water from one tank to another, the water level in the lower tank being $Z = 2$ m below that in the upper tank.

We have (eqns. 45.4 and 45.5):

$$j = \frac{Z}{L} = \frac{2}{100} = 0.02 \text{ m/m (ft./ft.)}$$

We require to know the mode of flow (laminar or turbulent), although we do not yet know the liquid capacity. We shall accordingly proceed by trial and error. We have (eqn. 45.6):

$$R = \frac{UD}{\nu} = \frac{0.120}{0.9 \times 10^{-6}} = 0.133 \times 10^6 U$$

It is to be expected that R would be very high, greater than 10^5 . In this case, we would have (eqn. 45.13):

$$X = 0.01 \sqrt[0.314]{\frac{1}{0.12}} = 0.01 \times 2.75 = 0.0275$$

We should thus have:

$$j = \frac{X}{D} \frac{U^2}{2g} \text{ or: } 0.02 = \frac{0.0275}{0.120} \frac{U^2}{2 \times 9.8}$$

Hence:

$$U^2 = \frac{0.02 \times 0.120 \times 19.6}{0.0275} = 1.71 U = 1.31 \text{ m/s} \quad (U = 4.447 \text{ ft./s})$$

Checking the mode of flow:

$$R = 0.133 \times 10^6 \times 1.31 = 174,230 (> 10^5)$$

The flow is then (eqn. 45.1):

$$Q = \frac{\pi D^2}{4} \cdot U = \frac{3.14 \times 0.12^2 \times 1.31}{4} = 0.0148 \text{ m}^3/\text{s} = 14.8 \text{ l/s} \quad (0.6 \text{ cu.ft./s})$$

If we had found a value of R leading to a difference in the conditions assumed, or falling between the upper and lower critical figures, it would have been necessary to repeat the calculation, assuming different figures.

Example 2. Calculation of diameter of a pipeData: Q and j .Unknowns: D and U .

The tank is situated 12 m above a juice tank and a flow Q of 5 l/s of juice is required. The length of the pipe is $L = 20$ m. What diameter D should the pipe have to obtain the required flow?

We have:

$$f = \frac{Z}{L} = \frac{12}{20} = 0.6$$

We require the value of D which will give us this value of f .We shall try $D = 40$ mm, and calculate the velocity U (eqn. 45.1):

$$U = \frac{4Q}{\pi D^2} = \frac{4 \times 5}{3.14 \times 0.04^2} = 3.98 \text{ m/s} \quad (13.04 \text{ ft./s})$$

Hence the Reynolds number (Table 45.1, assuming brix = 20°, and $t = 40^\circ\text{C}$):

$$R = \frac{UD}{\nu} = \frac{3.98 \times 0.040}{1.10 \times 10^{-6}} = 144,690$$

This is well in the turbulent region. If the pipe is smooth, we may take (eqn. 45.12):

$$X = 0.0032 + \frac{0.221}{R^{0.237}}$$

$$\log R^{0.237} = 0.237 \log R = 0.237 \times 5.16043 = 1.22302. \quad R^{0.237} = 16.712$$

$$X = 0.0032 + \frac{0.221}{16.712} = 0.0032 + 0.0132 = 0.0164$$

Hence (eqn. 45.4):

$$j = \frac{X}{D} \cdot \frac{U^2}{2g} = \frac{0.0164}{0.04} \times \frac{3.98^2}{2 \times 9.8} = 0.33$$

Trying $D = 30$ mm, we find in the same way: $j = 1.32$, which is greater than the head (0.625) available. Hence the diameter required lies between 30 and 40 mm; we shall adopt 40 mm.

Pumping. The case of a pipe for pumping liquid is much simpler, as there are several solutions possible according to the pressure drop chosen. A velocity U from the table of normal velocities will be chosen; from this the diameter D is calculated, hence the Reynolds number

R , and the resistance coefficient X and so the total head loss J . A diameter and a velocity giving an acceptable heat loss will then be chosen.

When water, juices, syrup or molasses are transported in the sugar factories by means of a pump or even, for a short distance, by gravity, such calculations for the piping are not generally undertaken. It is sufficient to choose a pipe diameter such as to give a normal liquid velocity (see Table 45.2).

TABLE 45.2

NORMAL VELOCITIES OF FLOW OF LIQUIDS

	<i>(m/s)</i>		<i>(ft./s)</i>	
	<i>Suction</i>	<i>Delivery</i>	<i>Suction</i>	<i>Delivery</i>
Water	1 - 1.25	1.25-2.50	3 - 4	4 - 8
Juice	1 - 1.20	1.20-2	3 - 4	4 - 6
Syrup	0.50 - 1	0.75 - 1.25	1.4-3	2.5-4
Molasses	0.25 - 0.50	0.50-0.75	1 - 1.5	1.5 - 2.5
Massecuities	0.10-0.20	0.15 - 0.30	0.3-0.7	0.5 - 1

FLOW OF STEAM AND GASES

Flow of gases, and in particular of steam, is expressed precisely by the same laws as apply to the flow of liquids, and follows the same formulae given above. We give (Tables 45.3 - 45.5) the necessary properties of the principal fluids handled in the sugar factory.

Normal velocity for steam

In the sugar factory, steam pipes are seldom calculated by determining pressure drop. It is generally considered sufficient to adopt a diameter giving a velocity considered normal for steam. However, there is not general agreement on these normal values, and we give below (Table 45.6) those generally accepted by European designers as well as those employed by American manufacturers and indicated by Tromp⁷.

The longer the distance the steam has to flow, the lower the velocity we shall choose for the steam, so as to avoid excessive pressure drops.

Hausbrand⁸ gives formulae for calculating pipe sizes by determining the pressure drop. Those formulae are similar to those given above, and like them are little used in practice. However, for a given velocity of steam and a given pressure, the pressure drop increases rapidly when the diameter of the pipe decreases. From the point of view of the loss of head, which is obviously the most important consideration, it is therefore logical to adopt a higher velocity for the steam as the section of the pipe becomes greater. In that respect the values given in Table 45.6 are an oversimplification, and should be replaced by velocities taking into consideration the diameter of the pipe.

This can be very simply done by adopting the general formula:

$$V = 5 \sqrt{D} \quad (26 \sqrt{D}) \quad (45.14)$$

V = speed of the steam, in m/s (ft./s)

D = diameter of the pipe, in cm (in.).

TABLE 45.3

DENSITY AND VISCOSITY OF AIR

Temperature (°C)	20	40	60	80	100
(°F)	68	104	140	176	212
Density (kg/cm ²)	1.205	1.129	1.060	1	0.946
(lb./cu.ft.)	0.0752	0.0704	0.0661	0.0624	0.0590
Absolute viscosity η (10 ⁻⁶ kg s/m ²)	1.856	1.953	2.046	2.140	2.227
Kinematic viscosity ν (10 ⁻⁶ m ² /s)	15.10	16.95	18.94	20.95	23.10

Example. Viscosity ν of air at 100°C (212°F): $\nu = 0.0000231$ mVs =
0.0000231 x 10.764 = 0.000249 f.p.s. unit (see eqn. (45.7)).

TABLE 45.4⁴⁵KINEMATIC VISCOSITY ν OF STEAM (10⁻⁶ m²/s)

Absolute pressure p (kg/cm ²)	1	2	6	10	20	30
(p.s.i.g.)	14.7	28	85	142	285	427
Gauge pressure p' (p.s.i.g.)	0	14	70	127	270	412
100°C 212°F	22					
120°C 248°F	24.8	12.3				
160°C 320°F	30.2	15.1				
200°C 392°F		18	5.1	3.8		
220°C 428°F			6.7	4.2	2.4	
300°C 572°F			9.1	5.7	3.1	2.4
350°C 662°F				6.7	3.7	2.9
400°C 752°F					4.4	3.3

Example. Viscosity ν of steam at 427 p.s.i.a., 752°F : $\nu = 0.0000033$ x 10.764 = 0.0000355 f.p.s. unit.

TABLE 45.5⁶KINEMATIC VISCOSITY ν OF CO₂ AND SO₂(10⁻⁶ mVs, at 760 mm Hg)

Temperature (°C)	50	100	150	200	300
(°F)	122	212	302	392	572
CO ₂	9.7	12.8	16.3	20.1	28.6
SO ₂	5.7	7.6	9.8	12.3	17.6

TABLE 45.6

NORMAL VELOCITIES FOR STEAM AND GASES

	(m/s)		(ft./s)	
	American practice	European practice	American practice	European practice
Superheated steam	40 - 75	40 - 65	130 - 250	130-210
Saturated steam	25 - 35	25 - 30	80 - 120	80 - 100
Exhaust steam	30 - 45	30 - 40	100 - 150	100- 130
Bled vapour	35 - (50)	35 - 50	120 - (160)	120 - 160
Vapour under vacuum	(45) - 60	50 - 80	(150) - 200	160- 250
Compressed air	20 - 30		60 - 100	
Air under vacuum	12 - 18		40 - 60	
so ₂	20 - 30		60 - 100	
CO ₂	12 - 30		40 - 100	

This formula may be applied for all steam pressures. In reality the coefficient 5 (26) should be replaced by a coefficient increasing slightly with the pressure of the steam.

In practice, a calculation will first be made from eqn. (45.14), then the velocity indicated by Table 45.6 will be adopted, taking advantage of the range given in the table to approach the value furnished by eqn. (45.14). In the case of a large discrepancy, the complete calculation should be made.

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46. *Automation and Data Processing*

It is not easy to consider, in the space available in this chapter, all the questions arising from automation and data processing in the manufacturing process. We are restricted to a consideration of the essential elements, considering concrete cases and separating from them, as far as possible, various problems.

