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Technical Report 1982

by Niek Van de Ven

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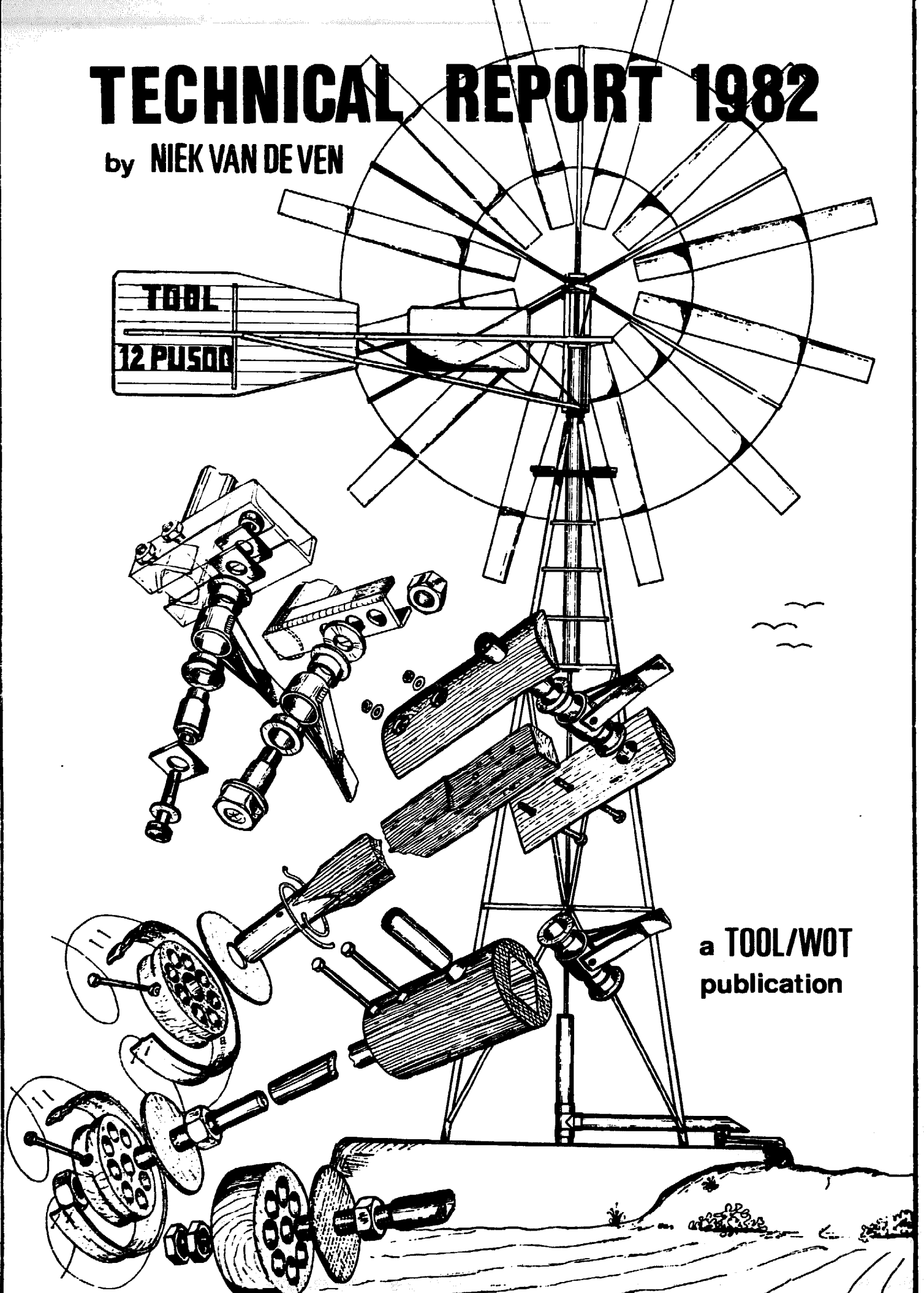THE NETHERLANDS

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TECHNICAL REPORT 1982

by NIEK VAN DE VEN

TOOL

12 PU500

a TOOL/WOT
publication

TECHNICAL REPORT

by NIEK VAN DE VEN

dealing with the

TOOL WINDMILLPROJECTS

over the period

AUGUST 1977 _ MARCH 1981

Acknowledgement

<u>TOOL</u> is a Dutch foundation participating in the global process of renewable development and application of socially appropriate technologies.

The broad objective is to promote greater freedom for groups which are deprived of full opportunities for local, self-programmed and self-sustained development.

The strategy is to provide and support information resource links among the practitioners, users and generators of appropriate technology for development.

The operations of TOOL are designed to match need and resource:
- technical advice and support
- research and development
- publications
- documentation systems and services
- applicstion projects
- education and training
- organisational linkages.

<u>WOT</u> is a non-profit organisation at the Twente University of Technology and gives technical advice in the field of wind energy, solar energy and water supply. The broad objective is to improve the position of the weaker sections in society and the advice should be appropriate to the local situations and circumstances. To support the technical advice the WOT has a testing field where various designs are being developed and tested. The WOT is mainly a voluntary organisation and consists mainly of students of the university. Several staffmembers take care for the administrative and technical support.

CONTENTS page

Short History

In 1975 the Organization of the Rural POOR (ORP) in Ghazipur
(India, U.P.) wanted to support small and marginal farmers
in growing an extra crop during summer season by means of
irrigation. As rather high wind speeds occur during summers
the idea was born to use wind energy as the power source
for pumping the irrigation water from the wells.

In cooperation with the TOOl-foundation in the Netherlands
some feasibility studies have been carried out and eventu-
ally a proposal for a pilot windmill project has been gran-
ted both by the Government of India and the Netherlands.
The latter sponsored the pilot project which started in
August of 1977.

The project was extended year by year while the 12 PU 500
Windmill design improved step by step. Expansion of the
project occurred in 1979 when the newly built ORP-TOOL-
Windmill workshop was inaugurated and the Windmill Research
and Developments activities were taken up at the Allahabad
Polytechnic Allahabad (APA). In the course of 1980 another
Windmill project started in Trichy (South India) where the
Workshop for Rehabilitation and Training of Handicapped
(WORTH) also adopted the same windmills for irrigation as
one of its products.(The latter has been financed by HIVOS.)

In January of 1981 the Allahabad Polytechnic Allahabad
started the "National Windmill Demonstration Project" which
was initiated and sponsored by the Department of Science and
Technology (DST) of the Government of India. During the first
three months thirty 12 PU 500 windmills have been succesfully
distributed all over India attended with trainingcourses for
mechanics and engineers, after which the TOOL-project ran to
its end.

The following year three hundred 12 PU 500 windmills have been

built by the Allahabad Polytechnic by order of the Department of Science and Technology while plans exist regarding a windmill production of several thousands during the next year.

1. The design criteria and first developments.

Previous to the introduction of the new waterlifting device sufficient knowledge concerning its feasibility and acceptance should be gathered in order to obtain relevant input for a proper set up of a pilot project.
Therefore short term missions were undertaken to Ghazipur to gather information about materials, equipments and expertise. Also information on winddata of the eastern part of Uttar Pradesh gave important information in order to calculate the size of the windmill which had to be designed.

Since the windspeeds in the Ghazipur district are rather low, rough calculations pointed out that even in the windy season a windmill applied for irrigation of a command area of two and a half acres (1 ha) needs a rotor diameter of at least 7 metres. This calculation was based on the assumption that the average water requirement of the irrigated plots amounts to $\pm$ 7 mm a day, the total pumping head increases to 14 metres (dry season) and that the windmill starts pumping with a total efficiency of 0.10 (if the windspeed exceeds 2.5 m/s.).

Other calculations also led to smaller windwheel diametres, very high starting windspeeds and extremely huge storage tanks on the one hand and a large rotor diameter, moderate tank-capacities and low starting windspeeds on the other.
An optimum must exist depending on material- and labourcosts and the loss of agricultural area in order to realize the needed storage tank. All these matters were taken into account to approach this unknown ideal of windmill size and storage capacity, but still a lot of uncertain aspects and doubts remained.

Moreover in 1977 the availability of expertise and successful designs of cheap, reliable and appropriate windmills were

rather limited in spite of the fact that many efforts had
been undertaken already to develop windmills in other
developing countries.
Such experiences always showed that the introduction of
the windmill is easily underrated. First the technology
has to be developed and implemented successfully under
the local circumstances after which a process of transfer
of technology should take place.

Due to the above mentioned lack of sufficient experience,
it was decided to design and develop first a proto-type
windmill in the Netherlands which should be completely
constructed with the materials, expertise and implements
which are locally available in Ghazipur. The total material-
costs should amount to Dfl. 1000.-- (Rs 3000.-- up to Rs 4000.--
1977).
Materials like gas-pipe, angle-irons, flat- and sheet-irons,
ball-bearings, bolts & nuts etc. would be available in plenty
there according to the information obtained.
The local blacksmiths in India are able to construct simple
implements from these, like wheat-threshers and many other
agricultural equipment.Moreover the electrification of seve-
ral villages sparks off more or less sufficiently equipped black-
smiths and small industries. So welding and simple machining,
like drilling- and lathe-work, are rather current.

In comparison with steel (1977: angle iron RS 2.20/Kg) wood
is very costly and later on it proved not to be available in
required size and quality. For example wood for the construc-
tion material for building a tower-structure would not be at-
tractive as far as the lifetime is concerned. Moreover the
Indian Government stimulates application of other construc-
tion-material rather than wood in order to prevent the severe
erosion of the country due to deforestation.

Such considerations and restrictions led to new ideas and
alternative designs of some essential parts of the windmill.
In September 1977 the construction of the first proto-type
windmill was carried out at the testing field of the Working

Group on Developing Techniques (WOT) which is a member of
the TOOL-foundation.

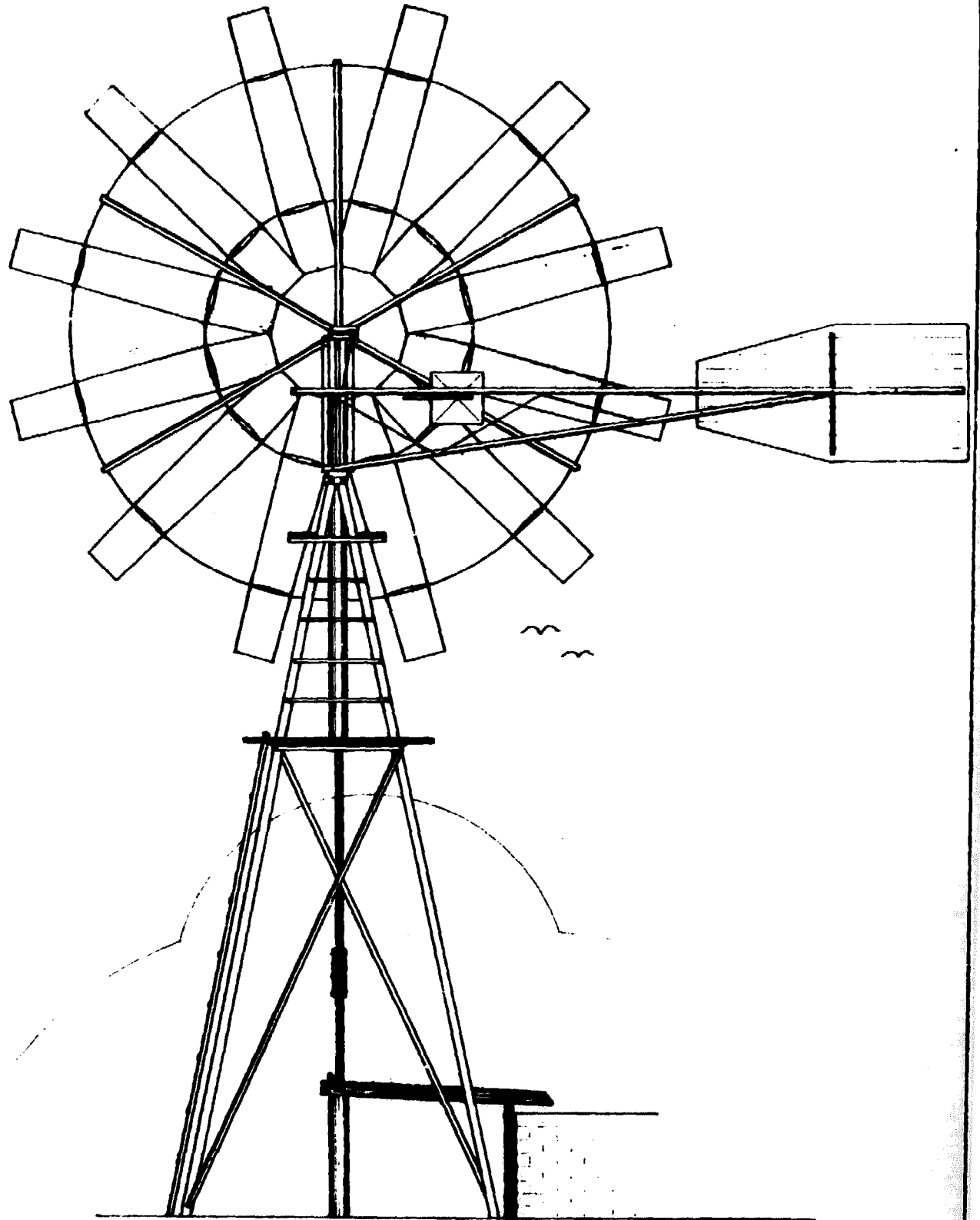## 1.1 The construction of the first proto-type

This first proto-type windmill was fitted with a 4m dia-
meter 16 bladed rotor which was left after an education
program of the WOT in which it was constructed. It seemed
to be a fruitful opportunity to integrate it in the design
which had to be developed at that time, but afterwards it
turned out that this rotor was too weak, too complicated
and too expensive. Moreover according to the estimation a
4 m. diameter-windmill is rather small to serve the purpose.
So after some trouble-shooting during the months of September-
November, the experience with the very first proto-type
caused a complete re-designing. This resulted in a set of
drawings of the tower construction, head construction, moving
parts and the windwheel of a new 5 m. dia. 12-bladed proto-
type. Besides also a small building manual of the new proto-
type has been edited,but actually the total set was not yet
completed. Still the design and the drawings of the most im-
portant and essential part of this wind-powered water-lifting
device, the pump, failed. In the next chapters extra atten-
tion will be paid to the pump designs which followed.

Basic designs

The expected rotor efficiency of the newly designed windmill
amounts 0.38 theoretically if the windmill is loaded at a tip-
speed ratic ($\lambda = \frac{\Omega R}{V}$: tipspeed divided by the windvelocity) of
two. Via a simple crank.-connecting rod-crosshead mechanism the
rotation of the millshaft is converted into an up- and downmovement
of the pumprod which drives the single-acting reciprocating piston-
pump.
The tail of the headconstruction carries a windvane which turns
the headconstruction around the towerpipe in order to keep the
windwheel perpendicular to the wind.
In case cf severe storm a simple security-mechanism will dislock
automat ically the headconstruction from the tailstructure after

which the head (and so the windwheel) turns 90^0 degrees out of
the wind. In drawing N̲o̲ 1 the windmill has been drawn in that
secured position.

The rotor-, head- and towerconstruction have been composed
of parts of angle-iron of 1.5" (except the four towerlegs: 2")
and flat-iron, 1.25" which are jointed by means of electric
weldings. The application of bolt and nut connections is limited
just to the mounting of sheet-iron parts and some connections
like hinges, pumprod-connection etc.

The millshaft (50 mm.) rotates in dustproof self-adjusting
bearings which are considerably oversized according to the
S.K.F.-specifications. The crankpin- and crosspin-bearings are
preferably furnished with nylon bearing-bushes, but bronze bearings
can also serve the purpose if lubrication is applied from time to
time. Also the crosshead which is composed of two specially shaped
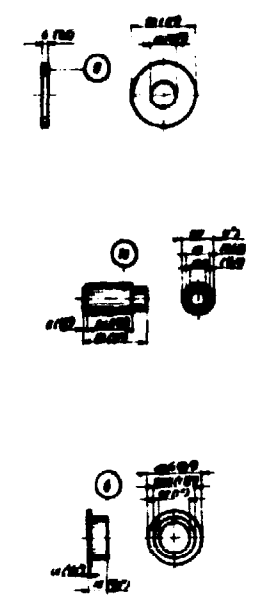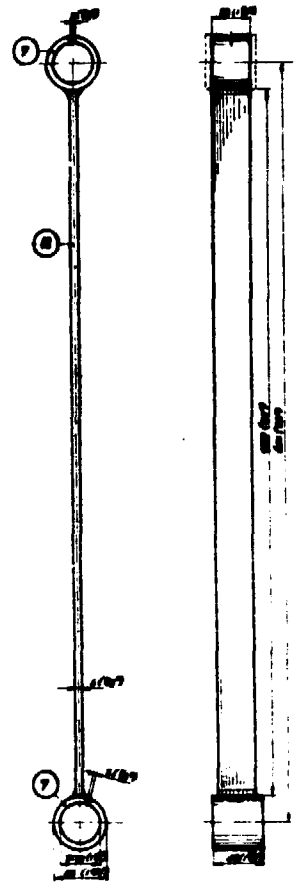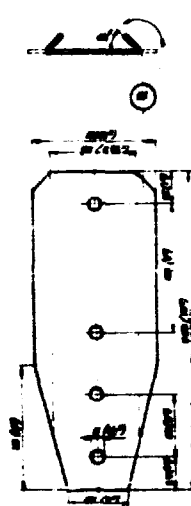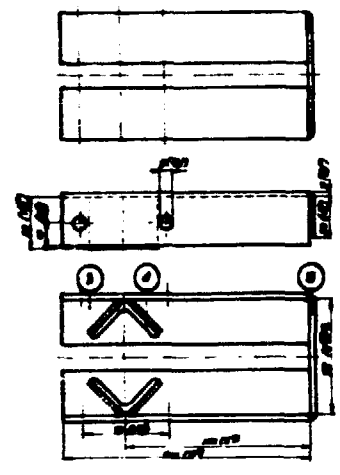sheets needs the lubrication (see next chapters).

Characteristic of the headconstruction is the surrounding of the
towerpipe by its framework of angle-iron which acts as a bearing.
In radial sense some tolerance is permitted , and is even necessary
due to the unroundness of the towerpipe.

Positive experience has been achieved already by applying this
remarkable joint of the head and tower-construction in a
wooden sail-wing windmill which has been designed and constructed
in May 1976. Axial forces can be carried-over directly to the
top-edge of the towerpipe, so the radius of coulomb's friction
acts very near to the centre.

The total structure has been designed for maximum windspeeds of
at least 20 m/sec and even the blade tips (free length of appr.
60 cm.) can in theory stand the forces which should act on it
if the windwheel runs free of load under static conditions at
a windspeed of 65 m/sec.

It is expected that the rotor is not the constraint factor of
strength of the total structure. However dynamic forces (e.g.
due to rotor unbalance) probably cause vibration in the natural
frequency of several parts which may even lead to fatigue cracks!
For example the natural frequency of the headconstruction situated
around the towerpipe comes to 4.35 rad/sec (theoretically)
which means that during operation of the windmill at a R.P.M.

number of approx. 45, resonance may occur. Also the slender
parts like the long angle-irons of the tower- and tail-con-
struction are free to vibrate within a scale of R.P.M.-numbers
of the windmill if unbalances exist in the rotor construction.
The single-acting piston pump applied in combination with the
first prototype windmill consisted of a 6" p.v.c. pipe in which
a wooden footvalve and piston with leather washers and sealing
were acting. This kind of pump, which also was applied in
combination with the former mentioned wooden sailwing windmill,
seemed to be very reliable as far as resistance against wearing
and tearing of the sealings and washers were concerned.