The classic example of automation of the sugar boiling process will be examined first; this will permit certain difficulties to be sorted out, so as to locate priorities and draw several conclusions.

We shall then be equipped to tackle a more general plan and consider the problem of automation in its modern context. In this connection, we first pay attention to the needs experienced and the methods available for operation, then we aim to show how practical solutions can be reached. The important criteria will be applied in the search for an effective system, characterized by a structure of which details will be given.

At this stage in the development of this treatment, it is appropriate to consider an example, which will treat the technical aspects of the crystallization station. This consideration of a concrete case finally allows us to emphasise the importance of the control of manufacture in general and to indicate how we can combine operation and control.

A HISTORICAL EXAMPLE: OPERATION OF SUGAR BOILING

Analysis of the process

For a detailed description of the process itself, the reader is referred to the chapter on Sugar Boiling. Here, we consider only the aspects of particular interest for our subject, reviewing, for example, the most complete case of graining charges.

Immediately after the formation of grain, the minute crystals which form in the super-saturated thick juice present a very minute surface area. The super-saturation of the solution then depends solely on the conditions of evaporation and feed of the liquor (thick juice or green syrup); the quantity of sugar which deposits on the crystals can, in effect, be considered negligible. During this phase of the process it is necessary to avoid the formation of false grain, by constantly controlling the super-saturation and varying the feed of liquor.

As the operation proceeds, the situation changes. The surface area presented by the crystals becomes more and more important, so that the system *gradually* becomes *self-stabilizing* as regards super-saturation: every increase in the sucrose concentration in the liquid phase causes an increase in the speed of crystallisation, which limits the increase in super-saturation. The objective is to obtain a massecurite containing the desired crystal content at the end of the operation; this can only be obtained by allowing the super-saturation to "float". It is thus necessary to maintain a uniform super-saturation at the beginning of

the operation, and then to control the strike in such a manner that, at the end of the operation, we obtain the desired crystal content.

Control properties (sensors)

It is readily seen that several characteristics of the *massecuite* will, at the beginning of the strike, be only partly influenced by the crystal content, the mass of which is then negligible, while, at the end of the strike, they will be strongly influenced by the crystal content. By choosing a suitable physical property of the *massecuite*, it should then be possible to maintain control, at the beginning of the strike, of the super-saturation and, during the course of the strike, of a parameter which is decreasingly dependent on super-saturation and increasingly dependent on the crystal content. The properties utilised for control of the process will be termed "control properties" or "sensors". The main ones are the following.

Boiling point elevation. This property depends, in effect, on the concentration of solid matter in solution. At the beginning of the strike, it is a good indication of the super-saturation. As the charge proceeds, the concentration of non-sugars in the mother liquor and the consequent increase in solubility of the sucrose means that, at constant super-saturation, the boiling point of the mother liquor increases as exhaustion increases (the effect is self-amplifying for each rise in temperature of the mother liquor involves an increase in the solubility). The value of the boiling point elevation is thus indirectly influenced by the crystal content of the *massecuite*.

This effect depends on the content of non-sucrose. In practice, the Clerget purity of the *massecuite* should be below 88 in order to render the boiling point elevation conveniently useful.

Conductivity. The effects ruling here are analogous to those described in the preceding paragraph. Although the volumetric concentration of ions increases slightly during the course of the strike, the decrease in their mobility (increase in viscosity), due to the increase in proportion of material in solution in the mother liquor, causes a decrease in conductivity.

Experience shows that the latter is a good parameter provided the Clerget purity of the

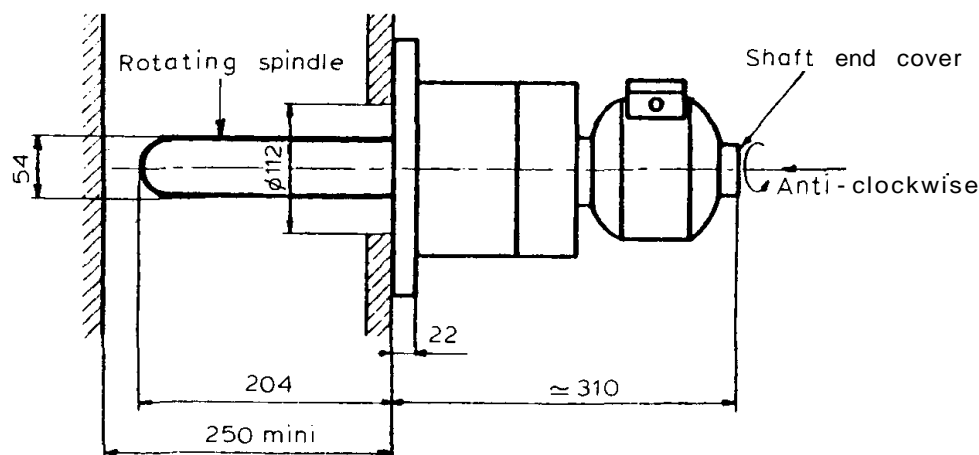


Fig. 46.1. Example of viscosity sensor ("Viscovib" made by Sereg).

massecuite is below 92. The electrodes should, of course, be cleaned periodically; daily cleaning may be necessary.

Viscosity of massecuite. Rheology measurements have given very good results, including those on pure solutions. Siline has, moreover, shown that the values are very much influenced by the crystal content.

Apparatus for determining rheology is mechanical, and consequently rather expensive and involving some maintenance. Various manufacturers, however, offer satisfactory equipment: Siemens, Fischer & Porter, Sereg, etc. (Fig. 46.1).

It will be noted that viscosity and conductivity still give good bases for control even if the vacuum varies slightly, whereas boiling point elevation requires a very constant vacuum to give useful control.

Solutions proposed

As these are familiar we shall recall only their main characteristics. Figure 46.2 shows the operations relative to a graining charge; those corresponding to a magma charge are readily deduced from these.

A sequence of automatic controls applies vacuum to the pan and controls the introduction of footing to the pan. The steam valve is then opened and the operations of concentration,

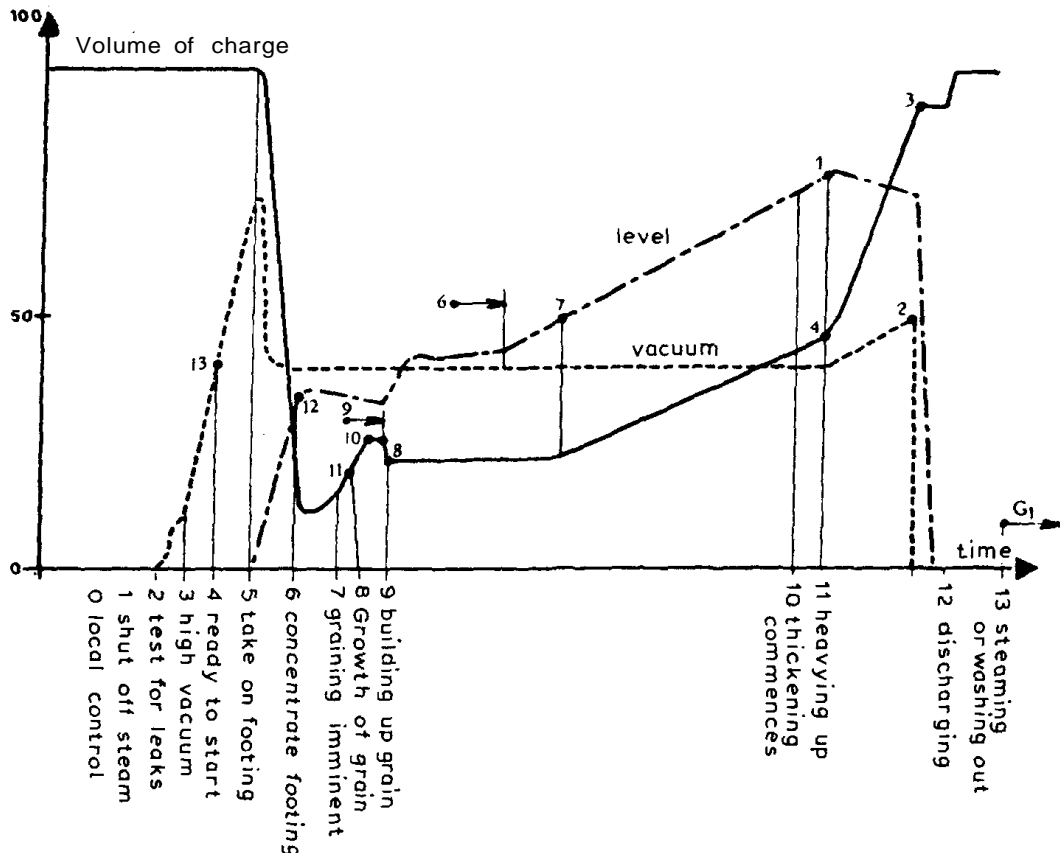


Fig. 46.2. Detailed diagram of graining strike operation.

graining and growth of the crystals take place as indicated in the diagram. The degree of control is then adjusted to definite values depending on the liquid level, by regulating the intake of thick juice or green syrup. Vacuum and steam pressure are generally controlled from start to finish of the operation.

As regards equipment, a control cabinet, or a microprocessor (Fig. 46.3), controls the sequential operations or transmits instructions to the controllers. The actuating devices (valves, jacks, etc.) are driven either directly from the control cabinet, or by controllers.

Conclusions

Several conclusions may be drawn from the preceding example as developed. For satisfactory realisation of automation of pan boiling, it is necessary:

- (a) to analyse the industrial process very accurately
- (b) to establish relations between the magnitudes serving as reference and the state of the massecuite at different stages of the operation
- (c) to study the sequential mechanism of the industrial process in detail.

In all of the following discussion, we shall rediscover the necessity for a very strict analysis



Fig. 46.3a Control cabinet for vacuum pan.

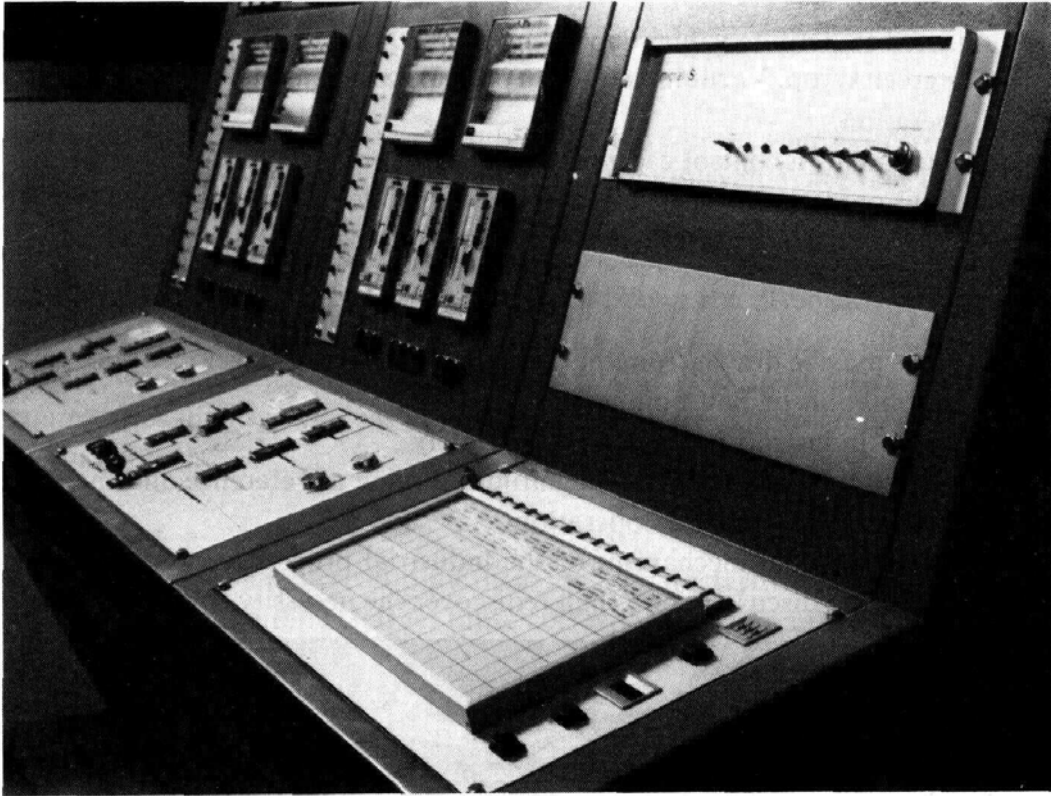


Fig. 46.3b. Example of a boiling control cabinet.

of the conditions but, as we shall be increasingly concerned with the control of items of equipment and complete stations in the factory, the analysis of physical phenomena will become less important than that of the management of the equipment as a whole.

Our example demonstrates the structure of automation: the actuating mechanism receives its orders from the controllers or the control cabinet, the controllers themselves are instructed by the control cabinet and it is the operator (such as sugar boiler) who decides the general starting and operation of the equipment. This order of predominance (hierarchical structure) of the factors will be specified in detail and developed in the following sections (Fig. 46.4).

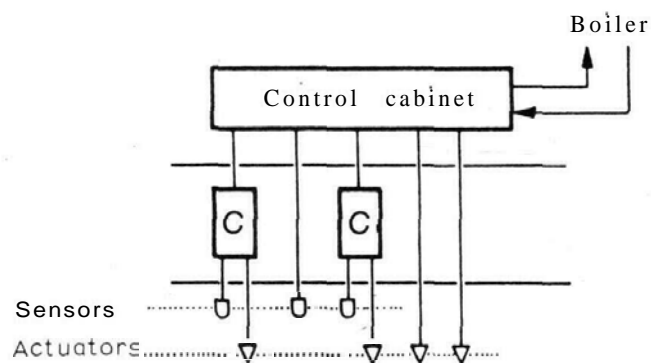


Fig. 46.4. The notion of hierarchical structure (C = controller).

Specific needs of the sugar process

Technical aspects. A fundamental element is the sequence of operation of the various installations. The seasonal character of the process necessitates a choice of arrangements adaptable to continuous service (24 hours per day) for a long period, but also (what at first sight appears paradoxical) for the complete shutdown of the plant for a period of some months.

On the other hand, the complexity of the system as a whole is such that a total interruption, even momentary, of the automation plant, is unacceptable. Solutions have to be found where this is impossible. We shall be prompted in the following text to quote this particular important element explicitly or briefly, several times.

Having indicated these constraints, we may remark, on the other hand, that the time of response in the process is relatively long, which constitutes a favourable element.

Finally, although this may not be the main object of the present chapter, we cannot approach the problems of automation encountered in the sugar factory without quoting the need of this industry in specific sensors. The unstable nature of the raw material, the peculiarities of certain sections of the manufacturing process, the importance of certain auxiliary operations (power production, control of waste products, etc.) justifies the importance attached by specialists to measuring devices which are not found in any other industry. According to the particular case, it may be necessary to modify existing sensors or to study and put into operation new instruments. These different elements, relative to industrial applications, have been illustrated by the example of sugar boiling.

Economic factors. The cost of the equipment is obviously important. However, considerations of efficiency and reliability must never be sacrificed to it; it is also necessary to take into account costs of operation and maintenance. The investments involved are by no means negligible and the installation of the equipment should be undertaken in a progressive manner, fitting in with the general plan.

Problems of personnel. It is equally important to undertake progressive installation, to avoid complete discord between personnel and the process. It is then desirable to plan the phases systematically for the adaptation of personnel who are concerned with the necessary changes.

Evolution of available equipment

Looking back a few years, we see that sugar factories were reluctant to adopt electronic and industrial information processing equipment, a reticence which appears justified. At that stage, electronics were not sufficiently robust for adaptation to the needs of our industry; information equipment, moreover, was cumbersome in size, and its application posed numerous problems. Automatic control of the factory was then the domain of pneumatic devices and associated relays. These techniques have now been well mastered by the factories, so, in the following discussion, we are now mainly concerned with electronic information systems which are in a state of fundamental change.

In the 1970s a veritable technological revolution took place in the field of micro-electronics. The appearance of large-scale integrated circuits (L.S.I.) and their integration in many outlets has had an important impact on industrial applications. It will be seen that this influence was predominantly concerned, in the first place, with the equipment or "hardware" (*i.e.* the units of equipment: the components, sub-assemblies and machines). In this domain, new components are appearing more and more frequently: the micro-processor, an integrated circuit which constitutes the central unit of an automatic control or data processing machine (Fig. 46.5). The equipment developed around it covers the whole field of applications: information sensors, controllers, measuring equipment, programmable sequencers, micro-ordinators, auxiliary equipment, etc. The introduction of the micro-processor in these items of equipment leads to:

- (a) an improvement in reliability
- (b) a reduction of dimensions
- (c) a reduction in cost.

As regards the logic or "software", that is the programming of the tasks undertaken by the equipment, this has lagged slightly, but is still equally substantial. We may, in general, say that the techniques have become industrialised: the methods of operation are becoming systematic and the results to which they lead are sufficiently reproducible to give a precise idea of costs.

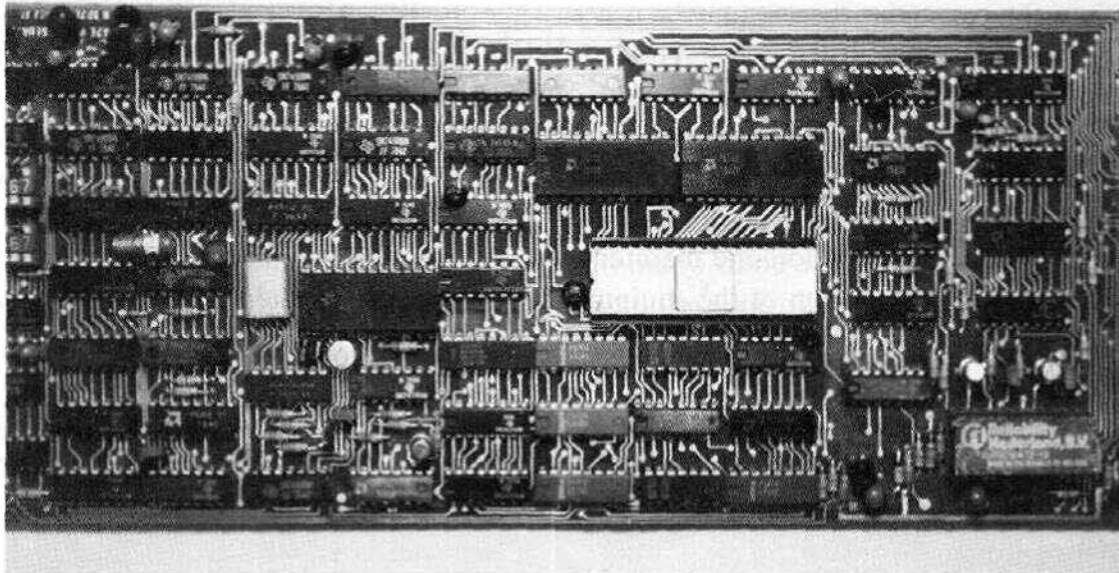


Fig. 46.5. A microprocessor, the heart of a central processing unit.