1.2. The first experiments in the Netherlands.

Due to shortage of time the prototype windmill was not yet kept up
to date according to the new designs, so the first experiments
have been undertaken with a rotordesign which was not represen-
tative of the prototype which would be introduced in India.
Moreover the elevation head in India was expected to be considerably
more than the 4.0 meter head which appears in the Netherlands.
In spite of that the first experience turned out already that
a proper fixing of the pump in the well as well as jointing the
wooden pumprods are rather difficult, especially if high R.P.M.-
numbers occurred. The entire pump-construction jumped in its
suspension and the pump-rod joints got loose caused by the
reciprocating forces after some time during operation. The cross-
head, constructed of two shaped sheets, started wearing out and
making noise. Neither did the connection of the crank to the mill
shaft turn out to be reliable. Although the security mechanism
functioned rather well, its critical adjustment did not remain
and demanded re-tightening of the adjusting bolt from time to
time.
Especially its required and accurate hinges cause that this
sensitive mechanism may never be successful, so other possi-
bilities had to be considered and developed in order to obtain
a simple and reliable security device.

A rather remarkable aspect has been discovered in those days.
Depending on the R.P.M. number the amount of water, delivered

per stroke of the piston pump exceeded the stroke-volume in a
percentage of more than 30 %. Later on this effect was recognised
as an inertia-effect: the kinetic energy of the accelerated water
column is converted into an extra delivery. Even after introduc-
tion of a small but permitted leakage in the piston sealing,
which is done in order to decrease the starting torque of the
pistonpump, this inertia effect still persisted. Also some experi-
ments with sealingless pistons has been carried out but
unfortunately the required time for a proper investigation
failed and the TOOL- windmill experts travelled to India at the
end of November 1977.

2.1. The first experiences with the 5 m. dia. windmills.

As mentioned above the experiences with the first prototype
windmill in the Netherlands were rather poor and still a new
and unknown security-mechanism and a complete new and untested
rotor was to be applied. In spite of that the realization of
the first windmill at the Technical School in Ghazipur progressed
smoothly. Also a quick prefabrication of the next two prototypes
in the R.T.I. workshop, later on, has been achieved by using
welding- drilling- and shaping-moulds. Small changes in the
original designs were caused by slight differences in quality
and size of the materials which are available locally.

Transport of the entire windmills to the site proved to be
rather uncomplicated. A truck or tractor with lorry can easily
contain all the separate parts (tower,head, windwheel) of a
complete windmill. Local people are always prepared to carry the
unloaded parts, lying on the roadside, to the site where the
windmill is going to be erected.
In general the windmill is built above an existing well, from
which water was drawn previously for irrigation purposes by
using bullocks as an energy-source. Since the maximum pumping
capacity of the windmill amounts to appr. 30 m^3/h the water-
supply to these wells is often insufficient.
In order to increase this capacity a tubewell has to be applied
at the bottom of the well. The length of these tubewells
varies between 12 to 15 meters depending on the depth of the
aquifers.
The erection of the tower is achieved simply by pulling and pushing
it upwards by means of offered manpower of the local people.
After the tower has been positioned(exactly in the centre above
the tubewell)the towerlegs are poured in with concrete in the
40 cm deep holes which have been made in advance.
After a five-days period the concrete is sufficiently hardened
out to stand the extra load acting on it during lifting and
positioning of the headconstruction and windwheel.

2. Technical developments of the windmills

In the beginning of December 1977 the windmill-project started
in Ghazipur. Although a workshop was not at the disposal of the
counterpart organization: The Organization of the Rural Poor
(O.R.P.), the facilities seemed to be sufficient in Ghazipur-
village.
Several blacksmiths and institutes proved to be interested in
the windmill-technology (there was no windmill-history in the
rather remote Ghazipur-area) and it was decided to construct the
first windmill in cooperation with the teachers and students of
the local Technical School.
On Friday the 20th of December, the first windmill (KK I) was
erected near the Kusumih Kalan village beside the national-
highway Ghazipur-Benares and it was baptised by local villagers
and authorities. During the month of January two other windmills
have been prefabricated in the workshop of the Rural Trainings
Institute (R.T.I.) where agricultural implements like plows
and threshers are manufactured by trainees and staff members.

In the course of 1979 a newly built workshop, situated at
the ORP-campus, came at the disposal of the windmill project.
A complete set up of windmill production, maintenance and trai-
ning facilities has taken place in this workshop. At the same
time all research- and development activities were taken up
at the Allahabad Polytechnic Allahabad (APA) where already a
modest windmill production was sparking off as well. During
the period of November 1979 up to March 1981 an intensive pro-
gram of pumpdesign, pumptesting, redesigning of windmillparts
and training was carried out over there.
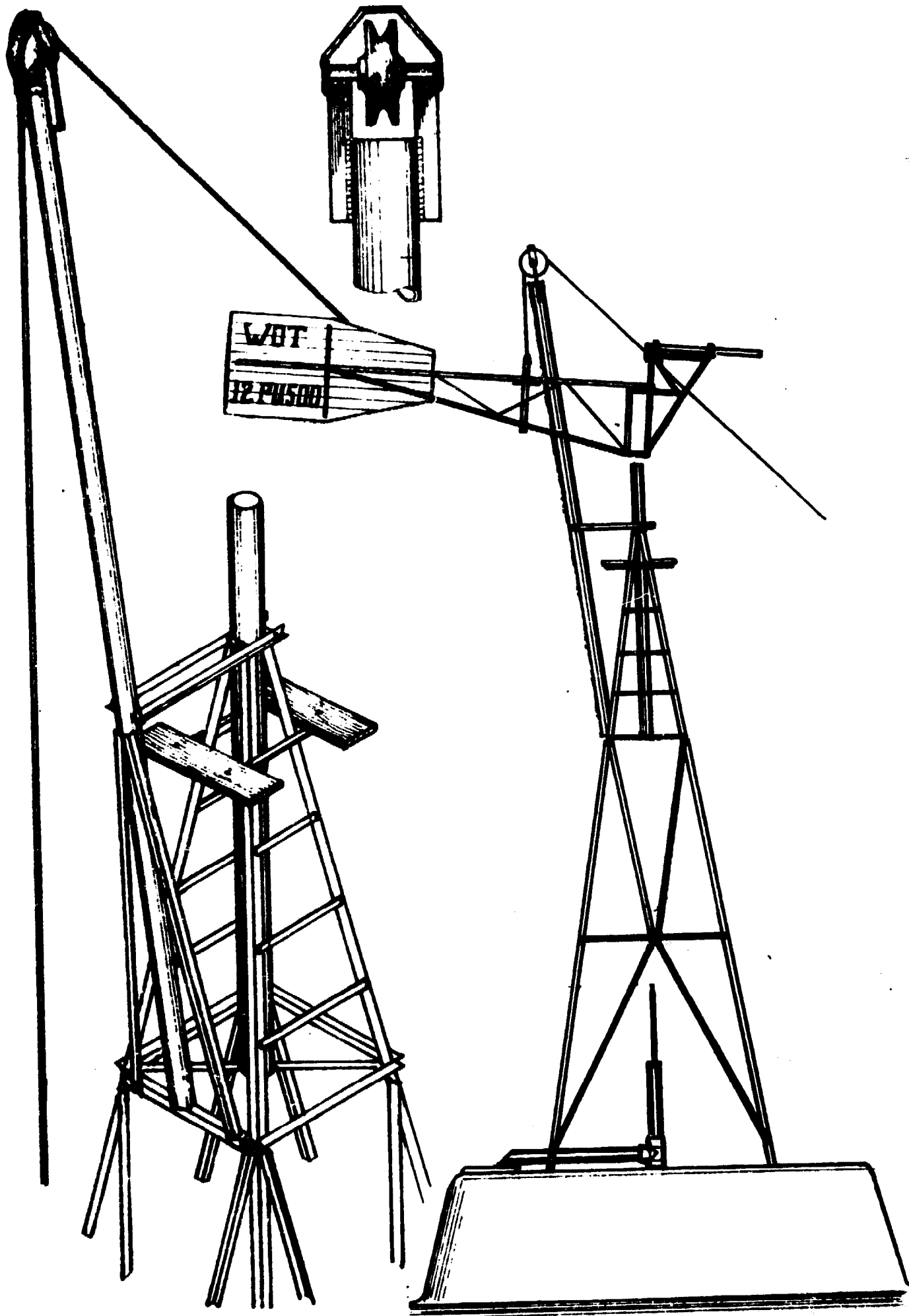
shows how these can be carried out quickly and safely by means
of a simple lifting device.

The needed storage tank is constructed from mud-made walls of
about 1 metre high, which are partly bricklined and plastered
in places where erosion or seapage is expected. The content of
a storage-tank amounts 50 up to 150 m^3, depending on the expected
need and the available area which can suitably be occupied by
the tank. An overflow pipe is masoned into the wall in order to
prevent that overflowing water will damage the mudwalls (if the
tank is full).
Since the local people often utilize the fresh water (delivered
by the windmill) for domestic purposes, like drinking, bathing
and doing the wash, the mudwalls in the environment of the deli-
very pipe definitely will be damaged. Therefore a simple
washing accomodation is provided by means of some masonry.
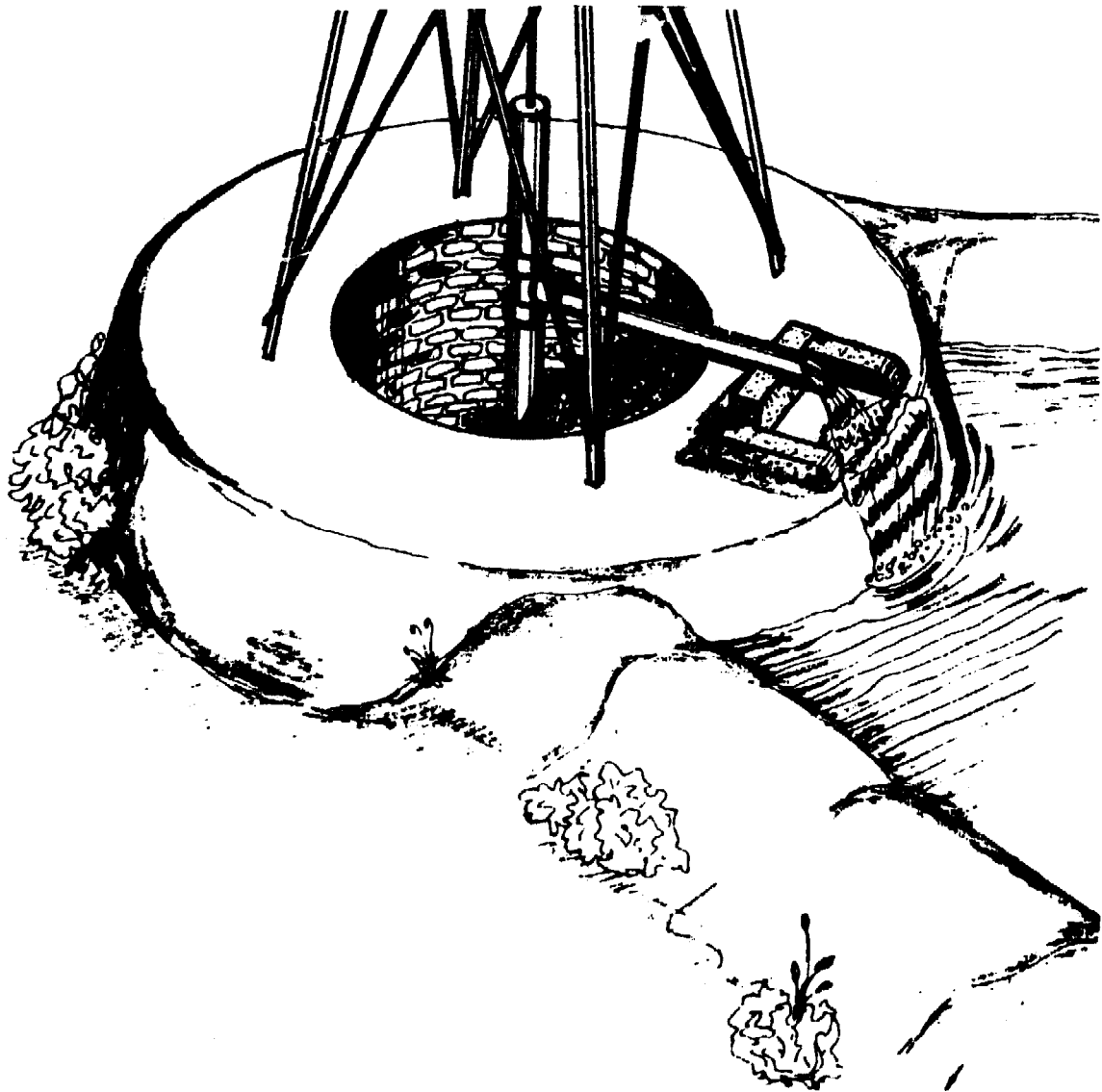
The water can be released from the tank by the farmer(s), using
a 2.5" hose, which acts as a siphon. Since this device is easy
to handle the water can be tapped at any place from the tank
and can be led to the plot(s), via one of the several irrigation
canals. The farmers have to be well-informed in handling the
windmills. During storms the windmills stop automatically, so
afterwards the farmers have to climb on their windmills in order
to reset the windmill into the wind. At the same time it is an
opportunity to inspect and lubricate the moving parts.

However, soon after the introduction of the first windmills
several breakdowns and long windless periods proved to be a
draw back for this particular technology. In spite of this the
farmers involved remained rather enthousiastic and local agen-
cies and Institutions developed interest as well.

2.2 The prototypes under development.

During field-tests some parts of the windmill components
needed redesigning in order to achieve a higher reliabi-
lity on the one hand and a costprice reduction on the
other.

Although this seems to be more or less contradictory
it is proved that both demands mentioned can partly be
conceded by surging for the most simple solutions and al-
ternatives which can be realized with the local means and
materials. Therefore it is a must to know the local market
and the means of production to which the designs have to
be adapted.

In practise still unknown factors like the extent of wear,
corrosion , vibration etcetera will remain and just a tho-
rough fieldtesting will provide the answers in the long run
and broaden the experience. This disadvantage of such re-
search is that failures can cause the distrust of the
local people in the windmill technology. To avoid this it
was decided to carry out the research and development as
much as possible under "laboratory" conditions. For this

reason a testing machine has been designed and built (see chapter 3) in order to speeden up the developement of the windmill.

These experiments as well as the field experiences led to the stepwise evolution of several windmill components which is described in the following paragraphs.

2.2.1. The towerconstruction.

The fabrication of the tower is very simple. The two pre-fabricated halves are composed: the towerpipe is pushed in, positioned and welded. In three hours the tower is completed with the help of three persons.(including the cutting and straightening of the materials).

The first and second platform are rather safe and comfortable to stand on but actually the first platform is superfluous: lubrication and repairs will always be carried out standing on the top platform.

Climbing on the tower is not done by means of a ladder: by coincidence the diagonals of the lower tower section just provide an adult the needed facilities to reach the first platform. For esthetic reasons the base-measurement of the tower has been reduced from 2.50 to 2.00 meters and still its foundation can surround the opening of the well which is seldom more than 1.8 meters in diameter.

Previously the total material costs of the tower amounted to ± 1/3 of the total material costs of the windmill, so a reduction in tower weight has been undertaken. Applying 1.5" instead of 2" angle-iron for the four towerlegs decreases the weight considerably but the legs became too slender which means that extra compartments in the structure must be applied in order to achieve sufficient strenght in the entire construction. Also some minor modifications, like guiding the towerpipe by flats inste..d of angles and cancelling the first platform de-creased the weight and costs. It was also proposed to reduce the length of the towerpipe,but extra angles and flats will be necessary to fix it properly and to provide sufficient rigidity to the tower-top structure.

However the original simplicity of the construction will turn over in an extra complicated one appearing during fabrication and defenitely resulting in loss of time due to mistakes.

Due to these above mentioned modifications the towerweight has been reduced from appr. 225 to 125 kg. Also another advantage is that the tower is easier to handle during transport and installation while the strength is almost the same. Climbing the tower is facilitated by applying some welded flats in the lattice.
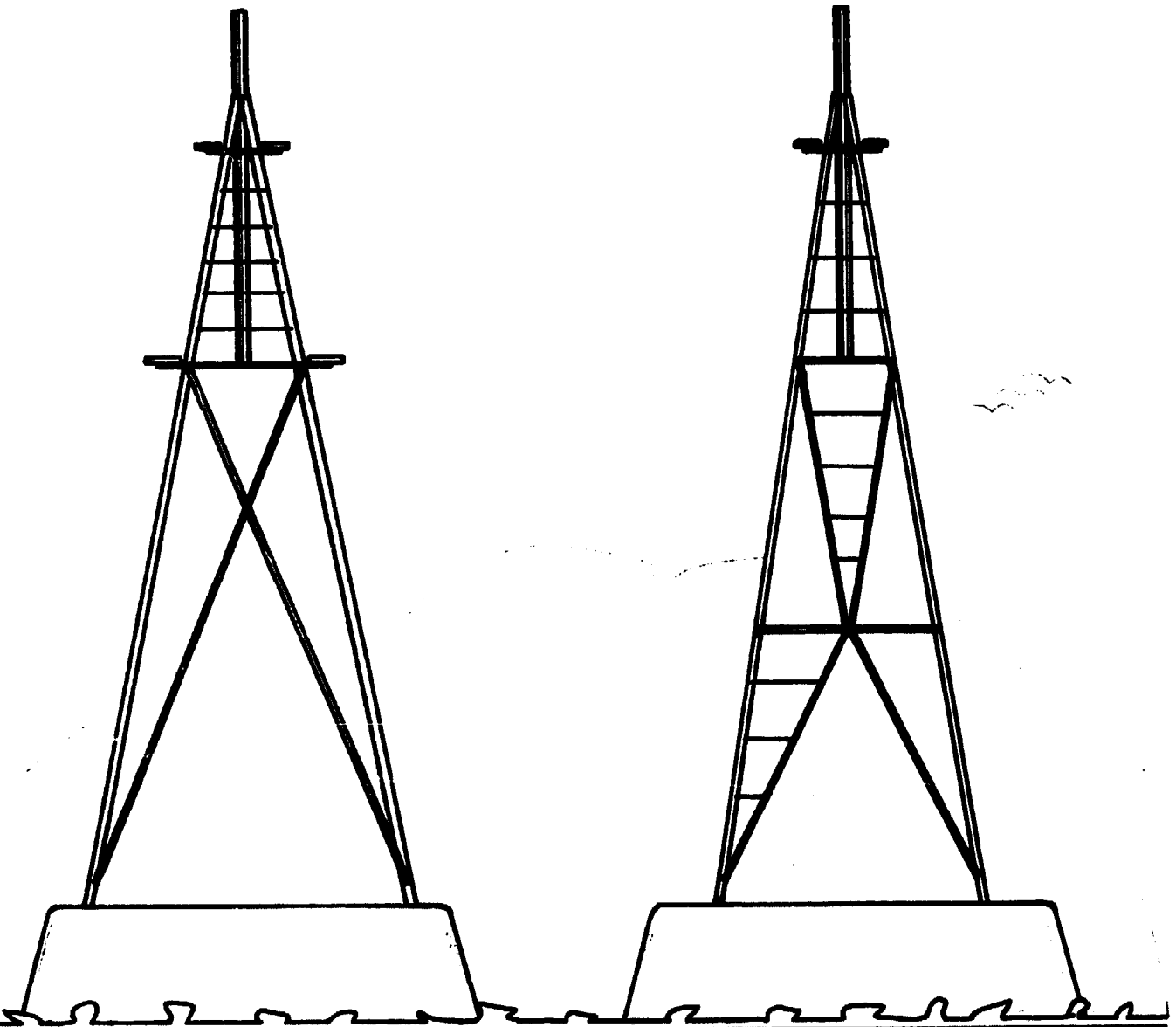
OLD TOWERDESIGN   NEW TOWERDESIGN

2.2.2. The headconstruction.

Some minor parts in the framework of the headconstruction
have been changed. The most important and valuable change
was reduction of the total length of the tail. This has
been done in order to obtain a better balanced structure;
the length amounts to 0.8 D and a better steering resulted.
Since the bearinghouses, which are available in India,are
rather wide some extra space had to be created for the support
of the rear-bearing. The flange of this support (angle-iron
2" x 2") is pointing to the front of the headconstruction in
order to achieve that.
Worthmentioning is that using a simple fixture for welding
the head saves time and moreover accurate work results.
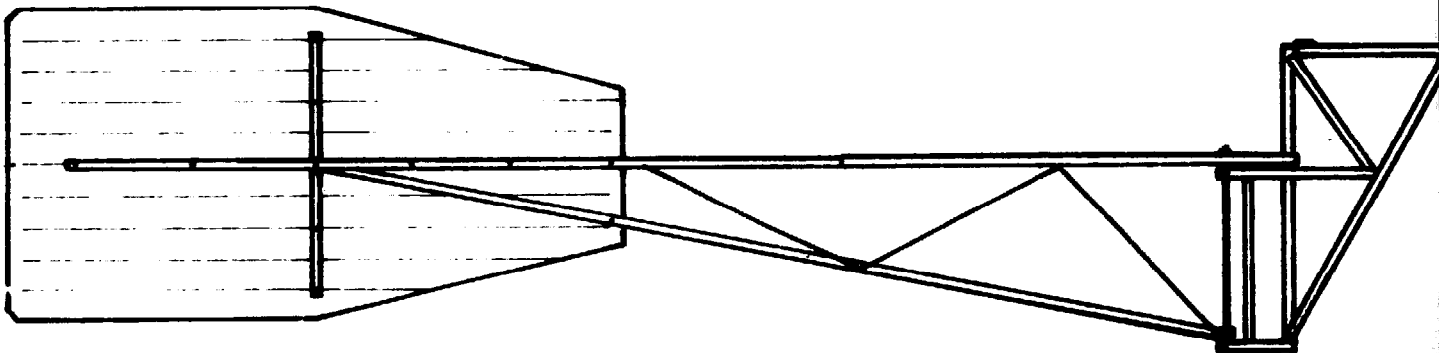
The security mechanism

In the very first designs the required accurate hinges of the
security-mechanism caused lots of problems concerning its
adjustment which did not remain constant. Initially a new
design has been introduced in all the Indian prototypes which
proved to be very simple and reliable. A helpvane,situated behind
the rotor and besides the headconstruction,is pushed backwards
by the wind if the adjusted preload of a spring is exceeded.
Due to that the achieved displacement unlocks the headconstruc-
tion from the tailstructure by opening a simple lockmechanism.
The moment, caused by the windpressure acting on the helpvane,
initiates a small rotation of the head (and so the windwheel)
with regard to the tail and winddirection.