Some practical recommendations

We indicate below a few factors which are relevant to the industrial application of hardware and software.

Standardisation. Although there is considerable room for improvement in standardisation, it is already possible to select standard components: an example which concerns interfaces

between computerised machines, is the "serial-asynchronous" mode, for which an international standard exists (RS-232C of the Electronic Industries Association, Comité Consultatif International Telegraphique et Telephonique ruling V24, type TTY (Teletype) for current transmission, etc.). This is an important interface mode satisfying most of the industrial requirements discussed above (its poor performance in some aspects, *e.g.* transmission speed, has little importance in industries such as ours) (Fig. 46.6).

Industrial application of hardware. This covers the requirement of efficient systems that are easy to maintain.

These should be based on practical features such as modular elements, that is, where interchangeable parts allow easy development of units with facilities for trouble-shooting and rapid repair of breakdowns.

With regard to inputs and outputs, we must also emphasize the importance of galvanic insulation; an item which has been designed in a laboratory for automation may not be suitable to the environment of industrial production. Insulation for more than 100 volts between analogue channels is sometimes needed in tropical climates.

Industrialization of software. This covers the requirement of systematic methods in "programming" and the provision of what the specialists call "software maintenance", that is, the capability of adaptation to process modification. We advocate the use of a method called "structured programming" for this double purpose, consisting, in particular, of a progressive transfer of the general problem towards detailed solutions; at each stage of the transfer, the task is divided into smaller tasks of which the function is precisely defined. The reader is invited to refer, for more complete information, to specialized works such as "Methods of Programming" by B. Meyer and C. Baudoin, published by Eyrolles, or to the equivalent works in English.

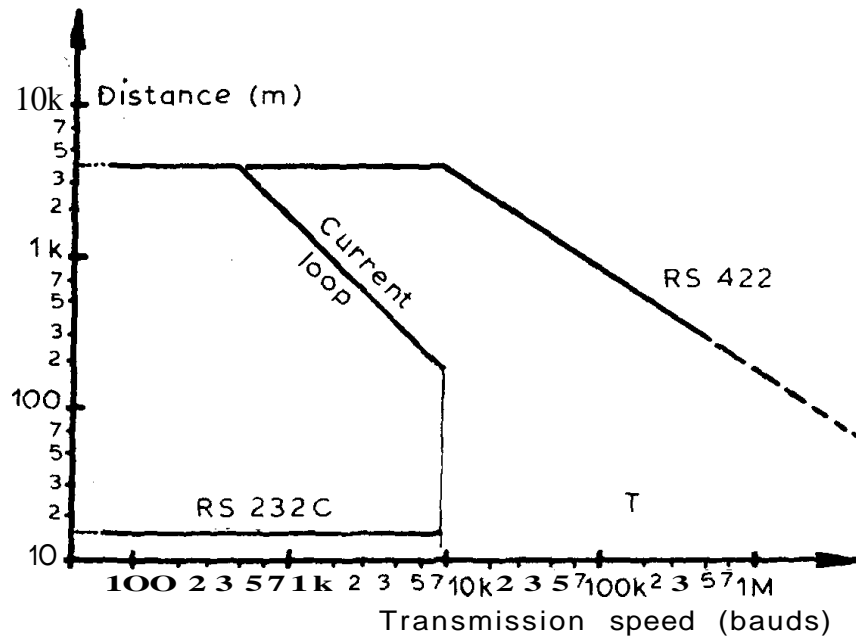


Fig. 46.6. Field of application of serial transmission (the baud is the unit of application of serial transmission = 1 bit/s).

Standardization of language. Once the analysis is carried out, the application must be "coded", that is, translated into a program that the machine can understand. An "advanced language" should be chosen for the purpose, such as FORTRAN, BASIC, PASCAL, etc., which considerably improves the ease of software maintenance. With regard to the sugar industry we suggest the retention of BASIC because of its universal nature (to scientific applications, real time, management, etc.) and particularly because it can be learnt easily by non-specialized personnel.

Importance of dialogue. We shall finally say a few words on the importance of dialogue in modern industrial processes - the exchanges in conversational mode between the machines themselves and also between machines and operators. Preference should be given to methods that specialists call "interactive", that is, organized for question and answer with, for example, menus that offer choices to the users; the instructions for use in some way form part of the machine itself.

A STRUCTURE OF AUTOMATION

Work has been performed and is still continuing throughout the world on automation of the various sections of the sugar-making process. Associations such as ISSCT or CITS report regularly on automation during their conferences and a number of authors have written comprehensive reports on the subject. For our part, we have tried to organize automation and optimization of this industry and have defined an overall approach which we now give as an example.

Organization of automation

The structure offered is primarily based on the process itself and, more precisely, on the traditional division into "stations", for example, extraction, purification, evaporation, crystallization, etc. Rather than adopting a highly centralized organization in which automation is entirely entrusted to a very powerful system, the tasks are divided into a number of facilities of small size which are described as "divided" or "distributed".

Several successive levels of division are defined for organization of the overall automation system. The breakdown may then be as follows:

(a) sensors and actuators providing the basic interface with the process, accepting information and issuing orders

(b) the so-called "basic" automation systems, that is, the traditional controllers or logic automation devices of limited function

(c) the automation systems that we have qualified as apparatus and of which the most typical example is the automated sugar boiling as already discussed

(d) optimal management of the manufacture station

(e) supervision of the whole system

If we examine these two breakdowns of the automation task, we see that the first is very well defined and follows the geography of the plant while the second distinctly depends on the data processed at each level (as progress is made in levels, the criteria relating to technology are replaced by criteria relating to the economic impact).

A good idea of a combination of the two breakdowns can be obtained from the diagram of Fig. 46.7. In the jargon of specialists, this structure is said to be "distributed and hierarchized". It has been given the name of "CHEOPS" (Conduite Hiérarchisée par Echange sur Ordinateurs pour les Procédés Semicontinus) a name that evokes the decision pyramid that it establishes.

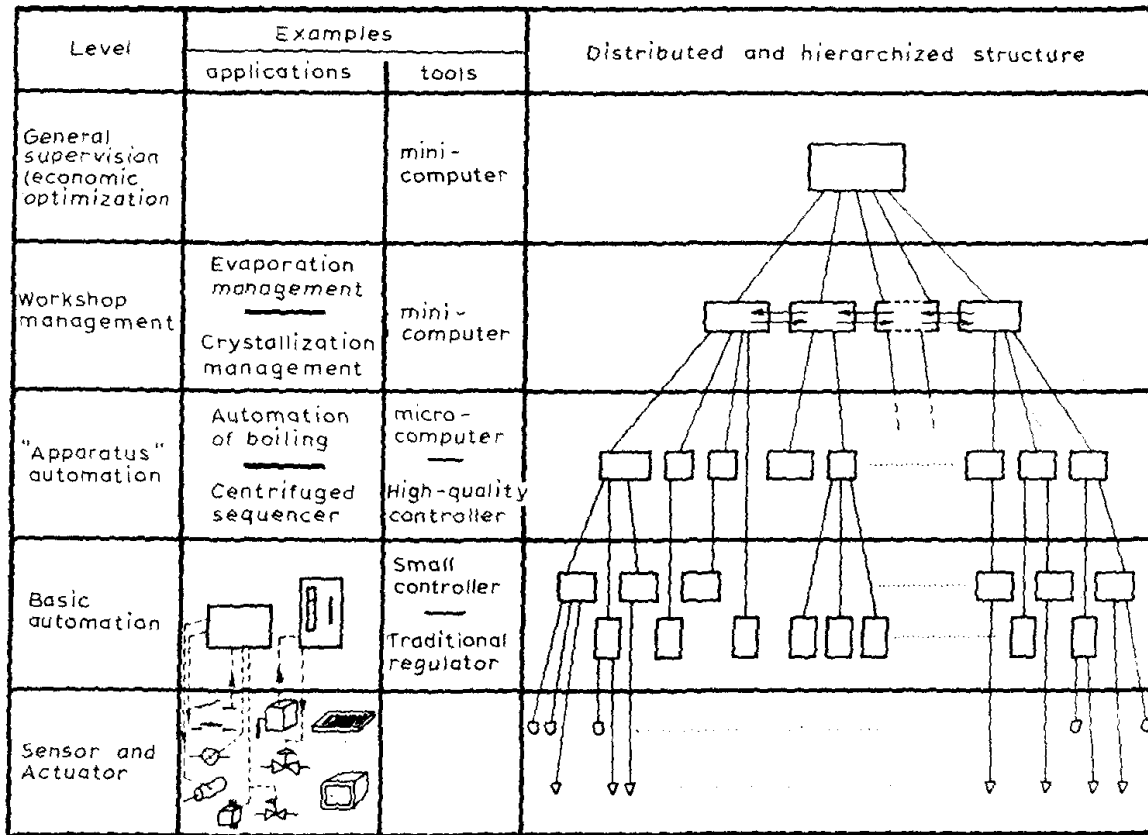


Fig. 46.7. Operational diagram of the "CHEOPS" structure.

It is noteworthy that a given level of automation can be implemented in a plant gradually and without truly starting a study of the higher levels. In order to build the next stage of the pyramid later (or part of the stage) it is sufficient to ensure that the manual *controls* of the automatic equipment can be replaced by signals transmitted from the higher level.

The principal practical advantages of this structure are:

(a) improved reliability of the whole by multiplication of devices (total and simultaneous breakdown is practically impossible)

(b) the possibility of gradual implementation and, consequently, staging of investments

(c) training of operating personnel and of staff in stages.