The windpressure acting on the windwheel provides a further
rotation of the headconstruction until the windwheel is
turned 90 degrees out of the wind and a lock-system finally
fixates the head to the tail.
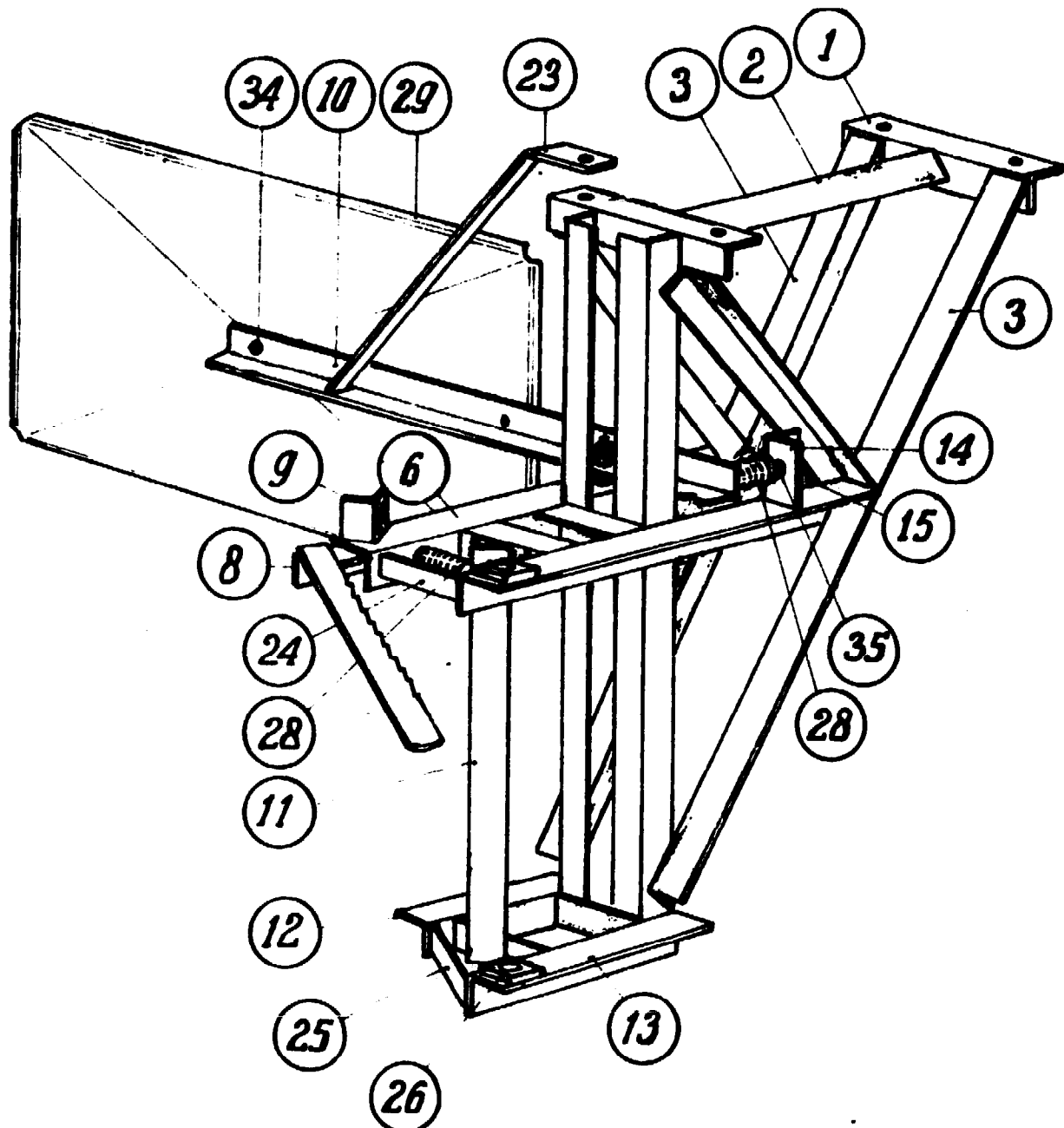
This lock has to be applied in order to prevent that during
storms , accompanied by sudden changes   of wind direction, the
tail and head will move separately which can result in very
high impacts in case of concussion of both parts. During this
securing motion a toothed handle prevents such separate
movements and it will not allow the headconstruction to turn
back with regard to the tailconstruction.
Since there is almost no damping in the entire security system
the kinetic energy, stored by the wind into the rotating

head_
tail-construction
properly balanced

old design

new design

gyroscopic moment
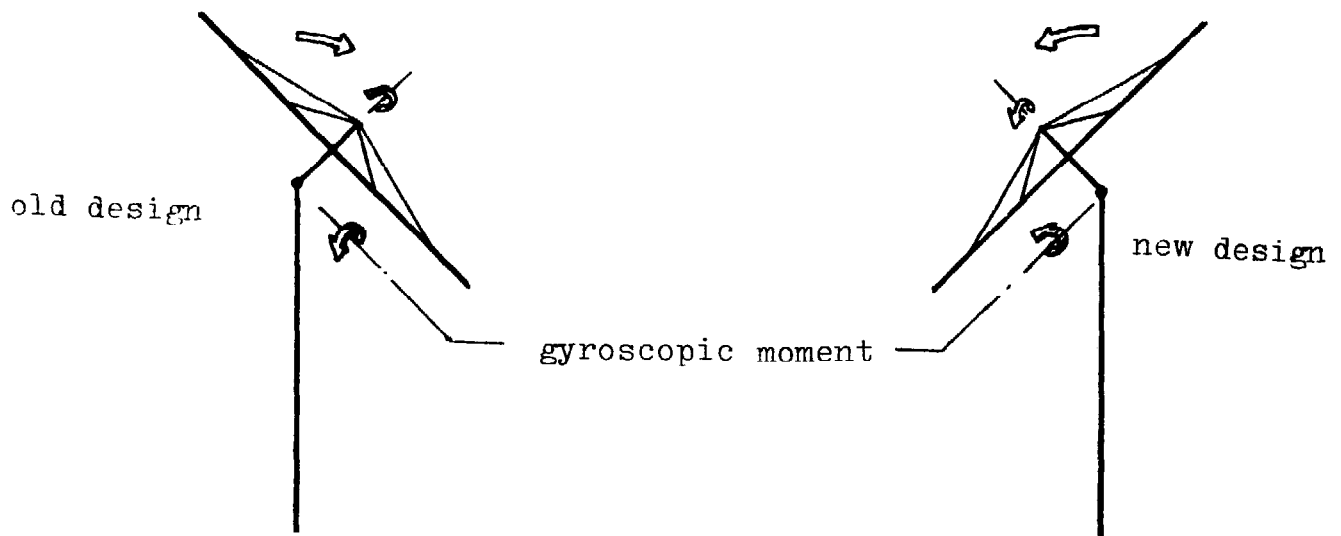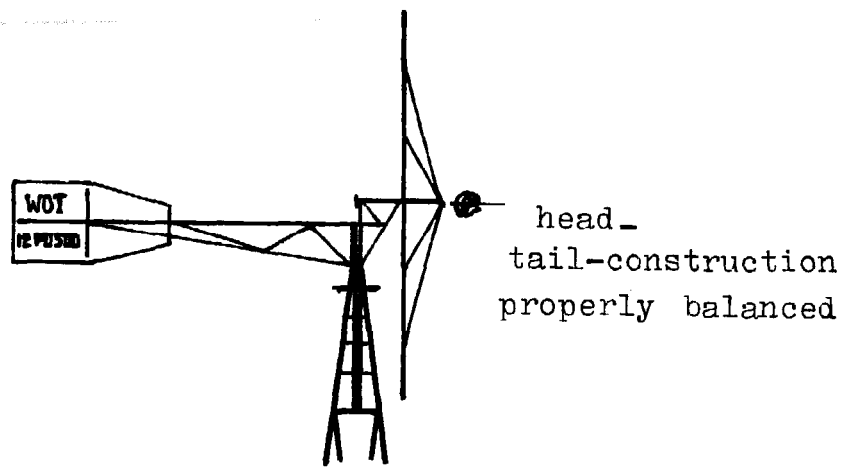
headconstruction, has to be dissipated. The impact taken by
the final lock, proved to be considerable at the end of such
movement of securing. Although there is the Coulomb's friction,
its dissipation capacity is too less and moreover its friction is
eliminated more or less by the gyroscopic moment which appears
during this security movement. So even this damping can be
neglected. In order to use this gyroscopic effect an extra
generated Coulomb's friction could be added in positive sense
by changing e.g. the direction of rotation of the windwheel.
Since the windwheel runs clockwise in connection with the bolted
crank pin, it is preferable to maintain this, but inverting the
direction of turning of the headconstruction with regard to the
tail will also lead to the same achievement.
Nowadays the headconstruction turns anti-clockwise out of the
wind (topview!), in order to obtain the extra needed friction
for damping caused by the gyroscopic moment. The impact is
reduced considerably and hardly audible.

Although the windwheel is kept out of the wind by the steering-function of the tail, it will still be hit by the gusts during storms. This is caused by the inertia of the total secured structure: during stormy weathers it was observed that the rotor changes its direction of rotation continuously : clockwise-anticlockwise, etc, which means that it is also attacked by these gusts from the back. This resulted several times in a forward bending of the blade-tips because its rigidity is low in forward direction.

In order to prevent such damage it was proposed to increase the rigidity by hammering a profile in the blades in the environment of the outside-ring blade-support. However, in this way the airfoil will be spoiled and so the efficiency will decrease. Another possibility is keeping the windwheel slightly into the wind in such a way that it will just not be attacked from the back. This is achieved if the windwheel is turned out of the wind less than 90 degrees: the rotor maintains clockwise running too. Nowadays the final lock of the security mechanism is situated in such way that the windwheel turns appr. 75 degrees out of the wind and forward bendings of the blade tips do not occur any more.
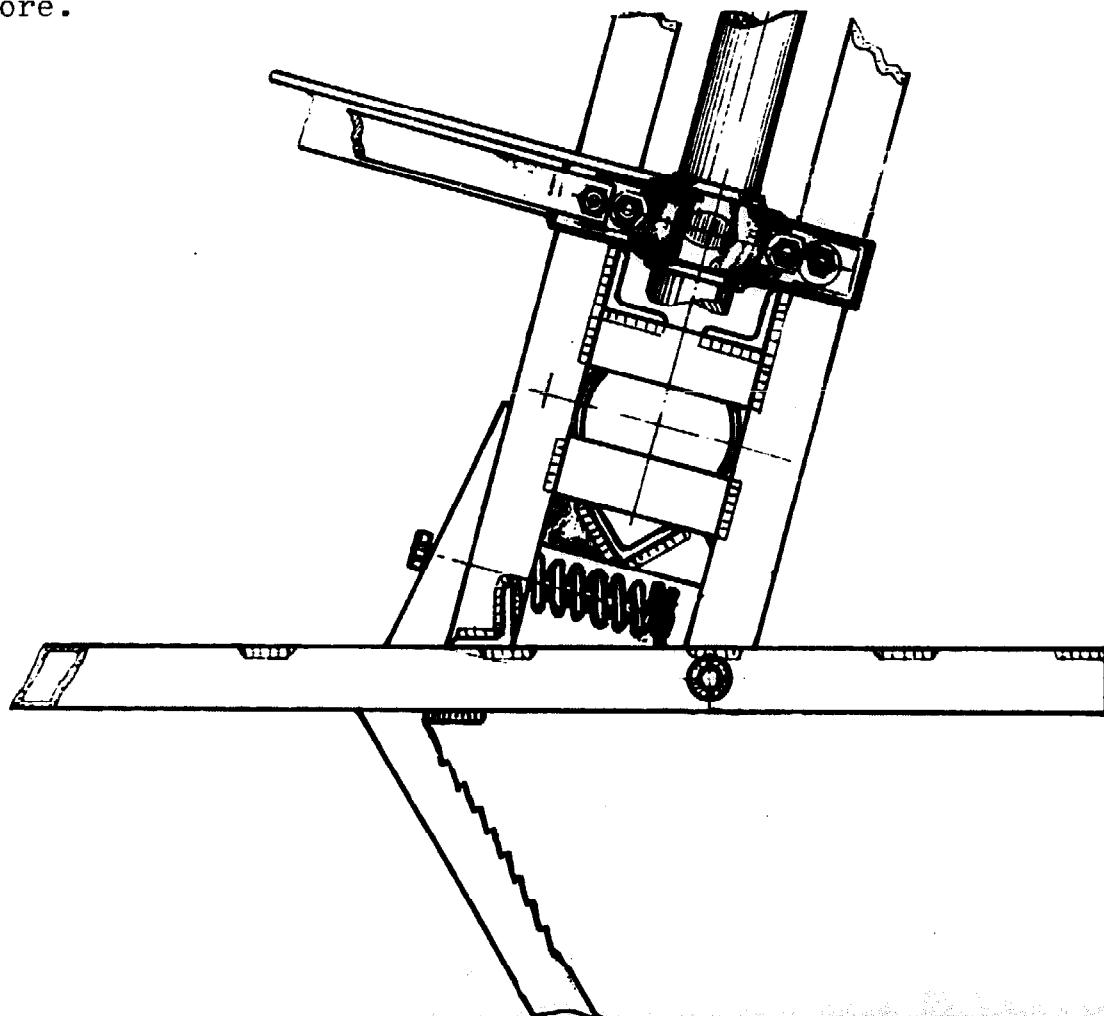
Though this automatic security device never failed the required reset procedure proved not to be practical in places where wind speeds fluctuate too much. Therefore efforts have been undertaken to develop systems in order to tackle this reset problem.

For instance a rope connected with the reset lever and led via the tail end to ground level facilitates a reset possibility which excludes the need of climbing the tower. However this system still demands the presence of an operator so it is not a great improvement.

As a matter of fact several fully automatic security devices are common in commercial windmills which principle is based on the momentum equilibrium of head and inclined hinged tail. However such mechanism demands a frictionless rotation of the head over the towertop which does not apply for the simple windmill design described here.

Therefore it was tried to derive a fully automatic security device from the existing mechanism which finally led to the following designs:
a. ratchet system
b. stepless system

Both new systems have in common that the helpvane releases the fixation between head and tail in order to allow the head to turn backwards out of the wind. This process continues till the helpvane does not detect any exceeding windpressure anymore with the result that fixation between head and tail is restored in that particular position. This fixation is achieved by means of a ratchet- and a stepless friction mechanism respectively in these systems.

Minor changes in wind direction cause minor oscillations of the tailvane which are most essential in order to return the head and rotor more or less facing the wind again if the storm decreases. This return motion is possible due to the "one way" character of the ratchet and stepless friction me-

chanism as long as the helpvane does not detect any over-
pressure in its newly found position.

So both the systems allow the head and rotor to turn out of
the wind to a certain extent depending on the actual wind-
forces. The oscillating tail vane will see for the automatic
reset. However for the simple ratchet system a minimum oscil-
lation of the tail vane is required during the automatic
reset procedure in order to reach for the next tooth of
the ratchet. In case of rapidly fluctuating wind directions
impacts on the teeth will occur which may lead to wear of
the ratchet and its counterpart in the long run. Also an
impact occurs at the end of the securing motion if suddenly
the helpvane puts the ratchet in action again. Observations
made during field tests just partly confirmed these disad-
vantages and presently the ratchet devices are still obser-
ved.

The "stepless" system, as the word suggests, responds to
very minor oscillations in a less hesitating manner than
the "ratchet" does. The pivoted catch induces sufficient
high friction between the disk (jointed to tailbeam) and
contact surface (jointed to head) resulting in a tight grip.
In this system also the sudden impact exists at the end of
the security motion. This disadvantge has been eliminated
by allowing slip in between the disk and tail beam in order
to dissipate severe shocks.
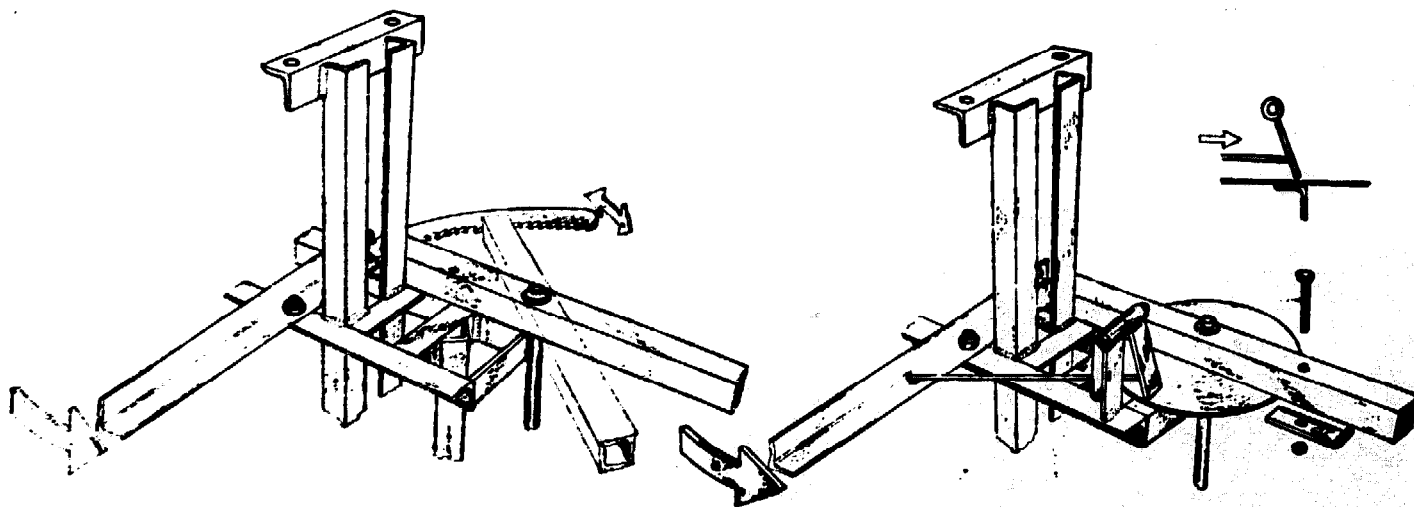
2.2.3 The moving parts: Crank-mechanism, bearings, etc.