It will also be seen that it clarifies analysis of the automation units. When large projects are being studied, it allows, in particular, the avoidance of certain errors in distribution of the tasks assigned to the different devices, for example, preventing a management process computer controlling a regulation valve directly and, thus, also enhancing the reliability of the whole automation system.

Operation controllers

We gave an example of operation controllers at the start of this chapter (sugar boiling) and a number of other examples come to mind: cane mills, clarification plant, thermo-compression plant, sugar dryer, etc.

These consist of combinations of basic controllers such as the traditional regulators, and often make use of logic operating sequences. We shall not dwell on this type of system but would note in passing that they constitute sub-assemblies that are highly independent of each other; their links are based particularly on the shop management controllers, which are described later.

When used traditionally, the operation controllers consist of "cabinets" similar to those described for the conduct of sugar boiling. The mathematical formulae governing the exchange of information are always relatively simple; in extreme cases, they can be applied by analogue methods (see Fig. 46.8) or by programmable sequencers for "all-or-nothing" processing (see Fig. 46.9).

Coordinated workshop management

Modules and their function. This level of technical management is thus conceived as a coherent set of modules following a breakdown which often depends on that of the process. The following sequence can thus be defined:

- (a) preparation and extraction
- (b) purification
- (c) evaporation
- (d) first crystallization
- (e) second crystallization

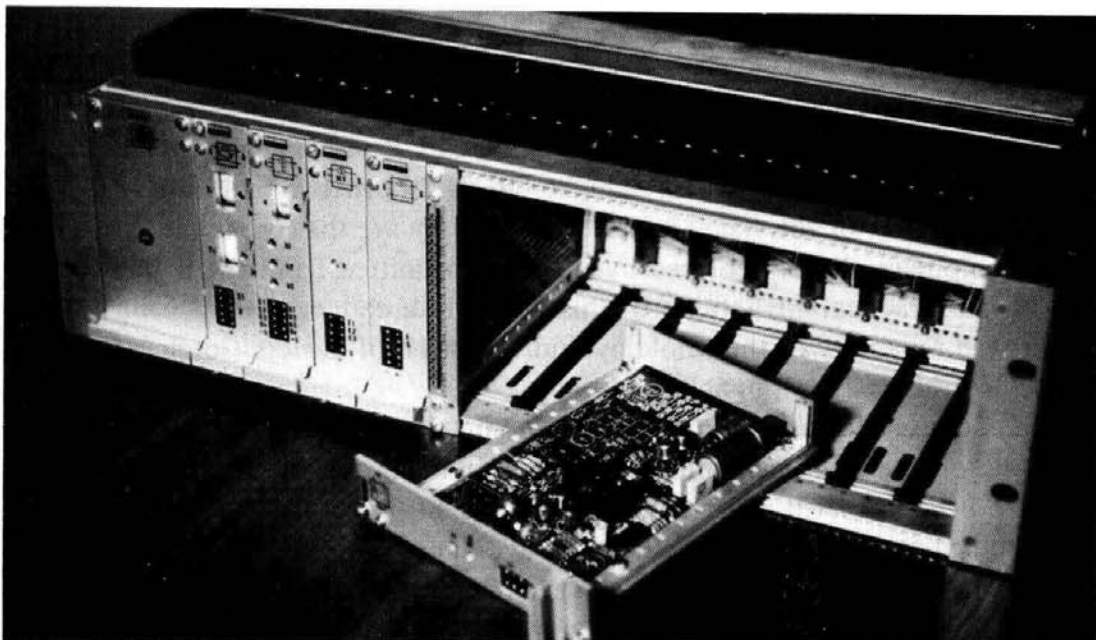


Fig. 46.8. An analogue computation unit (the Modumat made by Sereg) (consisting of basic plug-in modules).

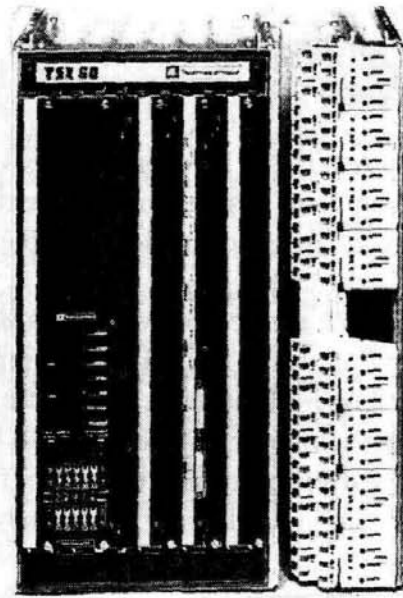


Fig. 46.9. Example of a programmable sequencer.



Fig. 46.10. An industrial computer (the MACSYM made by Analog Devices).

(f) third crystallization

(g) drying and storage.

The functions of these modules relate mainly to management of the flow of products in the plant; one of the chief objectives is to maintain steady conditions throughout the process, which improves other characteristics such as:

(a) capacity of any given unit

(b) quality of the sugar and by-products

(c) economies of energy

(d) life of equipment, etc.

In the various cases encountered, these operating modules also provide the necessary adaptation to the production parameters in order to optimize certain operating criteria, for example, to minimise sugar losses.

Interactions between modules. A relative independence is still the rule for each module: this is so true that one can easily imagine a single module used to manage a limited part of the process. The system is, however, designed to allow complete management of production and provides dialogues for the purpose.

Some of these dialogues are first organized between adjacent modules; they are reduced to the necessary minimum. The intention is to exchange comprehensive information, sufficient to ensure the consistency of operation of the production sub-units.

Other information, mainly of an economic nature, can also be transmitted by the modules to the supervisor at the top of the decision-taking pyramid. They are most often processed when passing in order to devise "high-level orders", namely, overall instructions for the management devices.

Industrial data processing

The algorithms used in this area of shop management devices are complex. The mathematical formulae are sophisticated and, in particular, can be totally changed depending on the conditions of operation.

We are then led to use calculators or industrial computers which are well suited to this type of processing. These data processing machines are easily connected to electronic automation devices by means of specialized input and output couplers ("industrial interfaces"); their connection to pneumatic systems is distinctly more expensive because it requires passage through converters (pressure/current or current/pressure type).

Such computers are most often provided with real time basic software, *i.e.* they are able to follow the chronology of industrial operations and to react to events that require priority processing (such as alarms).

With regard to the connection between several machines when required, one can always find in the range offered by suppliers, connections of "serial" type as described above, which are very suitable (current-loop connections for large distances).

Digital control systems

Without wishing to enlarge on this rather specialised subject, we shall briefly describe how digital systems can be inserted into the CHEOPS structure. Digital control systems have appeared recently, and allow the composition of automation units. Since Honeywell introduced the TDC 2000, all the principal manufacturers have progressively included such a method in their catalogues, for example:

- (a) Foxboro: the FOX 3
- (b) Fisher-Control: the PROVOX
- (c) Fischer & Porter: the DCI 4000
- (d) Sereg-Schlumberger: the MODUMAT 800
- (e) Controle Bailey: the MICRO Z
- (f) Siemens: the TELEPERM.

These systems are distinguished from the traditional machines by the fact that the information processed internally is digitized; they contain a certain number of micro-computers specialized for automation. Figure 46.11 illustrates the structure of such a system. According to the supplier, the organization is more or less decentralized and the function of the various sub-assemblies can therefore be modified. On the other hand, there may be variants of any given component, such as:

- (a) a communication system for the rapid transfer of digital information
- (b) control of process operation on black and white or color screens connected to the network
- (c) possibility of connecting a process computer to this network for optimization and monitoring.

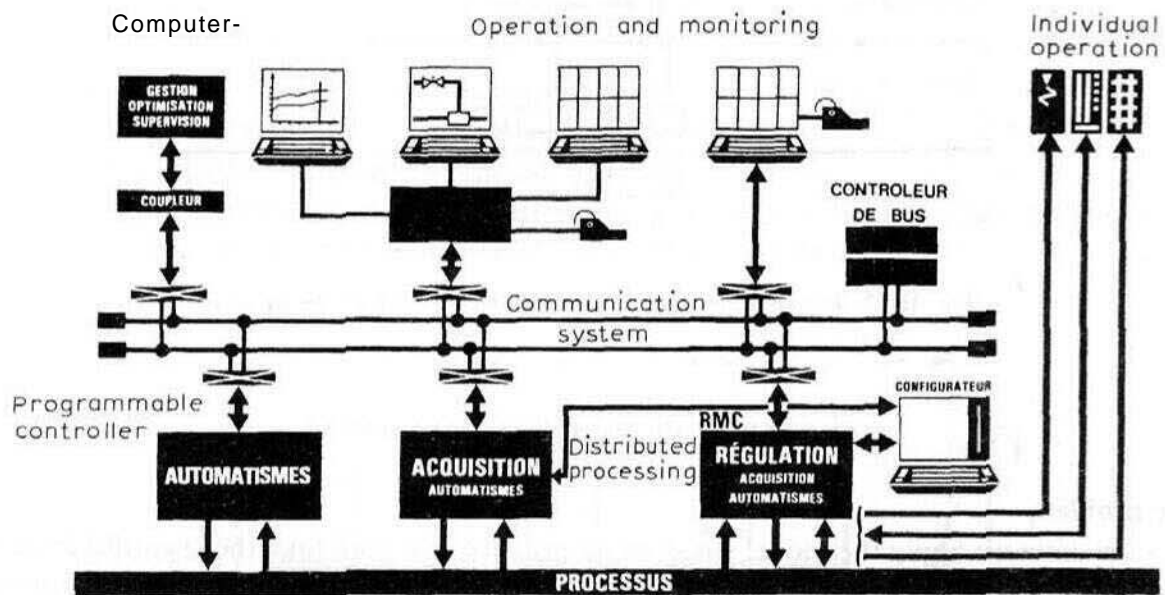


Fig. 46.11. Architecture of a digital control system.

Digital automation in structuring

In the context of applying these digital control systems, we have re-examined the CHEOPS structure and confirm that it could be adapted to digital control. The clarity provided by the structuring is obvious when studying the complete system.

The successive hierarchical levels can be combined in accordance with the diagram shown in Fig. 46.12 which is a functional representation but shows how the hardware components can be assembled in practice:

- (a) at the bottom, a layer connecting to the process by sensors and actuators
- (b) above, an automation layer that combines the functions of basic controllers and of operation controllers; the field of application of the digital control systems
- (c) finally, a management layer with the shop management and monitoring devices. This layer is entrusted to industrial computers.

It is noteworthy that each of the two upper layers is organized around a "network" through which the data pass. We shall call the first the "automation network" (which is ex-

actly that of the digital control system), and the second between the management modules "the optimization network" which can take diverse forms (it involves a true interface between hardware if several small machines are used jointly and is limited to a software link if a single computer combines all the various modules).

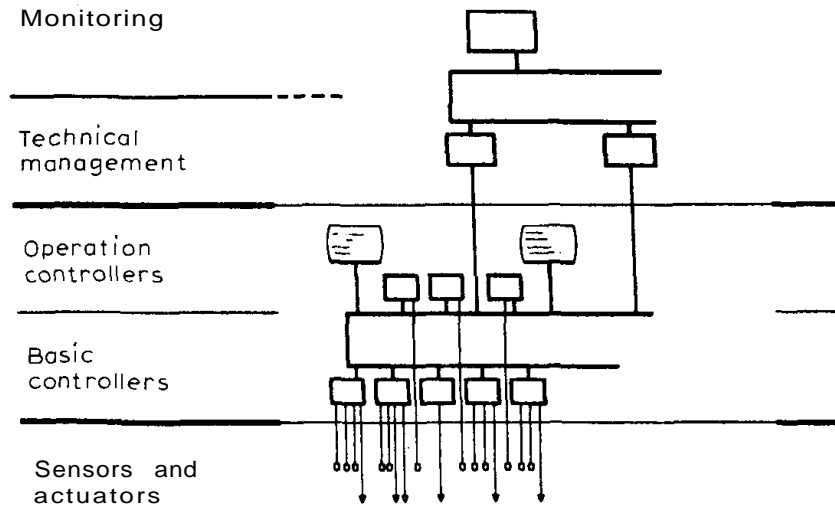


Fig. 46.12. Incorporation of digital systems in the CHEOPS structure.

AN EXAMPLE OF TECHNICAL MANAGEMENT

The problem

To show how we apply the model structure in practice, we shall take the crystallization station as an example. The flow of syrup, massecuite and steam is to be regulated so as to maintain a steady rate and stabilize the demand for steam while being able to deal with major transient events (*e.g.* sudden drops of flow rate of up to 50%). It must also be able to take action to cope with accidents.

In principle, the operator takes part in this program only for overall monitoring and to indicate which particular machine has been taken out of operation or which has been returned to operation.

Methods proposed

Models are often used in data processing; these are systems of equations that relate the various characteristics of the process and which are, in general, difficult to draw up. They are difficult to use in most cases because of their abstract nature.

In order to devise a simple but efficient model of process operation, we have proposed that it should correspond to a simplified diagram (see Fig. 46.13):

The process consists of successive modules in which there are always two fundamental elements:

- (a) stock of product to be processed
- (b) the component that adjusts flow rate in the downstream direction (valve, pump, etc.).

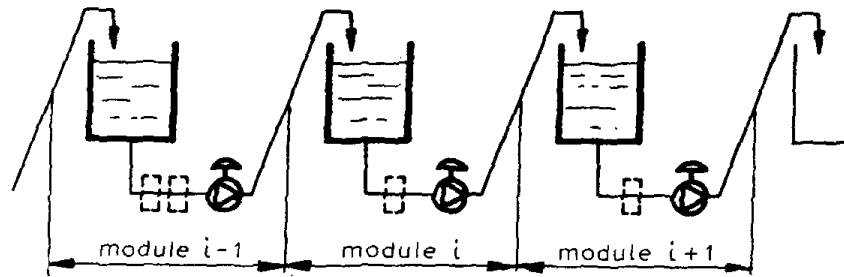


Fig. 46.13. Schematic diagram of division into modules.

In practice, the production machines are, of course, installed on this schematic diagram but they take no part in our present discussion (they are illustrated by dotted lines on the diagram). The physical meaning of the fundamental data of such a model is clear (a stock controlled may be, for example, the total material in the whole first crystallization station, with the downstream flow rate adjustment unit then considered as being the battery of centrifugals).

It is noteworthy that a series of two or more **modules** can be considered as a "super module" or a "global module", where the total stock is equal to the sum of the stocks in the component modules and of which the output component is the output component of the last module (Fig. 46.14). This property of associativity will be used.

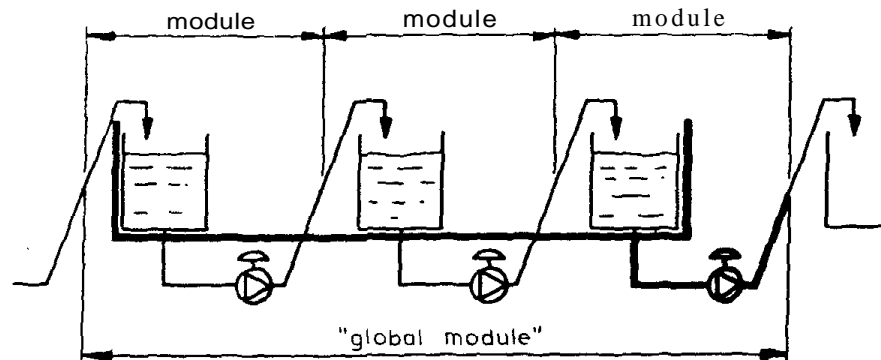


Fig. 46.14. Associativity of breakdown into modules.

Operating principle. We can clearly give now only the main points of the principle. We shall first point out that the flow rate of the plant appears logically to be imposed upstream (we shall modify this proposition below).

The flow leaving a module (or a global module) depends on the stock in the module. Its dependence is generally non-linear because it is plain that the situation becomes critical when stocks are too low or too high.

The meanings of the constants introduced are **also** very simple. Most of them are determined as soon as it is known what stocks are required for each operating mode.

The computer receives all the data required for the computation of stocks. It transmits to the process the set points required to set the flow rate. A basic safety rule prevents any module emptying itself into the next module if the latter is too full. The system, in fact, pro-

poses a reduction of flowrate at any control point when the stock immediately downstream is too high; this adjustment is generally non-linear.

Saturation at any point of the station, therefore, causes a localized slowing down which rises up the production system, progressively filling the available vessels. This allows maximum use to be made of the existing vessels in order to overcome incidents or short-time clogging. It should be noted that, in this case, flow rates can be controlled from downstream.

Adjustment of flow rates through a crystallization plant. Automation of the lower levels (sensors and actuators, regulators, sequential controllers) is provided on all the components of the equipment (pans, centrifugals, etc.).

Table 46.1 illustrates the modules for this section of the plant. It is, for example, evident that the centrifugal battery, which is the output component of a boiling section, is regulated by the stock of the "global module", which is the whole of that section.

The level in the standard liquor tank, which governs the starting time between two strikes is, in fact, a corrected level, allowing for the disturbance caused by the possible discontinuity of the strikes. The data required by the computer to compute this corrected level are the real level in the tank and the state of the progress of the strike.

The various pan strikes are connected through the syrup tanks. Two or three strikes can thus be processed successively and practically by identical programs. In order to regularize the steam demand, two pans are prevented from starting in the same quarter of an hour, for example. This condition is the only one that does not respect the above principles.



Fig. 46.15. Control room of a pan station (the two management computers are on the right).

The program can evidently be modified to allow for a case of continuous boiling. Figure 46.15 is a photograph of the control room of a pan stage. The three massecuite boilings of the plant are managed in accordance with the principles given, either discontinuous boiling or continuous crystallization.

A major consequence. We have stated above that the computer language and the structure selected should allow the operator to remain in control of the data processing system. Here, again, it will be noted that the technical management of the plant decides on the settings to be adopted and can modify them at any time, in full knowledge of the situations, because the parameters used are easy to interpret. It is this transparency which is important to maintain during the whole automation process.

TABLE 46.1

OVERVIEW OF THE CRYSTALLIZATION CONTROLS PROVIDED BY THE STATION MANAGEMENT SYSTEM

<i>Controls</i>	<i>Purpose</i>	<i>Constraints to be respected</i>
<i>Continual adjustment</i>		
(1) Centrifugal sequences	Adjustment of overall stock (liquor tanks + contents of pans and crystallisers)	Number of centrifugals available with limitation and alarm, if needed
(2) Flow rates of rotary pumps	Regulation of centrifugal distribution level	Distribution between crystallisers in relation to their stock and the quantities processed
(3) Time between two boiling starts	Stability of liquor stock	Priority indicated by the operator (with relevant management) and alert conditions (high and low)
<i>Safety</i>		
(4) Addition of water when needed before thickening	Full safety in level at crystallisers	Relating to heavying up and discharging strikes, slowing down in certain cases the strikes involved with the same set of crystallisers
(5) Steam slowing down (in anticipation)	Limitation of addition of water to strict minimum required for safety	Stage of progress of boilings concerned in order not to disturb them during the critical phases
<i>Liaison</i>		
(6) Set points adjusted towards upstream (stock and flowrate)	Consistent operation of all sections	Priority relating to the algorithms used, global stock or partial stock

Control and computerization

Whether the process is automated or not, it is important to supervise the production characteristics by means of regular control. Specific analyses of each part of the process are

therefore carried out by the control laboratory; on-line measurements are also made, for example, to monitor the thermal production parameters.