Although the principle of converting the millshaft rotation
into the up-and down-movement of the pumprod was still main-
tained, the bearings as well as the simple crank-mechanism have
been changed a lot. Many thousands of revolutions of the windmill
were necessary to indicate and to   eliminate the weak components
which finally resulted in a complete redesigning of these
essential parts which are still simple in manufacturing and
assembling.

The mainshaft bearings

Two kinds of loads actuate on the two main-bearings.
The front bearing carries a radial load of appr. 1000 N (rotor-
weight) and a thrust-load upto 7000 N, while the rear-bearing
takes the alternating forces carried out on the crank-mechanism.
If these bearings are nicely adjusted the above mentioned
assumption will be actual but unfortunately there are constraints
in practice.
Due to considerable unroundness of the shaft and poor quality of
the bearing-bushes (sleeve-type) a proper fixing of the bearings
to the shaft often led . to complications. After some time the
bearing-bushes got loose and the shaft started rolling and
sliding in them. This rolling effect will increase while the
shaft slides backwards by the windpressure acting on the rotor
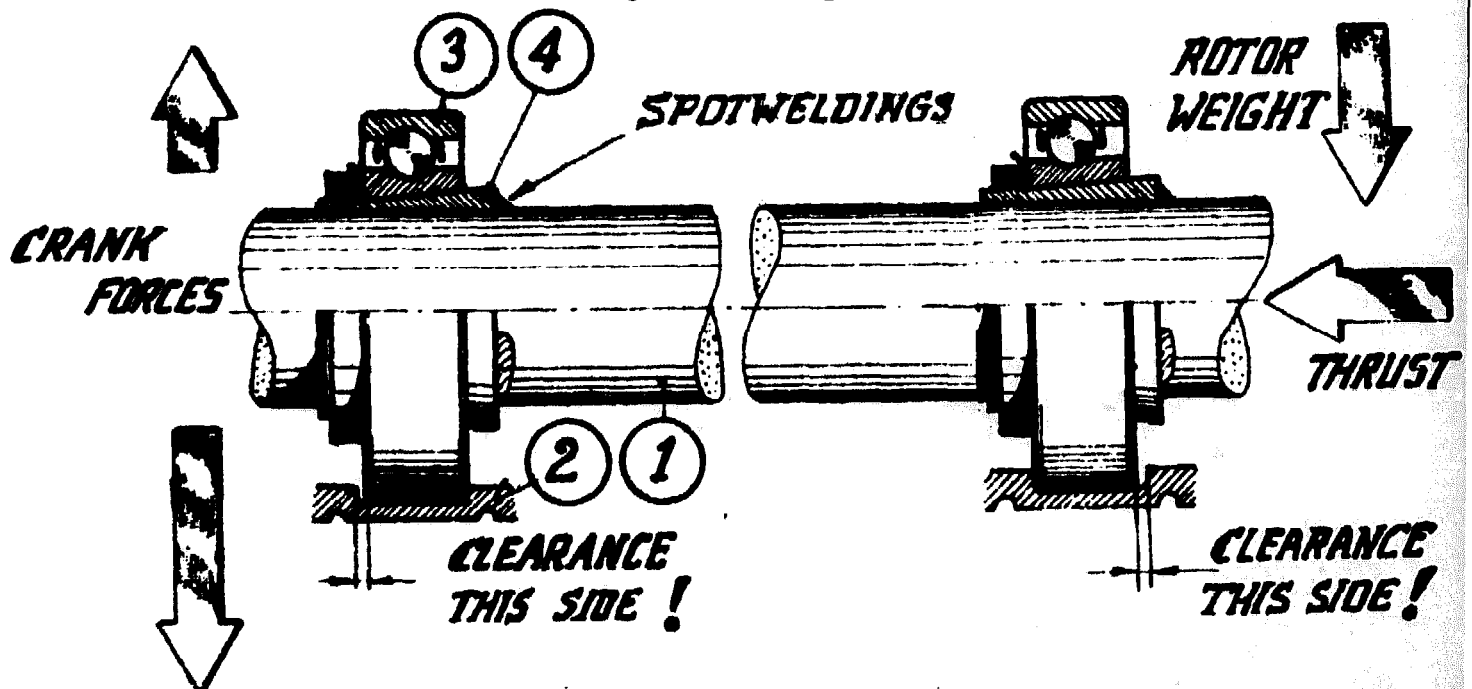
and the sharp edges of the angle iron hub start scraping at the
front of the bearinghouse.Of course,few thousands of revolutions
are sufficient to destroy the bearing-house completely and the
ball-bearing (due to the scraping-dust). This problem is solved by
locking the bearing-bush to the shaft by a <u>short</u> spotweld so
that the shaft cannot roll and slide. Proper position of the
bearings to the shaft with regard to the axial clearance of the
bearings between the inside edges of the bearing-housings will
assure which bearing is going to carry which load.

     Besides,there were the problems with the bearings them-
selves. Since the bearinghousings are bolted to a welded frame-
work self-aligning bearings (double-roller type) have to be
applied.
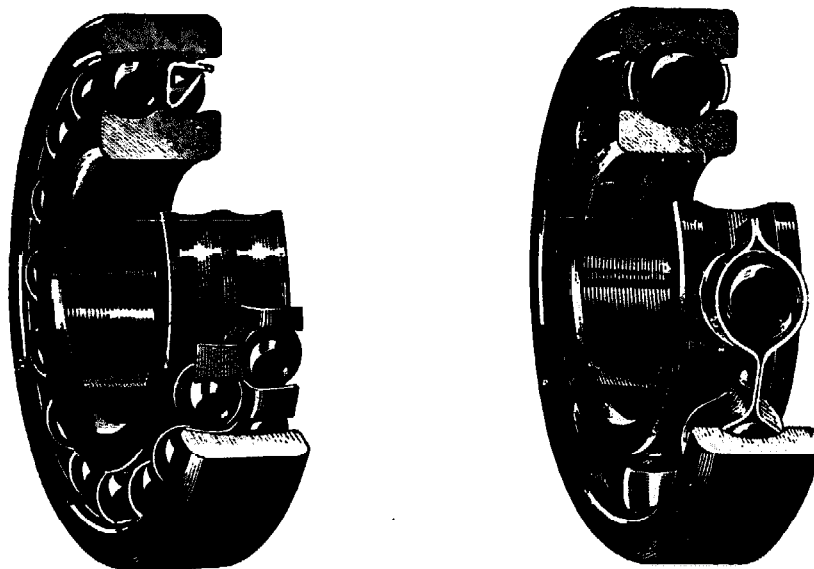
double-roller selfadjusting   single-roller

Although these bearings are rather oversized, according to the
S.K.F.-specifications, the lifetime proved to be too short
(appr. 1 ÷ 5 millions of revolutions). This means that these
specifications are not representative of the bearings available
in the local markets. Many bearings of several trademarks are
offered but specifications are unknown and the quality is not
guaranteed. Often these bearings are "repaired" old ones: the
rollersurfaces have been lightly ground and oversize balls are
applied. The result is surprising; only insiders can detect
the minor differences from the new ones . Since these bearings
are not able to stand the loads and S.K.F. is not easily

available one has to go for the best qualified trade
mark like N.B.C. The latter really offers good quality
bearings and sleeved bushes. Unfortunately the costs are
considerable.
The same counts for the "one piece" bearing blocks (im-
ported) which are of excellent quality but also quite
costly.

Another possibility is to apply single roller bearings
which can be loaded more than the double roller type due
to the bigger contact surface. But a proper alignment of
the housings is absolutely necessary and moreover the
rotorshaft should be greatly straight.
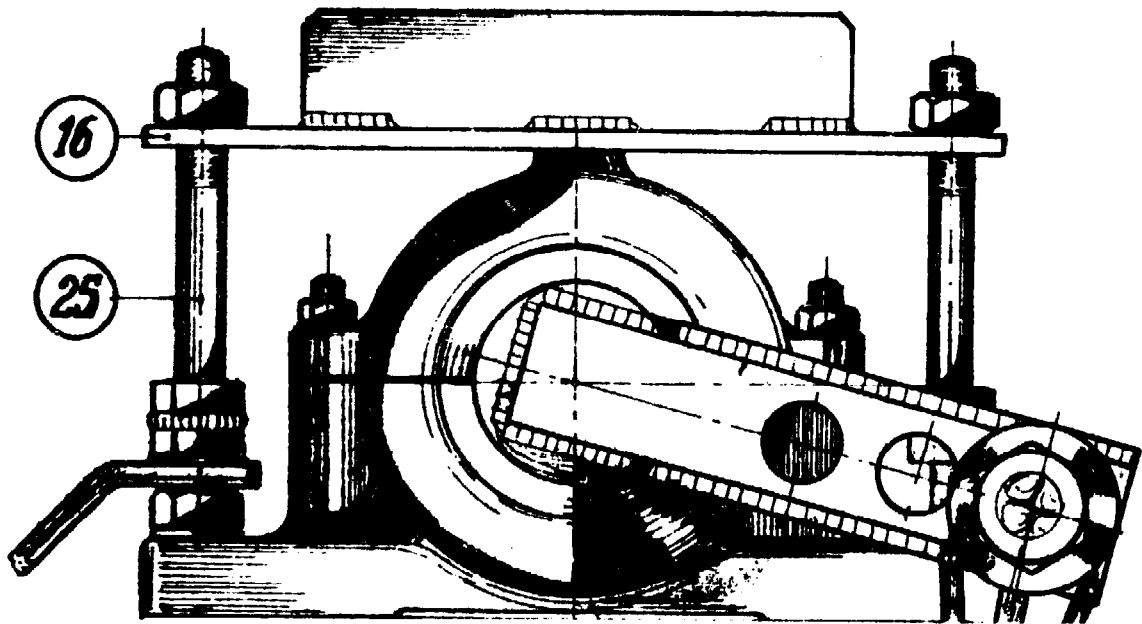
Since the rear bearing is loaded by alternating forces a crack
of the weak bearing-housing-top might occur. This leads to a
disastrous damage of the rotor, because it will tumble forward
and hit the tower. To avoid this a simple supporting bracket is
applied which supports the housing-tops in carrying upward forces.
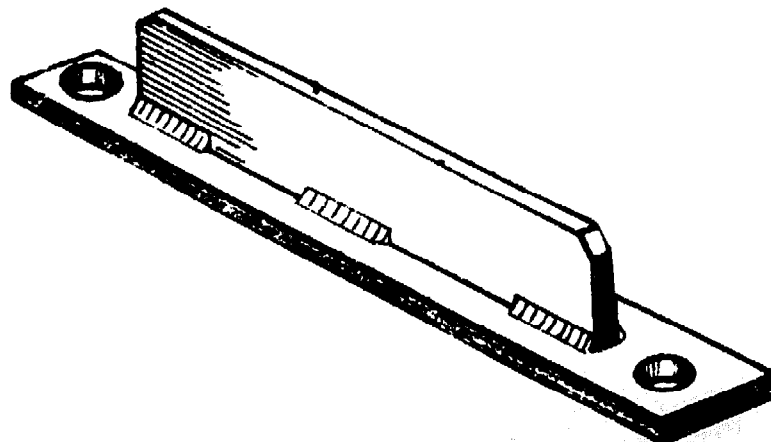
The crank

In the previous designs an "open" crank has been preferred to a
"closed" one . The advantage of the application of an open crank
is that the radius of the crankpin can be adjusted easily.
Moreover the assembly of the connectingrod is very simple because
there is no need of a partable construction of its bearing-housing
and bearing.
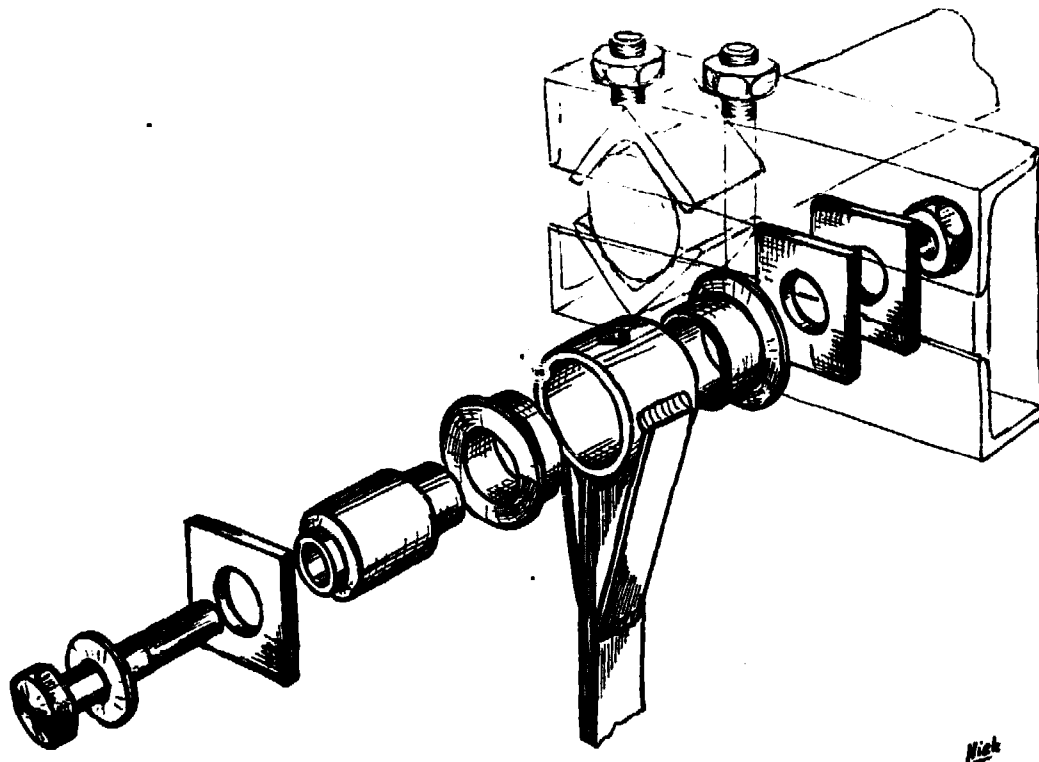
An adjustable crankradius always creates the opportunity to
"match" the windmill to the pump, depending on the actual
windspeed and the elevationhead.Therefore the first windmills
have been provided with a slotted crank in which the crankpin
can be positioned at any radius between 50 and 125 mm (2" and 5").
This slotted crank, composed of angleiron (1.5"), was clamped,
just like the hubconstruction of the windwheel at the end of
the mainshaft but this unkeyed connection proved not to be
reliable.Slipping led to fretting and the end of the shaft
as well as the crankhub were spoiled.Several efforts have been
undertaken (central bolts, keys),but a 100% reliable clamping-
device has never been achieved.
Welding of the crank to the shaft-end solved the problem as long
as replacing of the mainbearings of the millshaft was not
required. In general welding to a shaft is not advisable,because

straigthening is difficult to maintain., but also the metal-structure
will be affected or even spoiled due to the enormous heat.
Since this welding takes place at the end of the shaft which
un-straightning of it will occur in the rear section of the
shaft which can be eliminated in the crank itself. Also the affected
metal-structure is situated behind the rear bearing, so if a
fatique-crack will appear in this place the rotor remains and
a further damage is avoided.
Although the slotted crank served its purpose very well,it
needs a very tight fixing of the central crankpin-bolt in order to
prevent that it gets loose and starts sliding in the slot.
This was a result of experiments with 8" dia pumps (normal pump-
dia 6"). Moreover in practice only two of the infinite adjusting
facilities were used: maximum and minimum position! This led
to the idea to introduce a new crank with three or even two
crank radia only.
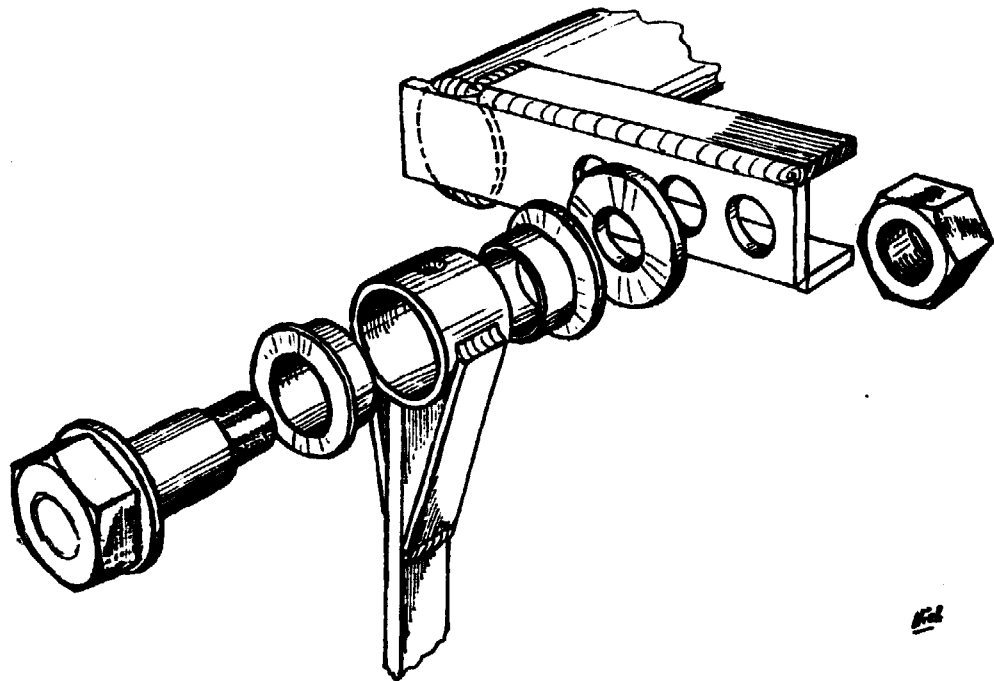
The crank is composed of several pieces of flat iron which
are welded to the shaft end. Later on it proved to be neces-
sary to create reinforcement by applying an extra flat
situated near the shaft in order to obtain sufficient
strength. The U-shaped crank provides a sliding fixation of

the crank pin nut which facilates assembling. For aesthetical and safety reasons the sharply edged tip of the crank is closed by welding a small piece of flat to it.

A crank made of a solid steel block, directly screwed to the shaft, showed also satisfactory results especially regarding the revision to the rear bearing. However, generally, the less sophisticated construction of the composed crank, as described sofar, is preferred.
The bearing surface of the bolt-type crank pin loading the crank plane proved to be too small for yield of the material occurred resulting in a loosening crank pin- One can imagine that this leads to disastrous damage of all the components concerned (including the connecting rod and its bearings).

Improvement has been achieved by applying a collar bush combined with a central crank bolt which latter ensured a sufficient strong connection of the bush to the crank plane. No indication of yield or loose crank pins have been discovered so far.
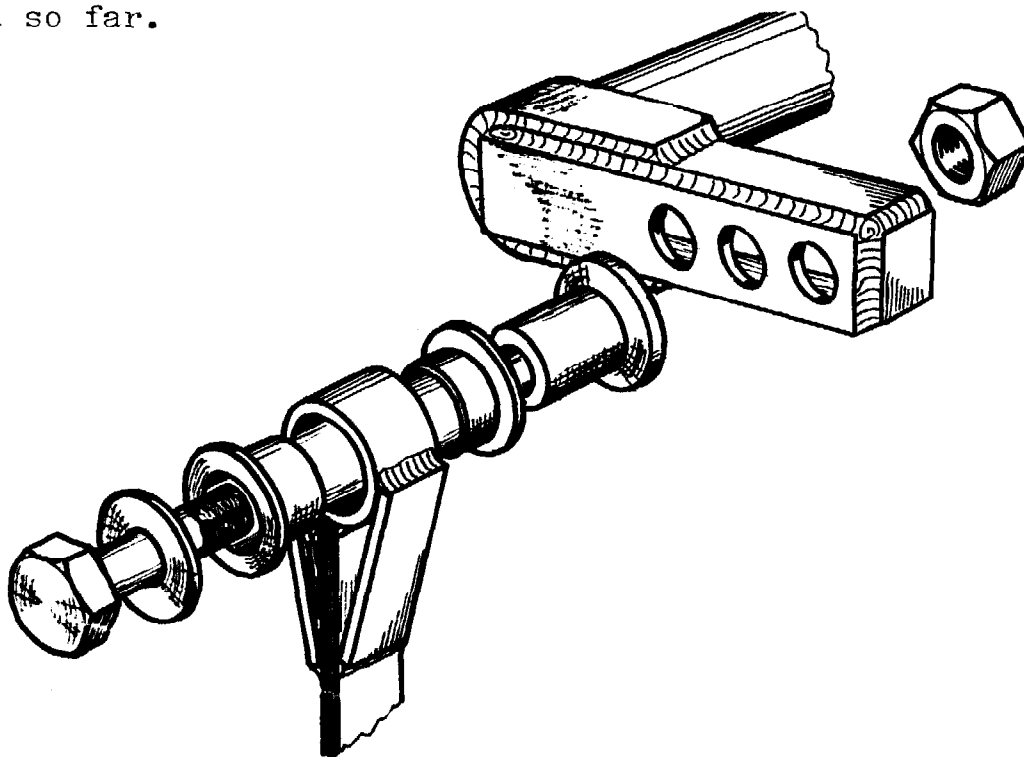
The connecting rod

To the upper-and lower-end of the connecting rod (flat iron) cylindrical steel-bushes are welded which enclose the flanged nylon bearing-bushes. In the beginning these steel-bushes have

been made of 1.5" gaspipe sockets after the screwthreads have been
removed by turning. The remaining thickness in combination
with the poor metal properties lead to unroundness and even cracks
of these bearing houses, in spite of the fact that reinforcement-
flats have been applied. Therefore it was decided to apply
stronger bushes machined of scrap which remains from the millshaft
if it is sawed on the right length. Nowadays the thickness of the
bushes amounts to $\pm$ 6 mm. During composing and welding the
connecting rod, a simple device is applied in order to obtain a
correct parallelism of both bushes. The unroundness of the bushes,

WELDING HELPDEVICE
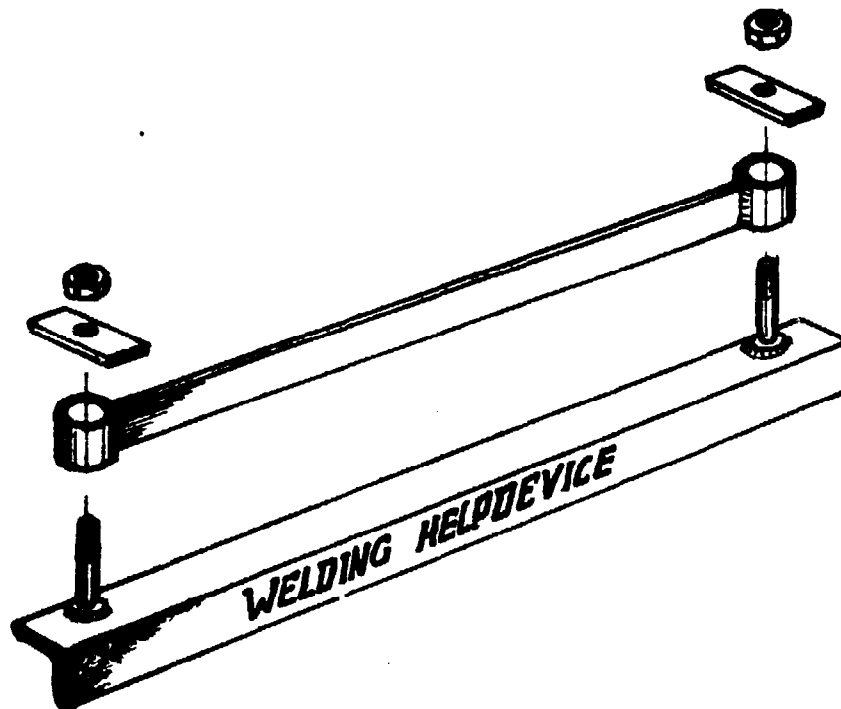
caused by the welding-procedure, does not affect a smooth
function of the nylon bearings which fit nicely over the
crankpin with 0.2 $\div$ 0.3 mm. clearance in radial sense. An axial
clearance of appr. 1 $\div$ 2 mm. is usual. Up till now no nylon-
bearingbushes wore out ,and were replaced accordingly.

The crosshead

The crosshead is situated and guided in the towerpipe. As
mentioned before a rather high rate of wear of the very first
crosshead has been noted, so another one had to be designed.
Since the towerpipe (gas pipe) always has a seam inside a
proper guidance of the crosshead is impossible unless the entire
seam is removed. This can only be done by grinding, but the

required grinding equipment is rather advanced, not always available
and moreover the resulting surface remains rough.
Efforts undertaken in obtaining an inside surface of an
acceptable roughness were not succesful during constructing the
first prototype (KK I) in India.
The idea was born to install a P.V.C. insert inside the towerpipe.
This insert is made of a   4" P.V.C.-pipe of sufficient length
from which a strip is cut in such a way that the circumference
measure is a little less than the inside-one of the towerpipe.
The opening just provides enough space for the seam which is
completely counter-sunk then. At the bottom a small bolt ,
screwed through the towerpipe,locks the insert in axial sense.
Since P.V.C. is rather soft the crosshead must have a sufficient
wide contact surface in order to prevent high specific loads.
Also sharp and hard edges must be avoided (scraping!). It was
decided to compose the crosshead of hardwood because this
material was expected to be a better counterpart for the
P.V.C.lining than e.g. steel.
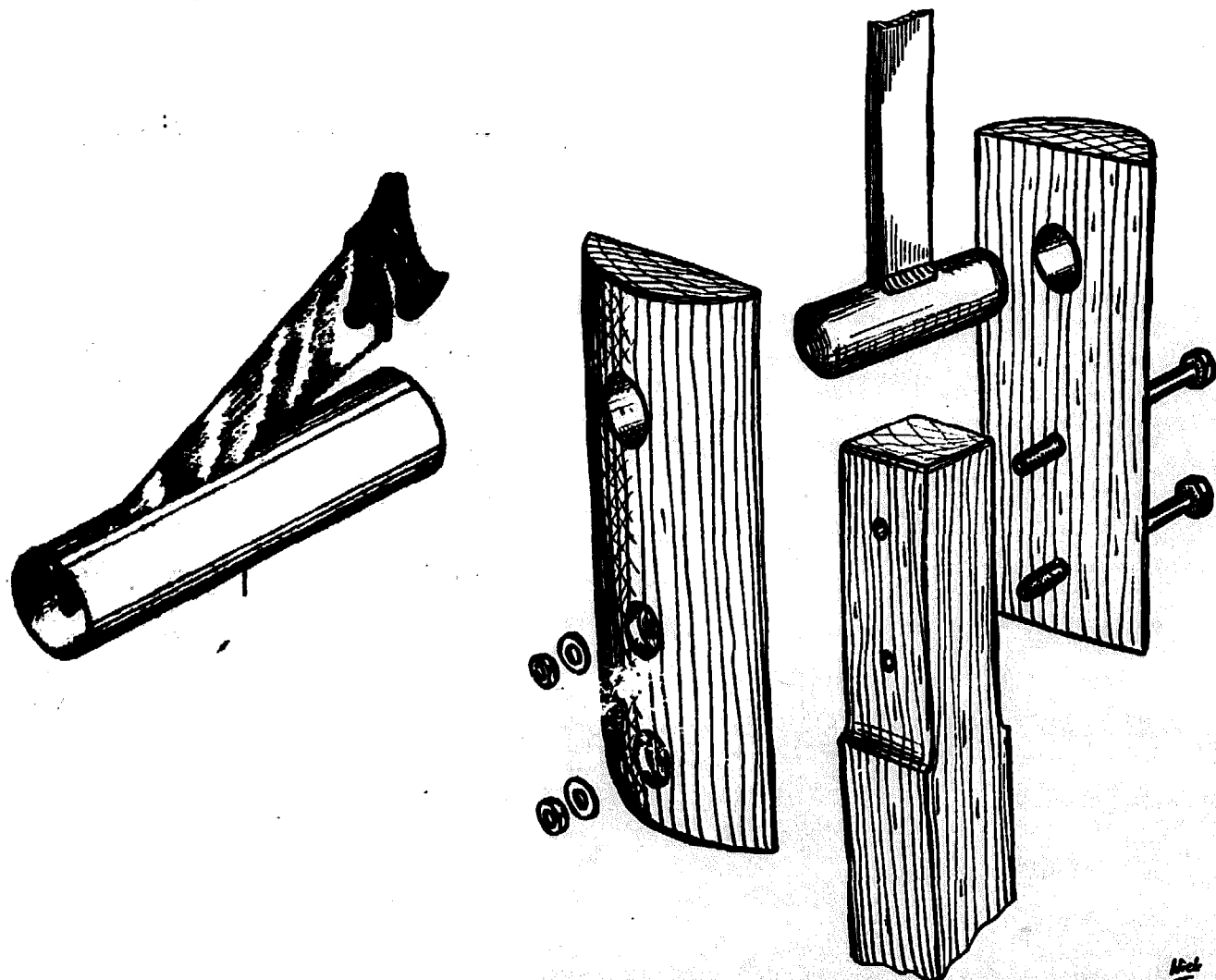
Previously the first pumprods also consisted of wood (ϕ 1.5";
see next chapter) and the first crosshead had been composed
of two parts of sesam-wood which clamped the pumprod by
tightening two 3/8" countersunk bolts. The first experiments
seemed to be satisfactory but after some time the upper-portion
of the two halves started wearing due to the gap of bearing-
surface between both parts and the crosshead became more or less
conical. Later on the steel pumprod has been introduced,which
caused a complete redesigning of this crosshead. After some
trouble-shooting the crosshead evolved, in a simple cylindrical
wooden part in which the steel pumprod is hammered and clamped
by two 3/8" bolts at the bottom side. In the top a square hole
is made by chiseling in which the lower-section of the connecting-
rod is enclosed.
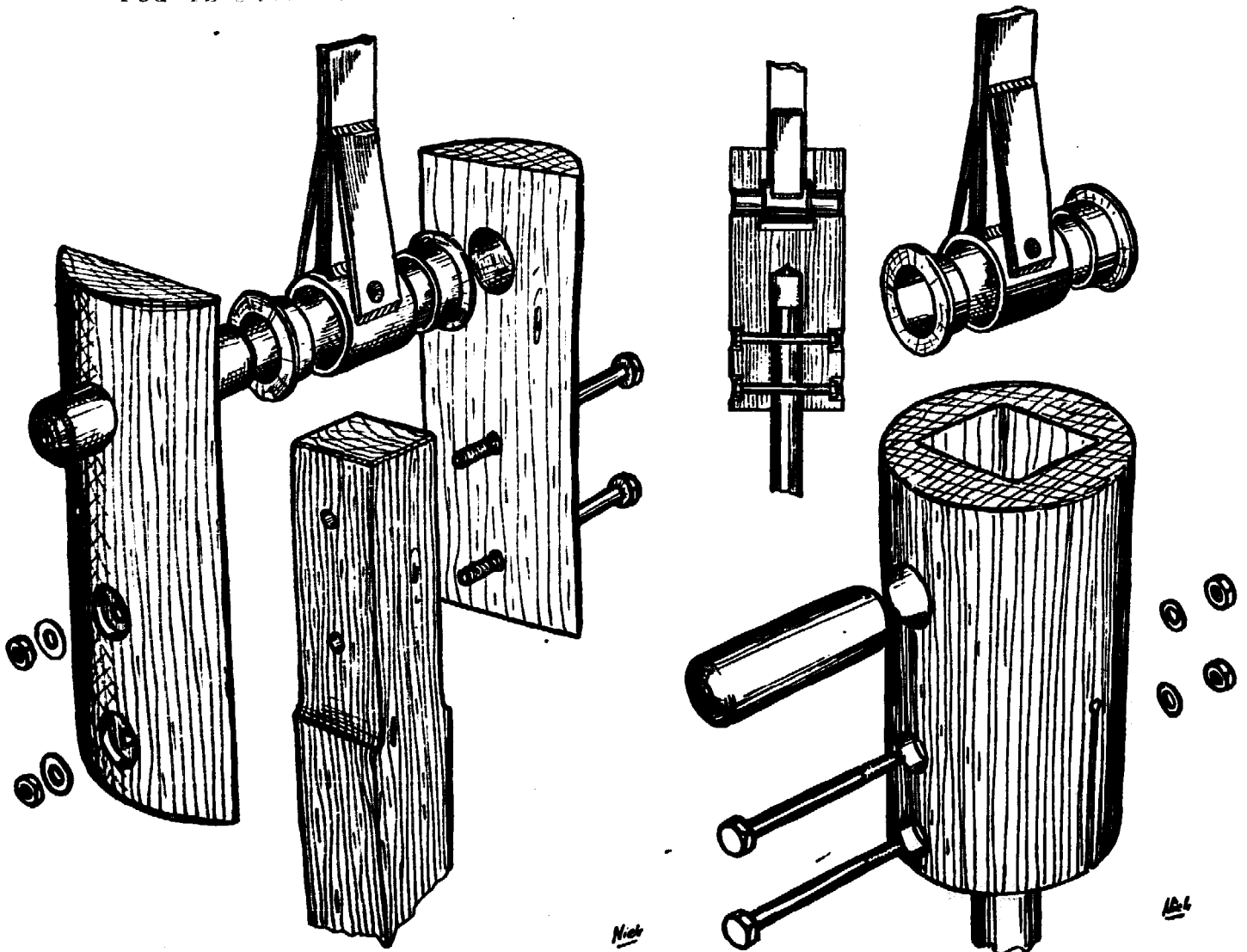
The measurement above the crosspin of the wooden crosshead
amounts to appr. 2" in order to obtain sufficient strength.

Lubrication is applied by grease from time to time, but also soaking in expired motoroil before assembling assures a smooth motion, and oil in the square hole provides a continuous lubrication

2.2.4. The rotor

The twelve-bladed rotor is composed by the rotorframe, bladesupports and blades which are completely prefabricated in the workshop. During installation of the windmill the rotorframe is lifted and installed first and the blades are fixed afterwards.
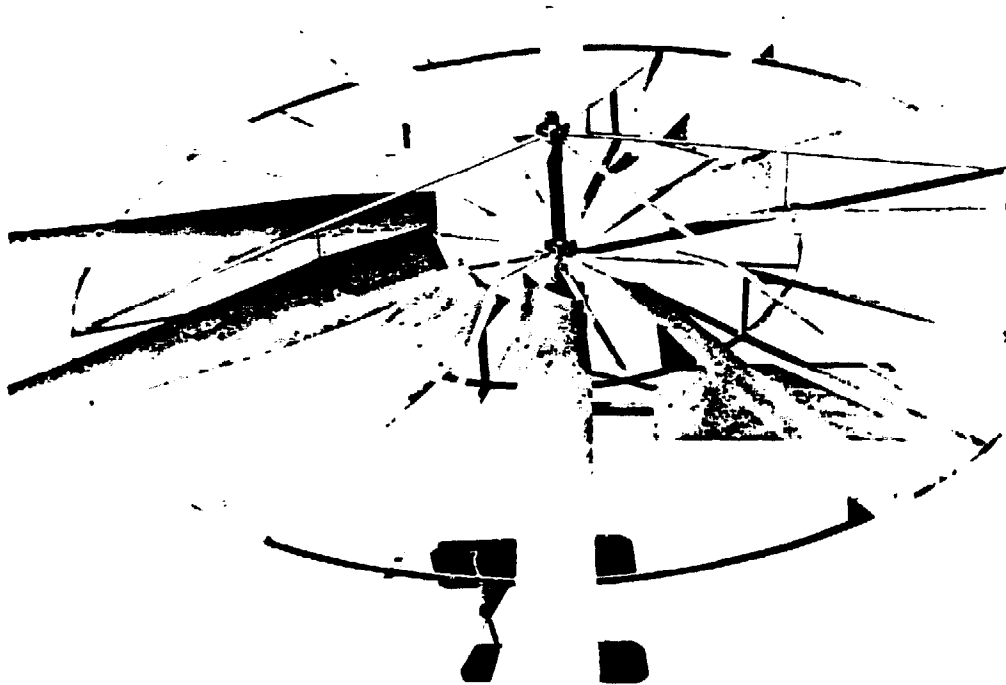
The rotorframe.

Using the shaft, bearings and headconstruction as basement during composing and welding of the rotorframe the latter is composed very accurately. However the disadvantage was felt during transportation: the diameter of the rotorframe amounts to : 3.8 m, so its transport can be complicated. Nowadays the rotorframe is composed of two halves: which can be obtained easily due to the fact that the hub is partable.

In the beginning there were some doubts concerning the
strength of the inside ring which is supposed to carry the
centrifugal forces acting on the blades. Although no defor-
mation has been indicated the rotorframe of the next four
prototypes have been reinforced by extra flats which provide
extra support to the inner-ring. Later on this precaution
proved to be superfluous bacause an overhaul of the KKI-
windmill pointed out that the inner-ring still was perfectly
circular . Since the rotorframe proved its reliability in
combination with the broad blades of the old design no
doubts remain concerning its strength in combination with
the smaller blades. Nowadays the rotors are constructed
without these extra reinforcement flats.

The hub.