This process of production monitoring is still in a state of development but remains "routine", that is, it is organized around measurements most of which are intermittent. The measurements must, of course, be consulted by the plant management in order to allow updating of the various production parameters.

It must be acknowledged that, until now, communication at this level has often been small in volume and rudimentary in method. It is now plain that modern data processing facilities provide highly effective methods of transmitting data within the plant.

There are a number of groups of plants in the sugar industry that exchange data. The major companies of course centralize data to their head office or central laboratory and associations exist at different levels between plants. One may imagine that the local structures organized could be later connected together and/or with central offices depending on the organization of the group concerned.

Computerized data management

Any solution to problems posed by computerization of production control must, in our opinion, be:

(a) *adaptive* and, by corollary, *evolutive*, so that it can easily be transmitted to any sugar plant and must, even more so, be able to evolve efficiently in the plant where it is installed

(b) *standardized*, for increased capability of "software maintenance". It is therefore necessary to remain strictly within the framework of standard available software and, more generally, select work methods that are sufficiently well known

(c) *easy to use*, for the users have different levels of ability. In order to considerably facilitate introduction to the equipment, a number of practical methods are used such as "menus", "formulae", etc., allowing for the fact that experienced operators quickly become familiar with new procedures.

It is seen, when proceeding further into the study, that the major difficulty arises from the diversity of cases processed. This diversity occurs at different levels:

(a) *the location of the plants* and the relevant requirements of hardware and connections

(b) *the work habits* of the workers in the plant and the traditional facilities that they possess

(c) more fundamentally, differences of *layout and of hardware installed* in the plants.

For our part, we have always abandoned the traditional approach to the subject with a common basis and adaptation to each plant. We prefer the approach that consists of supplying the user with the systems required to organize his own specific application, what we call a "configurable system".

Example. Following these criteria, we have proposed* a universal system that can be briefly described. It operates with a mini-computer with a central memory of 256 or 512 K octets and a "hard-disk" storage system of capacity at least 10 mega octets (see Fig. 46.16). A

* G. Windal, Sucrerie Française, April 1983.

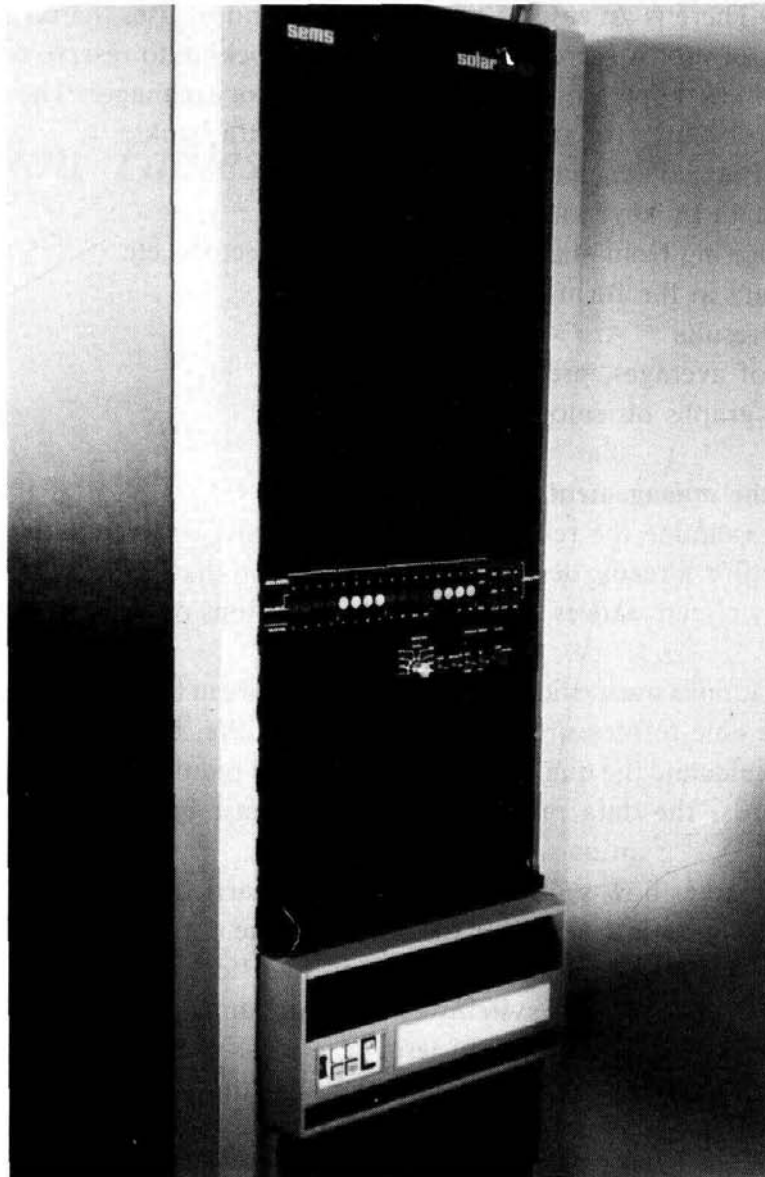


Fig. 46.16. A mini-computer designed to manage technical data (CLEOPATRE installed on SOLAR 16).

peripheral is added whose size depends on the plant. There are generally 2–5 VDUs and 1 or 2 printers.

The software uses BASIC, a universal language that allows transposition to a mini-computer or to a group of micro-computers when needed; this choice is moreover compatible with that adopted for the process operation described above. It is nevertheless noteworthy that a process computer makes use of "real time BASIC" whereas in this case the system makes use of "shared time" which is accessible to several users simultaneously and gives each the impression of using the machine alone. A tool designed for managing dialogue on the screen has moreover been used, the presentation of formulae, tests of responses, and correction of typing errors are organized easily.

The software is organized for application of structured programming methods to facilitate

later maintenance. There is an ease of commissioning and of use, thanks to the interactive feature at all levels of use. A secret code provides interlocking to reserve certain procedures to one or more privileged operators, typically the laboratory manager. The system also comprises safeguard procedures for the software and the data banks.

The following principal functions are available:

- (a) entering of data by keyboard
- (b) "on-line" entering from analyzers, measuring collectors, etc.
- (c) display of data in the form of tables
- (d) print-out of results
- (e) drawing up of averages, preparation of balances
- (f) comparative graphs on colour screen.

Incorporation of the management system

It is of interest to examine the relationship that may exist between such a system and the management structure already described. It is apparent that the two approaches have a tendency under any circumstances to improve the conditions of operation of the production process.

It is then seen that links must evidently be provided between the two devices; it is necessary, for example, to be able to measure the stock of masecuite, both to manage flow through the section and to calculate the quantities "pending" and required for the balance of the factory. In other words, the data management system must be connected to the operating system.

Figure 46.17 illustrates how such integration can be carried out. The "layer" adopted is that of optimal management because of the nature of the data to be exchanged. A check is made that the hardware used on either side are in fact "calculators".

It is seen, however, that the two systems remain fundamentally different. The operating system is destined for short and medium-term reactions, while the control system, in particular, supplies data allowing longer-term updating. It is therefore preferable to entrust these different tasks to different computers.

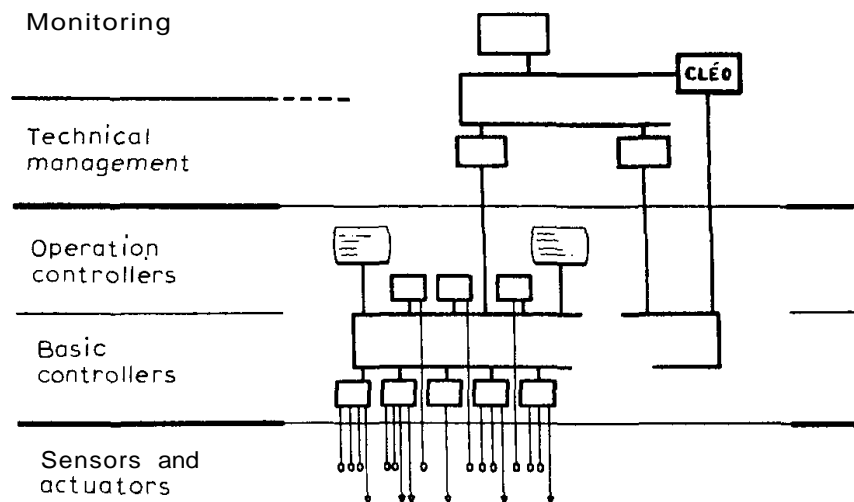


Fig. 46.17. "CHEOPS - CLEOPATRE" integration.

Conversion Tables — Metric to British Units

Length

1 cm	0.3937	in.
1 metre	3.28084	ft.
1 metre	1.09361	yard
1 km	0.62137	mile

Area

1 cm ²	0.155	sq.in.
1 m ²	10.764	sq.ft.
1 m ²	1.196	sq.yards
1 ha	2.471	acres
1 km ²	0.3861	sq.mile

Volume

1 cm ³	0.061024	cu.in.
1 dm ³	0.0353147	cu.ft.
1 m ³	35.3147	cu.ft.
1 hl	3.53147	cu.ft.
1 litre	0.2200	Imp. gal.
1 litre	0.264166	U.S. gal.