During the introduction of the bigger pumps (6" and 8" dia)
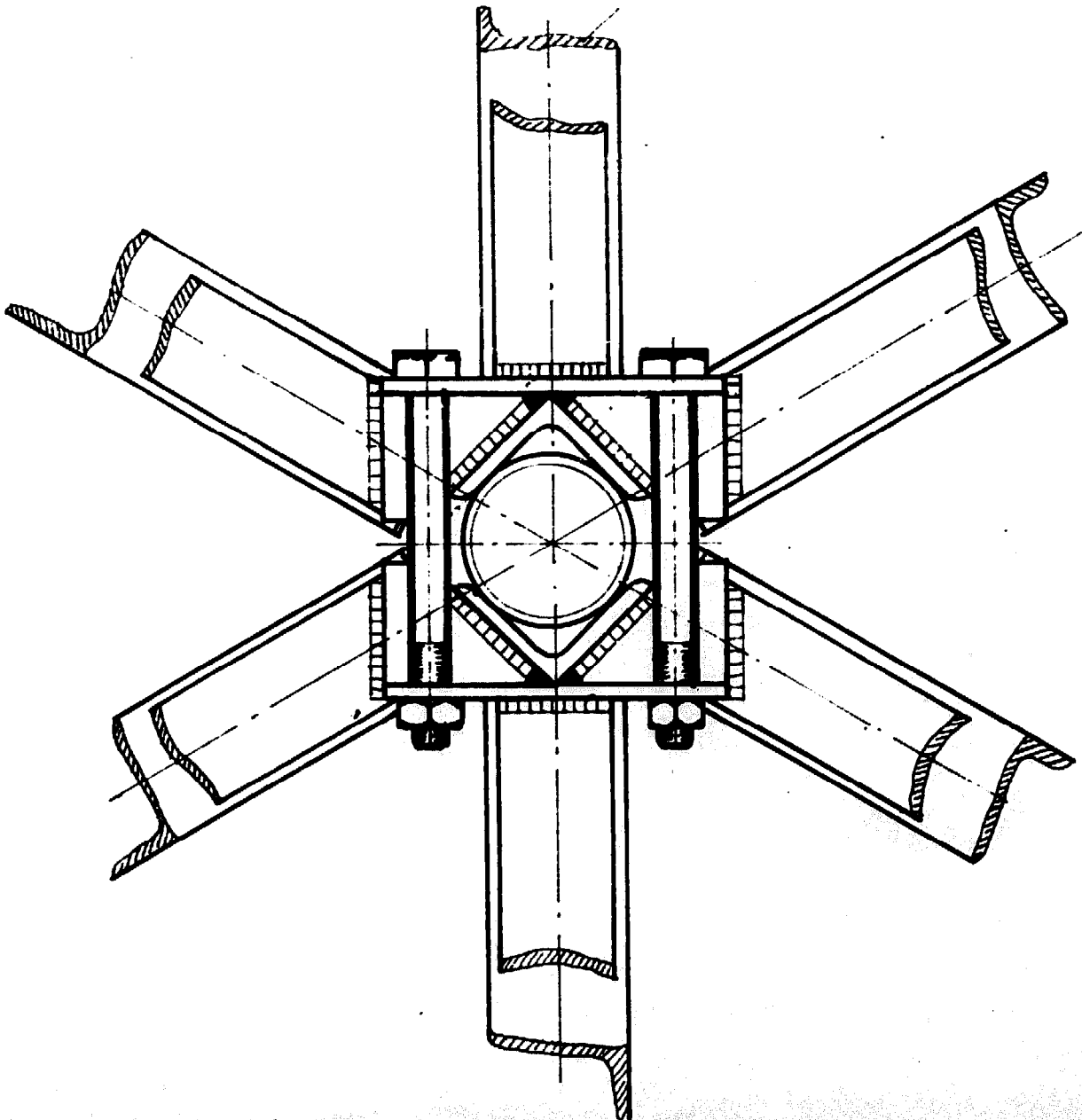
and allowing the windmill to run in windspeeds higher than
12 m/s the hub of the rotor started slipping and screwed
backwards over the shaft.
Obviously the clamping of the hub was not sufficient
in order to prevent the rotor from slipping which results
in wrong position with regard to the shaft (Rotorframe
is fabricated on its shaft to eliminate unstraightness)
Defenitely a "shaking" windwheel will result,which leads to
vibrations and wearing and tearing will result of several
parts like the security-lock, head/tail-hinge and
head/towerpipe-joint, so avoiding this is a must.
Practice proved that just a bolt through the shaft acting
as a key is not sufficient: if slipping occurs the two
hub-halves (angle-iron) will be bent open and the entire
hub- and rotor-construction is spoiled.
Finally an acceptable solution for this problem was found
by applying two extra clamping angle-irons clamped firmly
by two 1/2" bolts.
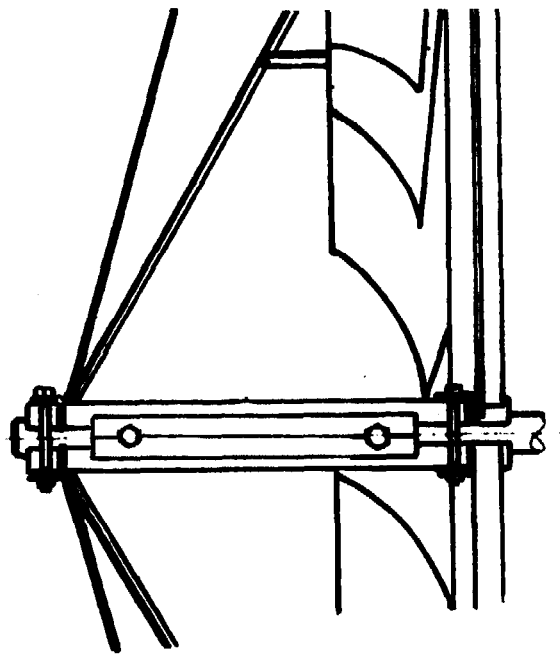
Blade-supports.

The blade-supports are manufactured from sheet-iron and if
one is able to prefabricate two sets of twelve equal blade-
supports an accurate running windwheel will result. This is
achieved by making the supports in sets of twelve,while the

first of each set is used temporarily as a simple shape during
cutting and drilling of the other eleven supports. After
marking and hammering nice blade supports result in satis-
factory shape and sufficient strength. However, in the first
prototype windmills the inside blade supports proved to be
not fatigue-resistant as cracks developed at the bolt-hole
edges, near the inside ring. This problem was solved by apply-
ing 2mm sheet iron for the fabrication of the inside blade sup-
ports. The outside blade supports served their purpose best
and are still made of 1mm sheet.

Some efforts have been undertaken to compose the blade supports
of pre-drilled and curved flats which are welded to the rings.
Unfortunately this experiment turned into a failure. Obvious-
ly due to the rigidity of these supports the forces setting of
the blades in them was often accompanied by initial stresses
resulting in increase of fatigue sensitivity of the blades
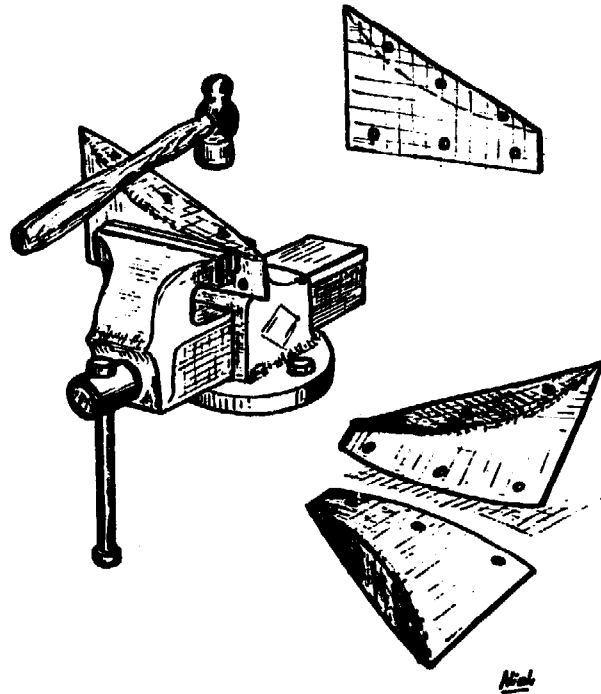
in places near the bolt holes. Such complications do not
occur if the sheet iron blade supports are applied for these show
"elastic" properties which are stress-relieving.

The blades.

The first three prototypes (KKI, KKII and RTI) have been
equipped with sheet-iron blades which were curved 10%.

Due to a mistake in the assumption the blade-shape was calculated for 7% airfoils which resulted in broader blades than was actually needed. The production of the blades from 1 metre wide standard-sheets proves to be rather
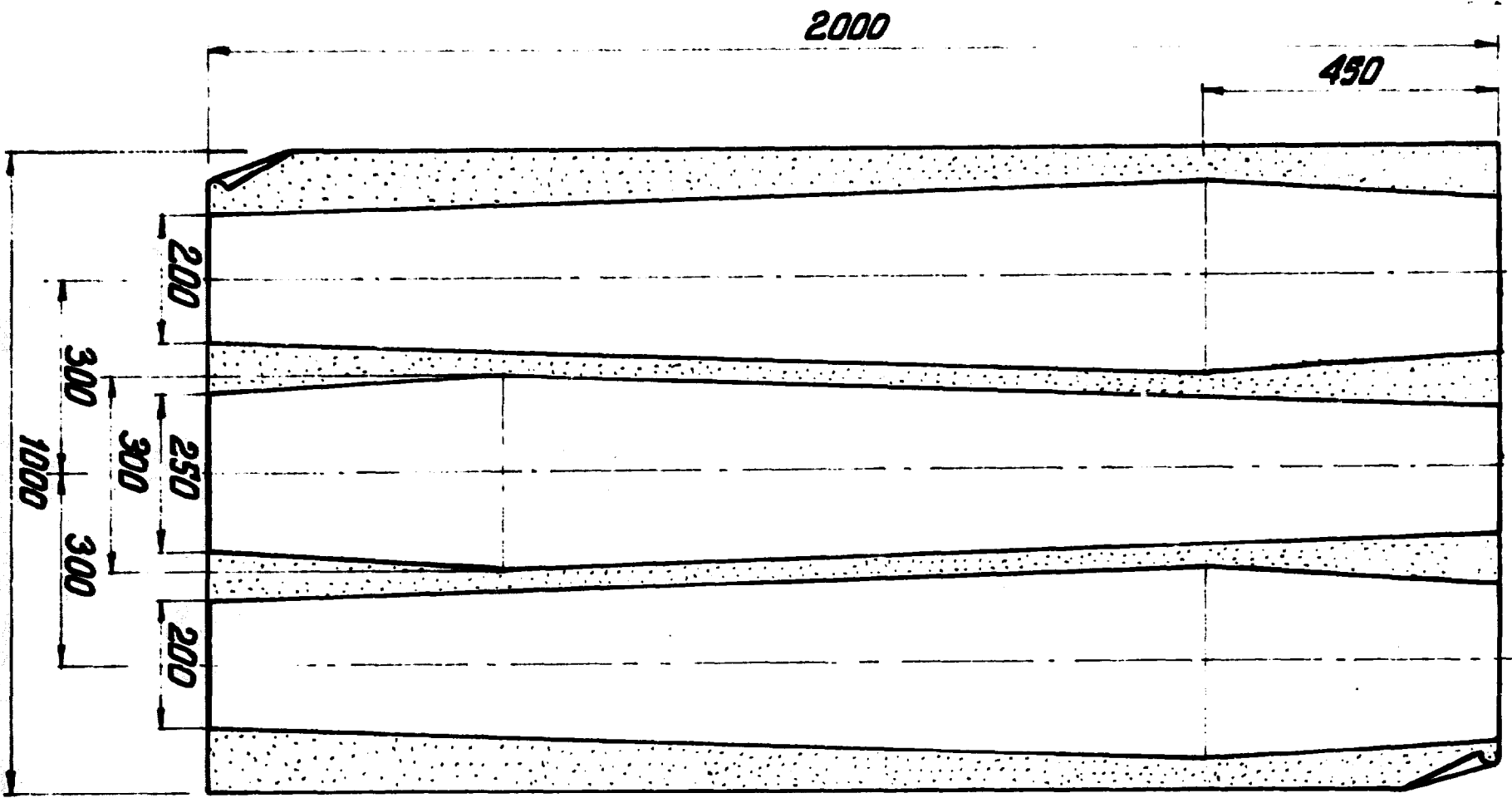
uneconomical (two blades per sheet). It was decided to narrow
the blades by applying the properties of the 10% airfoils
which resulted in a new blade-shape: three blades can be cut
from one standardsheet.
Unfortunately the starting·torque ·coefficient of the rotor
decreased from 0.19 to 0.14 but the advantage of costprice-
reduction made this quite acceptable.
It was also proposed to cut the blade tips in a circular
shape which might increase the efficiency by appr. 2.5 %.
Although wind tunnel tests on this topic did not show
any significant improvement. Some of the 12 PU 500 –
windmills have been fitted with blades of circularly
shaped blade tips and are accepted (more or less) for
aesthetic reasons.

2.2.5. The pistonpump.

From the beginning the major part of the time and energy
spent in developing the windmill-installation has been
required for the evolution of the pumpsection.
Many problems had to be solved concerning the fixation of
the pumps and delivery-pipes into the wells and also the
design of the pistonpump changed from time to time.

Since the windmill drives the pistonpump with R.P.M-
numbers of 15 up to 120 the application of air-chambers
proved to be necessary in order to decrease the acceleration-
forces acting in the pumprod, pump suspension and trans-
mission. These forces were always underrated and several
times cracks of the pumprod-connections and loose pump
suspensions resulted. In some cases even the entire tubewell
was pulled out and led to damage of the pump and pumprod
as well.

The pump rod and pump rod connections

As mentioned before the first windmills have been fitted
with wooden pump rods which proved not to be successful.

The wooden rods, if available, do not exceed the length of appr.
2 m and they are always unstraigth. Since the length of a total
pumprod is at least 10 meters many connections have to be applied.
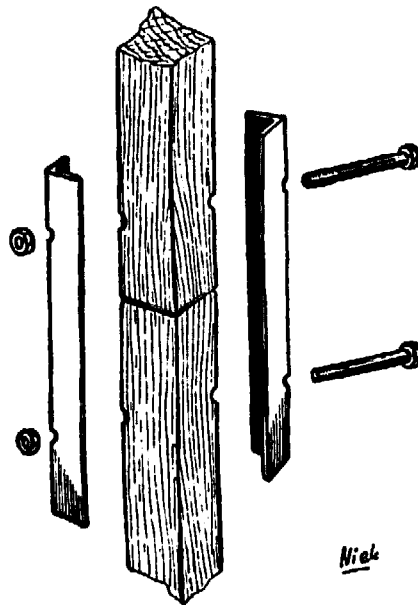
These connections are realized by using simple clamping
devices of angle iron. These seemed to be very strong but
sooner or later, due to the reverse of pumprod forces, these
connection got loose. Actually these failures resulted in
applying steel-pipe as a pumprod. In order to minimise the total
weight the first steel-pumprod has been composed of 1" pipe with
1 mm. thickness only, but still the problem of simple joints
persisted.

A simple rod connection, made by two bolts through it, did
not serve the purpose. Such joints also got loose, worn
out and finally cracked. Flanges made of flats which have
been welded to the pump rod ends offered a strong and reli-
able joint after bolting them.

The disadvantages of such connections were that the pump
rod sections were difficult to align and moreover the
assembling of the pump-rod guides became complicated. The
latter had to be partable. Moreover the thin walled pump
rods needed many guides in order to prevent collapse during
downward motions. Therefore normal 3/4" gaspipe has been
applied for all the pumprods and the introduction of a new
and simple joint solved all the pump-rod troubles.

3/4" screw-threaded rods are hammered into the 3/4" pump rod
and welded to it. Two or three nuts provide a simple threaded-
bush in which the threaded ends of the pump rods,with a spring-
washer in between ,are locked by an extra nut.
These pump rods and its joints serve the purpose very satisfac-
torily and no complications, concerning alignment, cracks or
looseness, appeared anymore.

PUMPROD-
CONNECTION
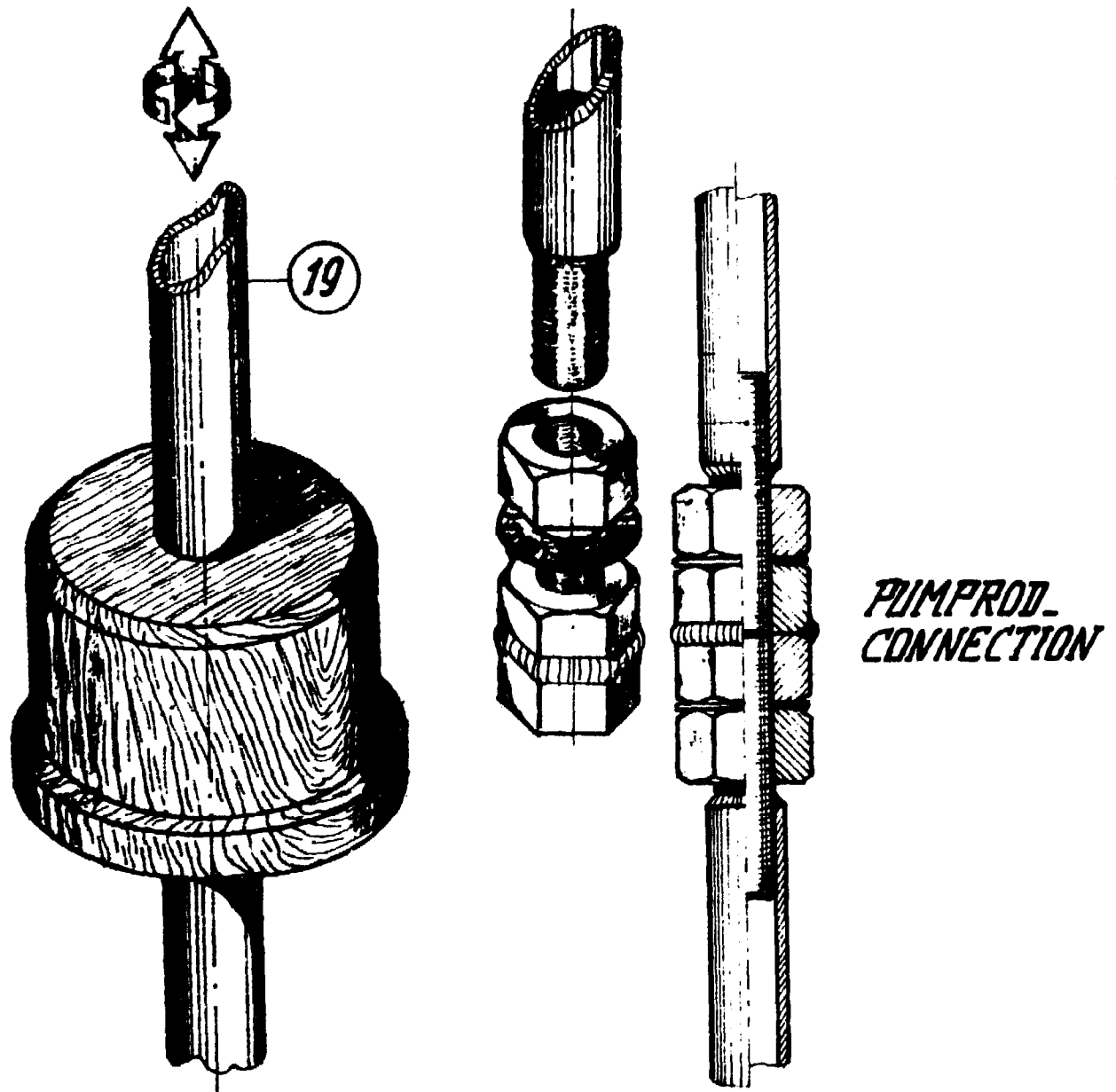
The first prototype-pistonpump

The first prototype existed of a set of P.V.C. pipes of 5"
dia in which the wooden footvalve and wooden piston with
leather sealing were situated near the watertable.

Besides the developing of the pumprod and its connections the
piston became the bottle-neck when the stronger pumprods were
introduced. Since the piston is made out of wood its strength
was improved by increasing the thickness from 1.5" up to 3".
The leather piston sealing proved to be very suitable as far
as wearing and tigthening were concerned, but due to the
high friction coefficient of the leather/P.V.C.-combination
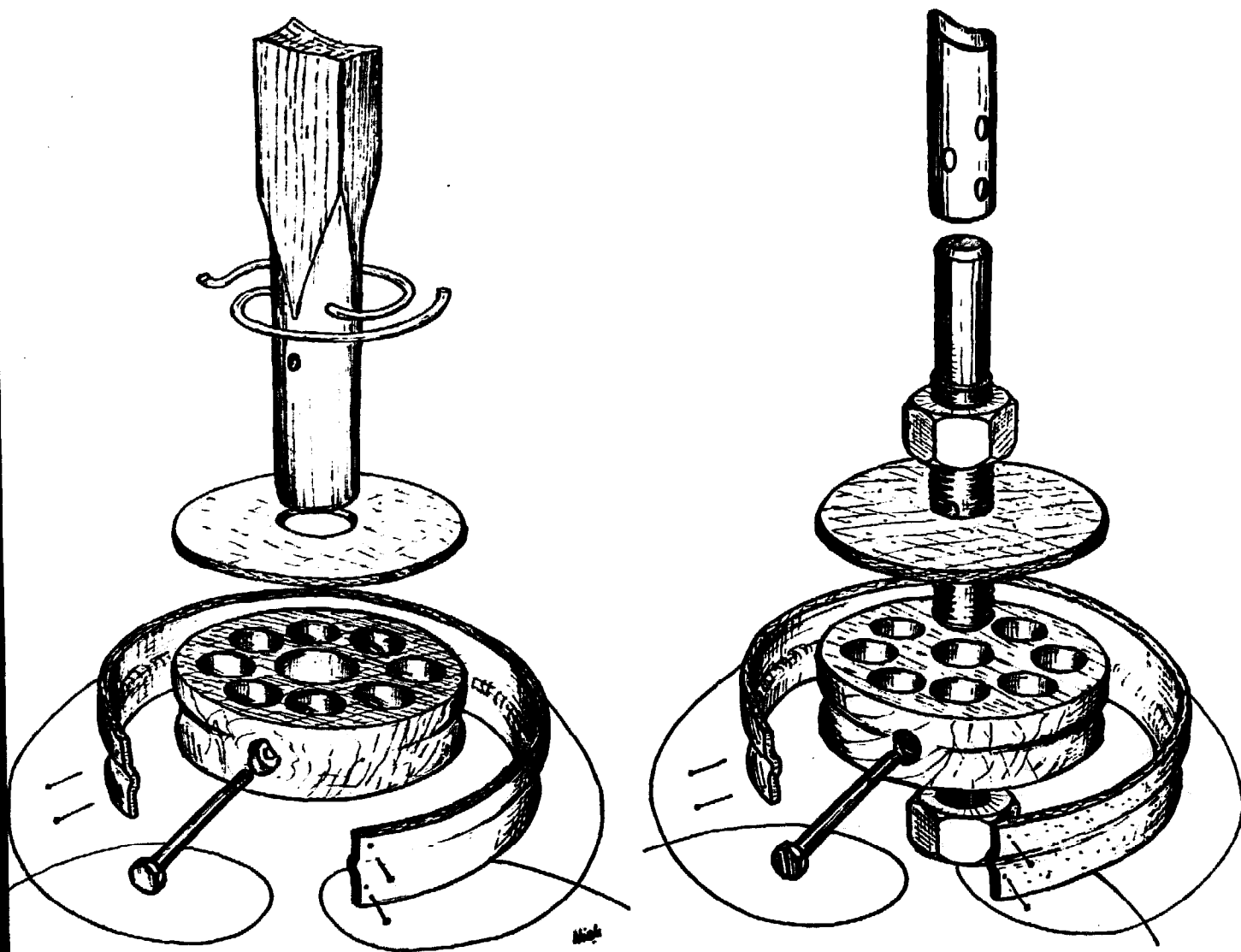
and the high static pressure the leather sealing set up and
caused a sticking slide of the piston in the cylinder resulting
in a very high startingtorque. First a rather high windgust
(upto 6 or 7 m/sec) was required to start the windmill while
the minimum runningspeed amounted to 3 m/sec. The idea was
born to apply a small but permitted leakage in the piston in

such a way that the windmill can slowly move the piston in the top-position and is able to start running unloaded during the first half revolution. Probably the stored kinetic energy in the rotor might maintain the rotation so the starting-windspeed can be reduced.

During these experiments the inducted friction due to the set up of the leather sealing still persisted and just a small improvement has been noted.
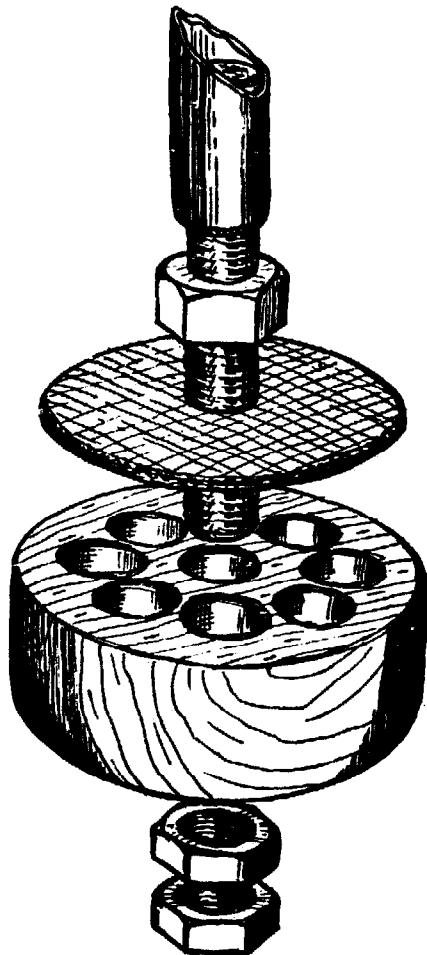
A new experiment with a <u>sealingless</u> piston showed a considerable improvement due to the absence of the Coulomb's friction acting on the P.V.C.-leather-surface. In spite of the fact that the volumetric efficiency is badly affected at very low R.P.M.-numbers this efficiency seemed not to be influenced if higher R.P.M.-numbers are actual. Although the expected lifetime was quite uncertain concerning the increasing of leakage (decreasing of volumetric efficiency) sooner or later this sealing-less piston has been fitted in the existing prototypes. The experience with the sealingless pistons in practice is rather positive: depending on the sliding-fit at the beginning, the purity of the lifted water and the actu-

A

B

SUCTION.
AIR.
CHAMBERS.
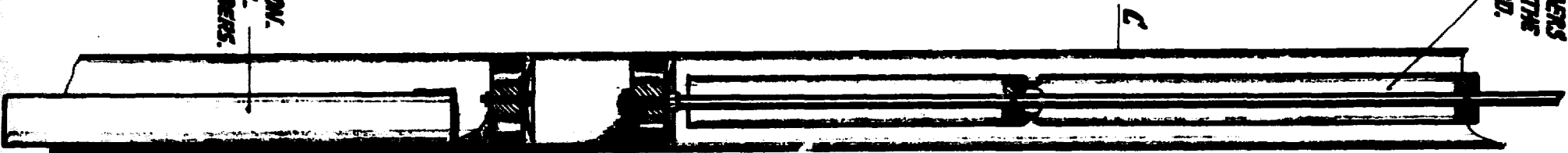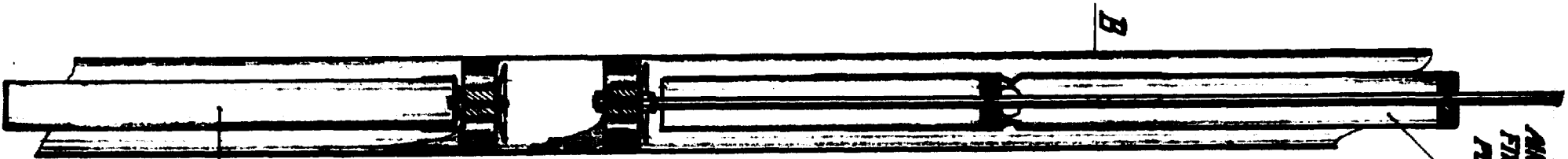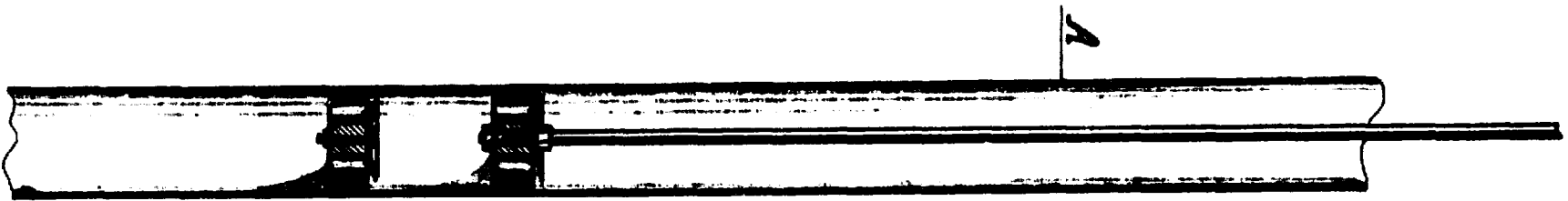
AIR CHAMBERS
FIXED TO THE
PUMPROD.

C

D

SUCTION AIRCHAMBER

PRESSURE AIRCHAMBER

H

1.2D
D
0.6D
0.2D
20
12

$D \triangleq 150$

TO COUNTERSINK
M12 NUT

25
25
75
25
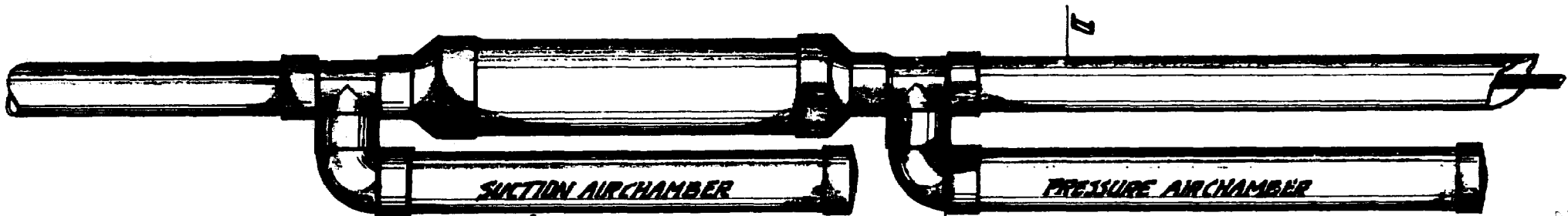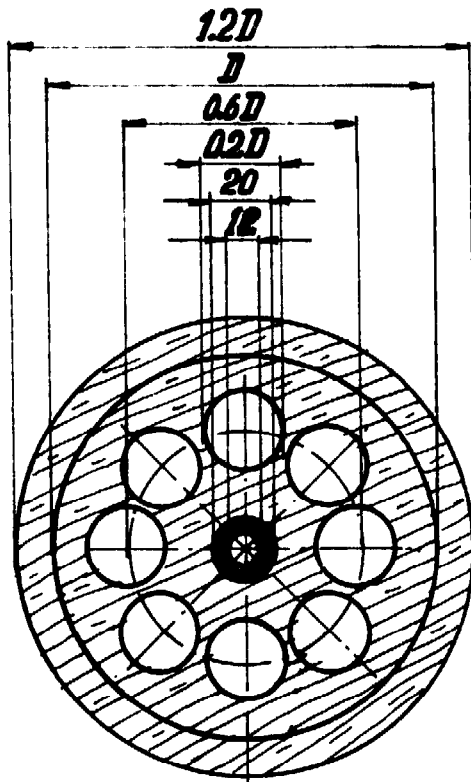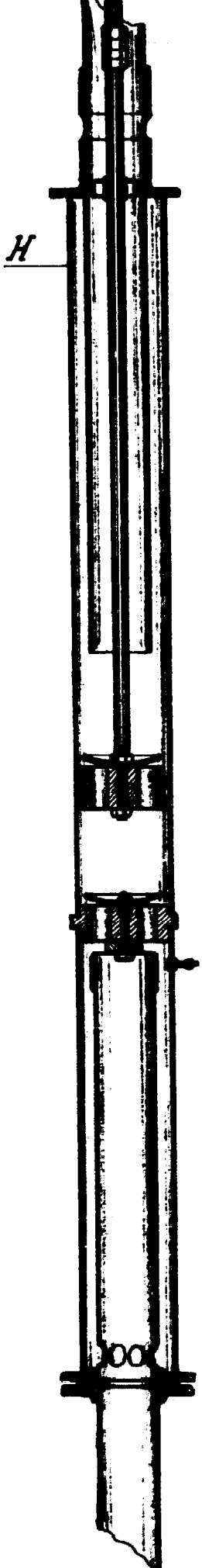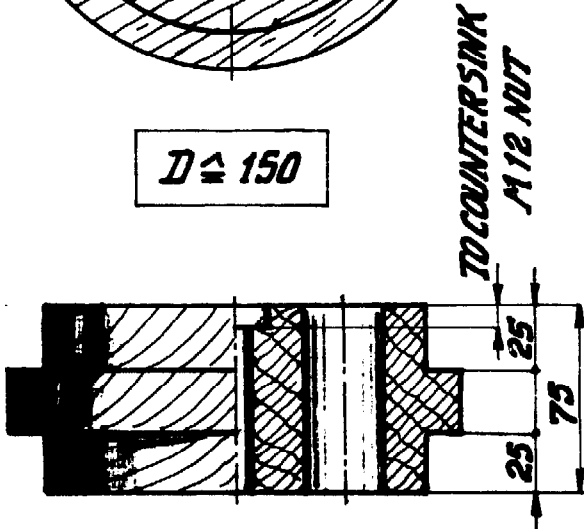
MEASUREMENTS IN WET CONDITION!
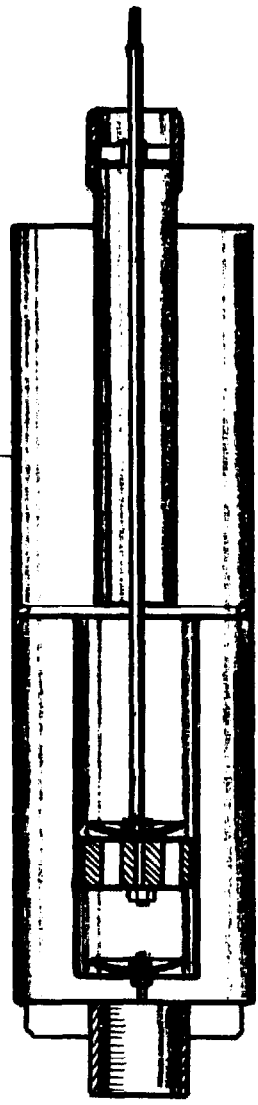
al accuracy of the pump-rod guidance (!) the lifetime
varies considerably.

A worn-out piston/cylinder-combination still operates,but the
efficiency becomes unacceptably low. Therefore maintenance
and repair must be carried out,and if possible in a quick
and uncomplicated way.

Since the leather valves need inspection and repair too the
above mentioned demands had to be taken into account with
regard to the pump designs. Finally the required airchambers
and the materials available determinated this pump design.
Right from the beginning the development of the piston pump
reached many stages resulting in type (A upto J) and still
the way of developing by trial- and error detected the
weaker sections in these designs. Time, wind, patience,
optimism and invention in particular proved to be the most
required tools in order to develop the piston-pump step by
step. Briefly types A up to J will be described.

A   Basic type; very simple and reliable (handpumps) as
            far as no high frequencies actuate!

B and C   The need of air chambers were felt. Forces acting on
            the internal chambers were underrated. Complete
            chamber sections got loose and caused damage to
            foot-valve and piston. Especially the airtightness
            of the pressure chambers was difficult to achieve.

D   When the P.V.C.-sockets, elbows and T-connections
            were available, external air chambers have been
            composed and applied.   But action-reaction effects
            in the external pipes caused vibrations while the
            presence of air in the pressure section remained
            doubtful after some time of operation. (Air solves
            into the water under high pressure conditions.)

E   Suitable for moderate elevation head (no pressure
            chamber). In this stage 7/8" pump rod and 7/8"
            connection have been introduced.

F   Cheaper and simpler construction; by applying an internal suction-air chamber several reducing sockets, T-connection and elbow were superfluous. Total section is clamped between two flanges.

G   Similar type but suitable for <u>higher</u> elevation head. Air in pressure chamber is maintained due to small but permitted leakage in suction section (by bicycle tube-valve). Due to the location of the suction-air chamber a foot valve with <u>central</u> supply has been applied. The latter did not operate satisfactorily in combination with the sole-leather valve disk which deformed in such a way that leakage resulted. The pump is simple to open and to inspect.

H   The "combination pump" combines the advantages of pumptypes F and G resulting in the following properties of the pump:
- double air-chambers (reduces pump-rod forces)
- reliable valves (self-priming)
- proper pumprod guidance (life-time piston)
- "automatic air injector"(pressure chamber remains full)
- total pump sections can be replaced easily (quicker repair, maintenance)
- pump is opened quickly (repair, maintenance is normally carried out in the workshop).

I   This pump is completely composed of steel components which are jointed by weldings. The pump is easy to open in order to carry out maintenance and repairs. The cylinde is made of gaspipe from which the inner seam is removed by means of a simple cutting device and final finishing is done by honing. Though its construction is more advanced than those of the other pumps its reliability proved to be better. (See chapter 4).

J. Same type but the footvalve is made of sheet iron. This
allows a shorter cylinder which facilitates removing the
seam. Moreover no doubts remain whether leakage occurs
between valve-body and cylinder as sometimes seemed in
the foregoing pumptype. Also this pump has been tested
thoroughly as described in chapter 4.

The air-injection system in both the pumptypes I and J has
been abandoned. as pumptests showed (by means of waterlevel
gauge) that the air in the pressure vessel still remained.
Results of thorough tests of the pumps latter mentioned
are described in chapter 4.

2.2.6 Pump suspension

Practice showed that fixation of the windmill pump can be
rather complicated. Especially during stormy weather consi-
derable forces are acting on all moving parts and the pump
structure in particular. One can imagine what happens if
the suspension shows some play. In some cases the entire
tube-well got loose and was simply pulled up till the
piston started hammering on the foot valve!

Besides the need of an adequate fixation in vertical sense
supports are also necessary to keep the long pipe lines
centered. Previously this was achieved by means of wooden
crosses which have been wedged in the well by means of keys.
Though such arrangements served the purpose some danger
exists as people tend to stand on the crosses during climbing
into the well. A simple tripod-"umbrella", combined with the
pipe-line socket, proved to be very useful . Centring is done
simply by adjusting the "pods" and locking them in the brick
wall by means of a few hammer blows .

Proper fixation of the entire pump sections have been achieved
by applying the following constructions:
- Anchors masoned in the well
- Expanding clamping-devices
- Tower-pipe connection
- "Bridge"

Anchors masoned in the well

In case of an excentrically placed tower, which is often done
in order to facilitate waterlifting by means of other devices,
the well-structure itself provides attachments for the pump
section. Anchors are masoned into the wall so that the delivery
pipe-line can be fitted.

Expanding clamping device

Provided that the well
casing is of proper con-
struction the entire
pump arrangement can
be fixed suitably by
means of an expanding
clamping device. The
latter is composed of
1"gas pipes which are
welded at right angle
either to the bottom
section of the pump or to
the suction pipe. Mutual
braces (angle iron)
ensure a rigid construc-
tion. The clamping arrange-
ment is obtained by fitting
longthreaded bolts or screw
studs into the extruding
pipe ends. Tightening their
nuts causes the expansion
and fixation of the frame.
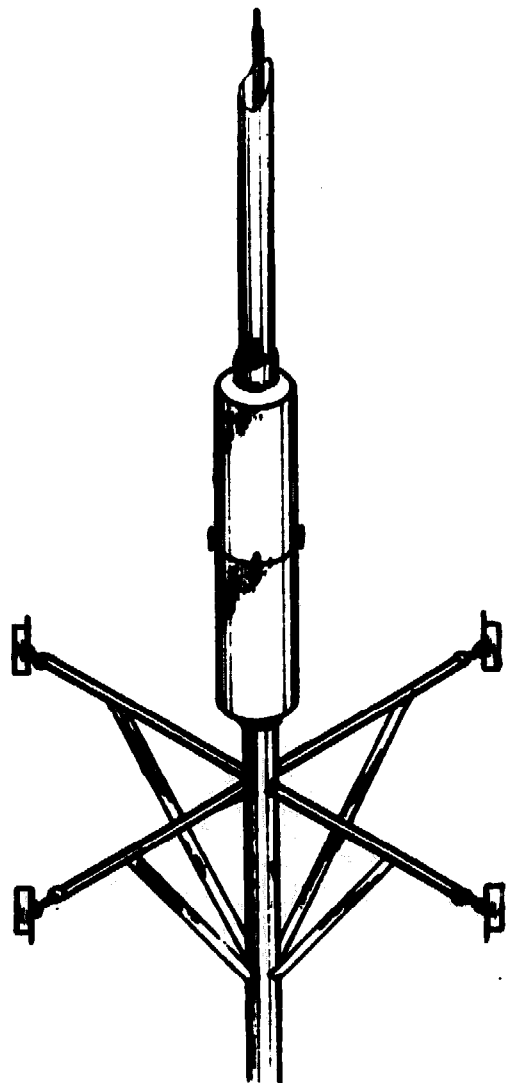
In order to prevent damage
to the bricks of the well
casing it is necessary to
enlarge the bearing surface
by means of pieces of angle
iron.

CLAMP ARRANGEMENT
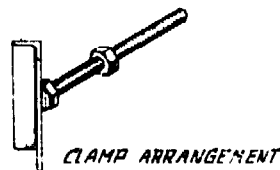
Such clamping device proved its
worth in combination with the
"bridge" construction.