Weight

1 g	0.035274	oz. (avoirdupois)
1 kg	2.204622	lb.
1 quintal = 100 kg	2.204622	cwt. (American)
1 quintal = 100 kg	1.9684	cwt. (English)
1 tonne = 1,000 kg	1.1023	short ton
1 tonne = 1,000 kg	0.9842	long ton

Force

1 Newton	0.102	kg
1 daN (deca Newton)	1.02	kg

Pressure

1 bar = 1.02 kg/cm ² = 10 ⁵ Pascal	14.500	p.s.i.
1 kg/cm ²	14.2233	lb./sq.in. (p.s.i.)
1 kg/m ²	0.204816	lb./sq.ft.

1 t/dm ²	10.2408	short ton/sq.ft.
1 t/dm ²	9.1436	long ton/sq.ft.
1 cm of mercury at 25°C =	13.56	cm of water at 20°C
	13.70	cm of water at 50°C

Fibre loading

1 kg/m ²	0.204816	lb./sq.ft.
1 kg/m ² /m	0.062428	lb./sq.ft./ft.

Conversion Tables - British to Metric Units

Length

1 inch	2.54 cm
1 foot	0.3048 m
1 yard = 3 ft.	0.9144 m
1 mile = 1,760 yards = 5,280 ft.	1,609.344 m

Area

1 sq.in.	6.4516 cm ²
1 sq.ft. = 144 sq.in.	0.092903 m ²
1 sq. yard = 9 sq.ft.	0.8361 m ²
1 acre = 4,840 sq. yards	0.4047 ha
1 sq. mile = 640 acres	2.59 km ²

Volume

1 cu.in.	16.387 cm ³
1 cu.ft. = 1,728 cu.in.	28.317 dm ³
1 cu.ft.	0.028317 m ³
1 Imp. gallon	4.546 l
1 U.S. gallon	3.7854 l

Weight

1 oz. (avoirdupois)	28.350 g
1 lb. = 16 oz.	453.5924 g
1 cwt. (American) = 100 lb.	45.36 kg
1 cwt. (English) = 112 lb.	50.80 kg
1 short ton (American ton) = 2,000 lb.	907.185 kg
1 long ton (English ton) = 2,240 lb.	1,016.047 kg

Pressure

1 lb./sq.in. (p.s.i.)	0.070307 kg/cm ²
1 lb./sq.ft.	4.88243 kg/m ²
1 short ton/sq.ft.	0.09765 t/dm ²
1 long ton/sq.ft.	0.10937 t/dm ²
1 in. mercury at 80°F =	{ 1.130 ft. water at 70°F
	{ 1.143 ft. water at 130°F
1 lb./sq.in. =	2.316 ft. water at 80°F

Fibre loading

1 lb./sq.ft.	4.88243	kg/m ²
1 lb./sq.ft./ft.	16.0185	kg/m ² /m

Density

1 kg/m ³ = 1 g/dm ³	0.062428	lb./cu.ft.
1 kg/dm ³ = 1 t/m ³	62.428	lb./cu.ft.

Heat

1 kcal	3.9683	B.T.U.
1 thermic = 1,000 kcal	3968.3	B.T.U.
1 kJ = 0.2389 kcal	0.000968	B.T.U.

Calorific value

1 kcal/kg	1.8	B.T.U./lb.
1 kJ/kg	0.43	B.T.U./lb.

Heat transmission

1 kcal/mVh	0.368669	B.T.U./sq.ft./h
1 kcal/m ² /h/°C	0.204816	B.T.U./sq.ft./h/°F
1 Watt/m ² /°C = 0.86 kcal/m ² /°C h	0.116	B.T.U./sq.ft./°F h
1 kW/m ² /°C = 860 kcal/m ² /°C h	176	B.T.U./sq.ft./°F h

Heat conduction through a plate

1 kcal/m ² /h/°C/m	0.672	B.T.U./sq.ft./h/°F/ft.
1 kcal/m ² /h/°C/m	8.0636	B.T.U./sq.ft./h/°F/in.

Combustion chambers

1 kcal/mVh	0.11237	B.T.U./cu.ft./h
------------	---------	-----------------

Temperature

C	5/9 (F - 32)
1°C (difference in temperature)	1.8°F

Work and power

1 J	0.7375	ft.lb.
1 kgm	7.233	ft.lb.
1 kgm/sec	0.01315	h.p.
1 ch	0.98632	h.p.
1 kW	1.341	h.p.

Evaporation rate

1 kg/m ²	0.204816	lb./sq.ft.
1 kg/m ² (from 0 to 100°C)	0.2428	Ib./sq.ft. (from and at 212°F)
1 kg/m ² /°C	0.113786	lb./sq.ft./°F

Capacity, area and volume .

1 m ² /t	9.765	sq.ft./short ton
1 mVt	10.9366	sq.ft./long ton
1 l/t	0.032036	cu.ft./short ton
1 l/t	0.03588	cu.ft./long ton
1 m ² /m ³	0.3048	sq.ft./cu.ft.
1 m ² /hl	3.048	sq.ft./cu.ft.
1 mVt	35.8814	cu.ft./long ton
1 m ³ /m ²	3.28084	cu.ft./sq.ft.

Moment of inertia

1 m ² kg wt.	7.233	lb.wt.-sq.ft.
1 m ² kg	23.73	lb.-sq.ft.

Heat

1 B.T.U.	0.252	kcal
----------	-------	------

Calorific value

1 B.T.U./lb.	0.555556	kcal/kg
--------------	----------	---------

Heat transmission

1 B.T.U./sq.ft./h	2.71246	kcal/m ² /h
1 B.T.U./sq.ft./h/°F	4.88243	kcal/m ² /h/°C

Heat conduction

1 B.T.U./sq.ft./h/°F/ft.	1.488	kcal/m ² /h/°C/m
1 B.T.U./sq.ft./h/°F/in.	0.1240	kcal/m ² /h/°C/m

Combustion chambers

1 B.T.U./cu.ft./h	8.90	kcal/mVh
-------------------	------	----------

Temperature

F	1.8	C + 32
1°F (temperature difference)	0.555556	°C

Work and power

1 ft.lb.	0.138255	kgm
1 h.p. = 550 ft.lb./sec	76.04	kgm/sec
1 h.p.	1.01387	ch.
1 h.p.	745.70	W

Evaporation rate

1 lb./sq.ft.	4.88243	kg/m ²
1 lb./sq.ft./h (from and at 212°F)	4.118	kg/m ² /h (from 0 to 100°C)

Capacity, area and volume

1 sq.ft./short ton	0.102408	m ² /t
1 sq.ft./long ton	0.091436	m ² /t
1 cu.ft./short ton	31.215	l/t
1 cu.ft./long ton	27.870	l/t
1 sq.ft./cu.ft.	3.28084	m ² /m ³
1 sq.ft./cu.ft.	0.328084	m ² /hl

Moment of inertia

1 lb.wt.-sq.ft.	0.138255	m ² /kg wt.
1 lb.-sq.ft.	0.04214	m ² /kg

Equivalents of Foreign, British and Metric Units

Brazil	Area	= 1 alqueire	= 5.98	acres	= 2.42	ha
Cuba	Area	= 1 cordel	= 1/324	caball.	= 0.4142	ha
	Area	= 1 caballeria	= 33.16	acres	= 13.42	ha
	Weight	= 1 lb. (Spanish)	= 1.014	lb.	= 0.460	kg
	Weight	= 1 arroba	= 25	lb. (Spanish)	= 11.502	kg
	Weight	= 1 long t (Spanish)	= 2,240	lb. (Spanish)	= 1,030.400	kg
Egypt	Weight	= 1 sack of sugar	= 325	lb. (Spanish)	= 149.500	kg
	Length	= 1 cassaba	= 11.647	ft.	= 3.55	m
	Area	= 1 kirat	= 1/24	acre	= 0.017	ha
	Area	= 1 feddan	= 1.038	acre	= 0.42	ha
Formosa	Weight	= 1 cantar	= 99.04	lb.	= 44.928	kg
	Area	= 1 koh	= 2.40	acres	= 0.97	ha
India	Weight	= 1 picul	= 133.3	lb.	= 60.460	kg
	Area	= 1 bigha	= 1/3	acre	= 0.135	ha
	Weight	= 1 seer	= 1/40	maund	= 0.933	kg
Japan	Weight	= 1 maund (bazar)	= 82.2857	lb	= 37.325	kg
	Weight	= 1 kin	= 0.01	picul	= 0.609	kg
Java	Volume	= 1 koku	= 6.37	cu. ft.	= 180.4	litres
	Area	= 1 bouw	= 1.77	acre	= 0.7163	ha
	Weight	= 1 picul	= 136.2	lb.	= 61.780	kg
Mauritius	Volume	= 1 blik	= 5	Imp. gal.	= 22.7-	litres
	Area	= 1 arpent	= 1.043	acres	= 0.422	ha
Peru	Area	= 1 fanegada	= 7.156	acres	= 2.896	ha
	Area	= 1 cuadra	= 1.730	acres	= 0.700	ha
Phillippines	Weight	= 1 picul	= 139.44	lb.	= 63.250	kg
	Volume	= 1 cavan	= 75	quarts	= 85.2	litres

Puerto Rico Area = 1 cuerda = 0.97 acre = 0.3930 ha

South Africa Area = 1 morgen = 2.1165 acres = 0.8565 ha

N.B. The short ton (2,000 lb.) is used in the American countries: Louisiana, Puerto Rico, Hawaii, also in Cuba, Canada and South Africa (the Philippines use the metric ton).

Most countries previously using the long ton (2,240 lb.) have now adopted the metric system.

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