Tower pipe connection

In cases of weak or brittle
well casings the pump rod
forces are hardly carried
by masoned anchors and other
devices to such well-casings.
Therefore it is preferable to
carry over the forces direct-
ly via a tower-pipe extension,
which is composed of two long
angle irons and some clamps.
As a matter of fact the entire
pump section hangs from the
tower pipe and all acting for-
ces are led through this sus-
pension instead of through
tower structure, fundation,
well casing and the pump
suspension respectively.
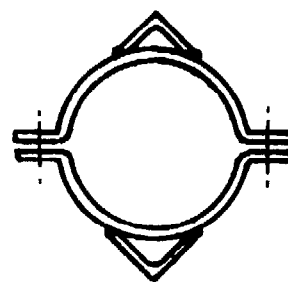
Sideward vibration hardly
occur and if so,retaining of
the pump section is simply
achieved by means of applying
the "tripod umbrella" (as men-
tioned earlier), masoned anchors
or the "bridge".

"Bridge"-suspension

If a strong and proper well casing is available the simple bridge construction provides a rigid suspension of the pump section in horizontal and vertical sense both.

The delivery pipe is clamped to the bridge by means of two clamps. By opening these clamps, and the pumprod-connection, the entire topsection of the pump (including delivery line) can be lifted in order to facilitate repair and maintenance the moving parts of the pump.
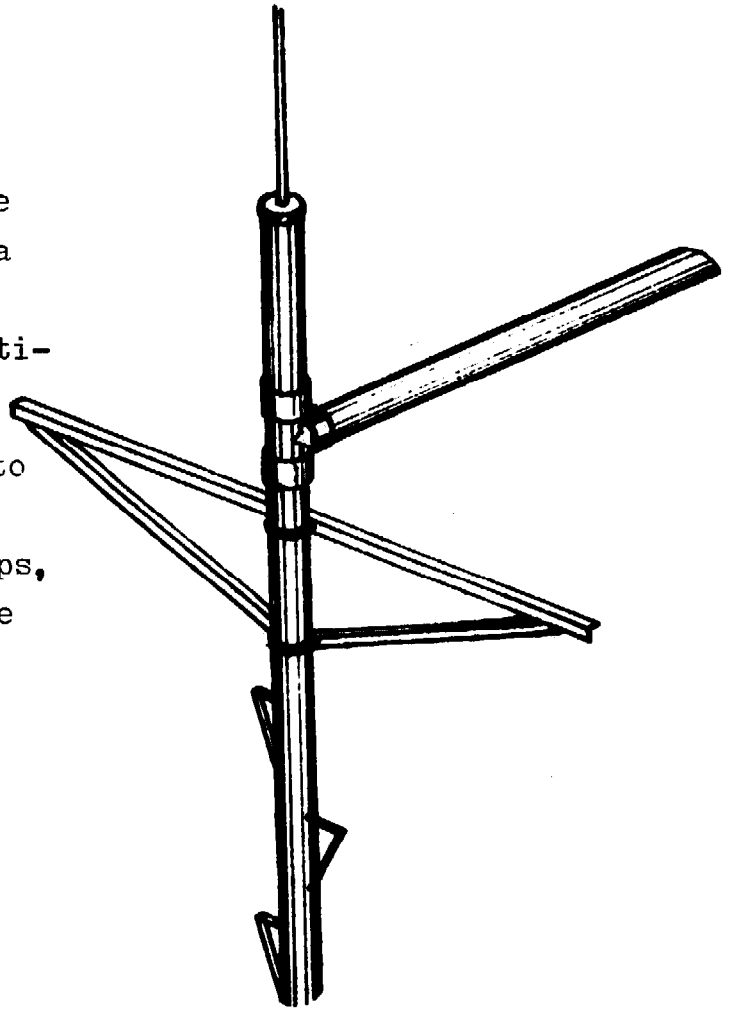
2.3. The 4-bladed 2.50 m. dia. windmill for low-water-lifting.

In winterseason surface water, which remained from the rain-
season, is used in some cases for the irrigation of the winter-
crops. Several small farmers lift this surface water by
manpower to their fields; the idea was born to develop a small
and cheap windmill which can do this job.
Although this low-waterlifting was not planned in the project-
schedule and its need was not actual, some time has been spent
by the Dutch windmillengineers in developing this windmill.
Therefore some attention is paid to this subject.
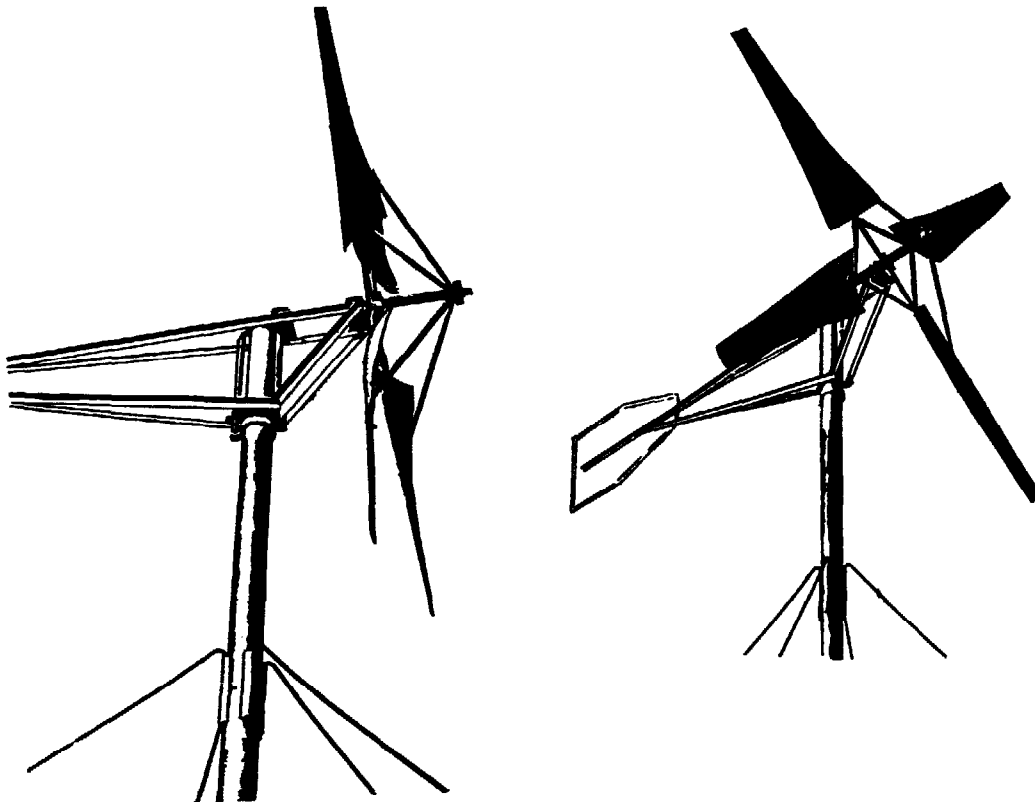
Actually it was the size of the scrap of steelsheet, which
was left from the production of the big windmills which
determined the dimensions of this small windmill. The first
prototype has been fitted with a 4" pistonpump combined
with a 3" towerpipe. The total structure stands on its pump-
section and is held in position by guys. The latter are
simply made of 3/8" steel rods and anchored by pegs to the
ground. The conversion of the millshaft-rotation into the
up-and downward -motion of the pumprod is obtained by means
of a crank-pumprod-mechanism. The elasticity of the rod allows
its outward bending besides the up-and down-movement so a

connecting-rod and crosshead are superfluous. Two wooden
guides, situated in the tower, provide a further support to
the 6 m.-long pumprod. The piston is made sealingless in
order to enable the windmill to start almost unloaded. This is
necessary since the starting-torque of the 4-bladed windmill
is very low. The measure of leakage is determined by trial
and error and a clearance of the piston in the cylinder of
appr. 1/2 % of the piston-diameter proved to lead to satis-
factory results. In spite of this leakage the volumetric
efficiencies are reasonable and even the value of 100% is ex-
ceeded due to the inertia-effects in the watercolumn if high
R.P.M.-numbers occur.

In 1978 the total material-costs of the small windmill amounted
to less than Rs. 500,- while the required manpower for
building it is estimated at appr. 50 manhours.

Later on 3 other small windmills have been built and installed.
These were slightly different in size and design (3.2 m. dia.,
tip-speed ratio 5) and consequently more costly.

In the longer run some technical problems (e.g. fatigue-cracks
in the blades) in combination with the low windspeeds made it
clear that the use of these small windmills was not quite
succesful in the Ghazipur district, but on the other hand its
simplicity and costprice caused that it was adopted by the
WOT (TOOL-member in The Netherlands). Developing and testing
there, proved that this small windmill can still be attractive
for low-waterlifting purposes while the basic design is main-
tained and applied in other WOT windmill types.

In the framework of these windmilltypes the small windmill is
coded: 4 PU 250 (4-bladed pumping unit of 250 cm. dia.)
Applying some small modifications the windmill can be fitted
with a transmission to groundlevel by means of a rotating shaft
for multi-purposes (4 MP 250). Also generating electricity
is possible by applying this basic design and a 4 EG 250 will
result.
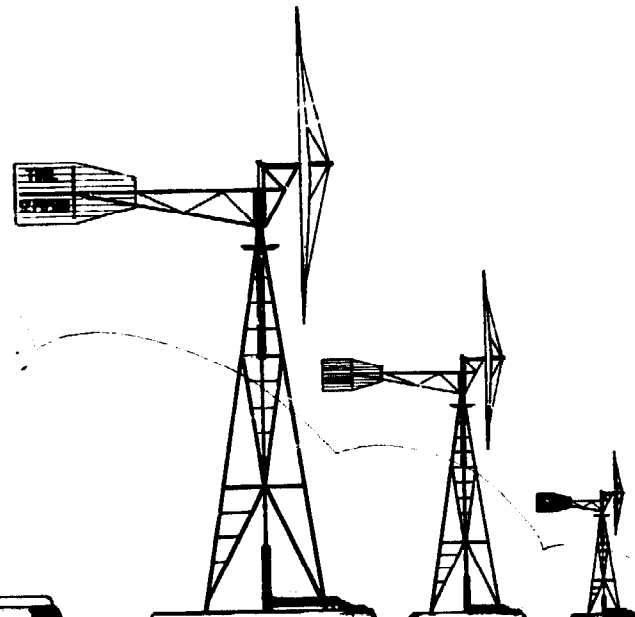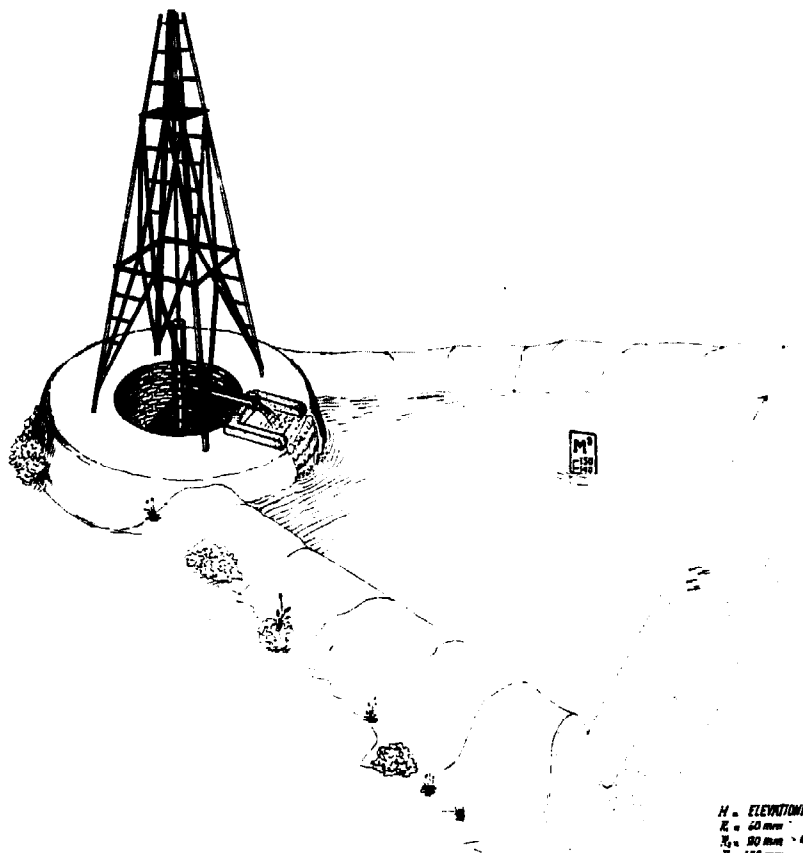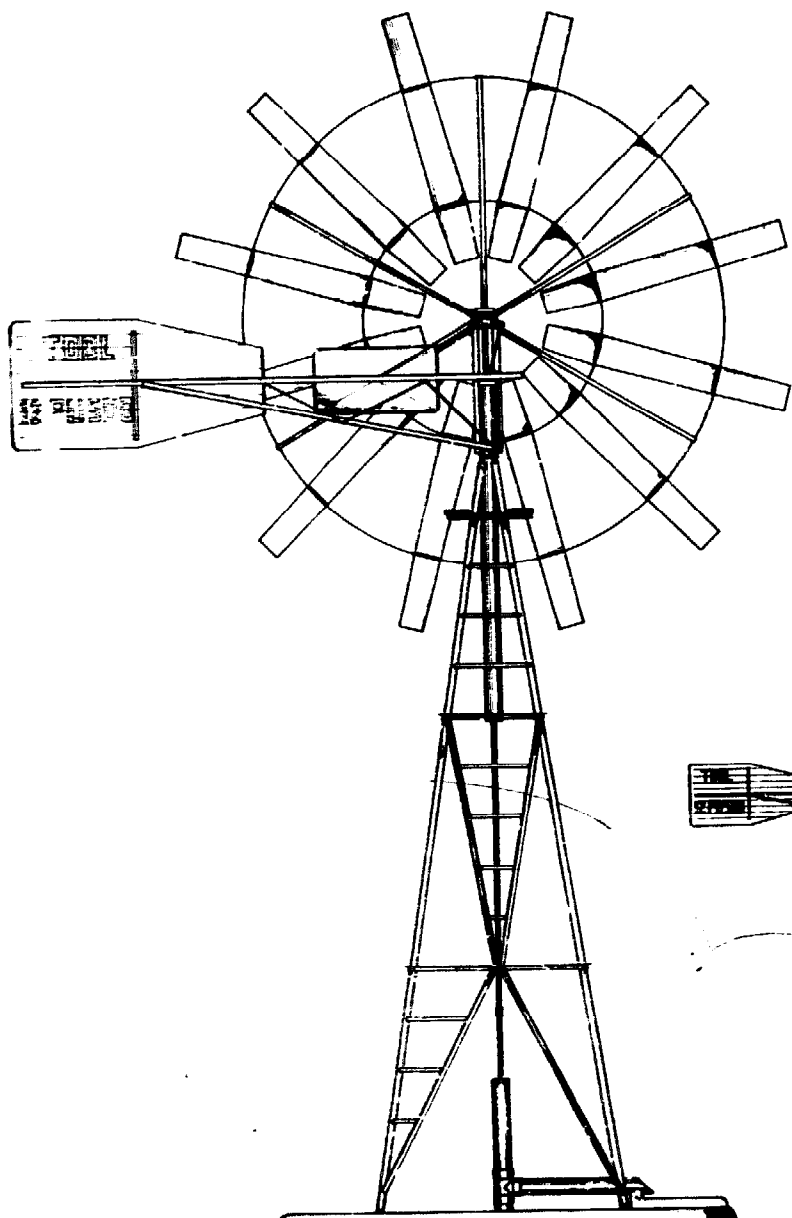
H = ELEVATIONHEAD
E_o = 60 mm
X_o = 90 mm - CRANKRADIUS
R_2 = 125 mm
FOR 150 mm PISTONPUMP

H = 5m

H = 10 m

H = 15 m

H = 20 m

WINDSPEED

1/sec

m/s

$\frac{1}{7}$ OF 7
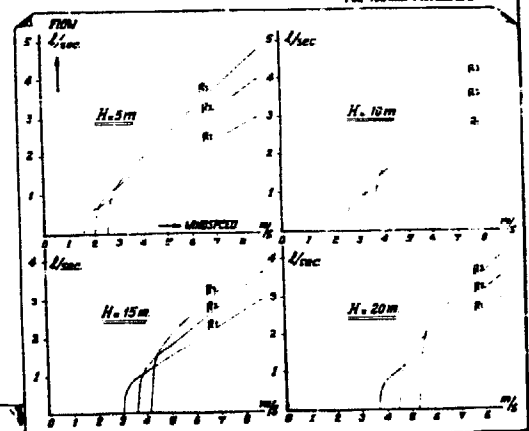
12 PU500 WINDMILL FOR IRRIGATION

FOR INFORMATION:

Workgroup on Development Techniques
T.H. Twente - Vrijhof 152
P.O. Box 217 Enschede
The Netherlands

WALLTHICKNESS TOWERPIPE NOT LESS THAN 3.5 MM

TOWERPIPE. TOWERLEG.
CONNECTION BY
GENEROUS WELDINGS

SCREW
LOCKING
DEVICE
(SEE DRAWING)
N° 4

TOWERPIPE.
SECTION

SCREW LOCKING
DEVICE FOR PART
N° 27 IN DRAWING
N° 4

OUTSIDE DIA. TOWERPIPE

HOLES ARE DRILLED
BEFORE WELDING
⑤ TO TOWERHALF

650

350

PLASTERED

THIS SIDE IS
PLASTERED

FIRST TWO OF THESE TOWER HALVES
ARE PREFABRICATED
BOTH HALVES ARE COMPOSED AND
THE TOWERPIPE. SECTION IS SHIFTED
IN AND A COMPLETE TOWER RESULTS

IF DESIRED THE LATTICE MEMBERS CAN BE POSITIONED
AND WELDED INSIDE THE TOWER LEGS RESULTING IN
A MORE ATTRACTIVE APPEARANCE OF THE TOWER.
HOWEVER THE FLANGES AT THE ENDS OF THE ANGLE.
IRON MEMBERS N° 7 AND N° 9 SHOULD BE CUT AT
ANGLES OF 45 DEGREES.

N°	NUMBER	MATERIAL/NAME/REMARKS/MEASUREMENTS		
21	2	FLAT. IRON	30x6 (1¼"x¼") x	220
20	2		— x	360
19	2		— x	500
18	2		— x	570
17	2		— x	440
16	2		— x	370
15	2		— x	100
14	8		x	SCRAP
13	4		x	430
12	4	BOLT. NUT. WASHERS	M8 (W 5/16")	
11	2	PLANK	50x200x700	SCRAP
10	4	ANGLE. IRON	30x30x3(1¼"x1¼")	SCRAP
9	4		x	1950
8	16		x	1960
7	2	FLAT. IRON	30x6 (1¼"x¼") x	700
6	2		— x	570
5	2		— x	440
4	2		— x	340
3	2	ANGLE. IRON	30x30x3(1¼"x1¼") x	660
2	4		40x40x4(1½"x1½") x	6500
1		TOWERPIPE	4" GASPIPE	2650

12 PU 500 - TOWERCONSTRUCTION

2
OF
7

FOR
INFORMATION:

Workgroup on Developmenttechniques
T.H. Twente - Wijfied 152
P.O. Box 217 Enschede
The Netherlands

BRACKET PREVENTS THAT BEARINGHOUSE CRACKS OPEN!

SPOTWELDINGS

ROTOR WEIGTH

THRUST

CLEARANCE THIS SIDE!

CLEARANCE THIS SIDE!

CRANK FORCES

26 SLIDES IN CRANCK

SCRAP

WELDING MANOEVER

FITS SLIDELY INTO P.V.C. INSERT

R=60
R=90
R=120

165
150
90
60
30

GENEROUS WELDINGS!

RECESS OF 0.2MM

PUMPROD FITS HEAVILY

SLIDES EASILY INTO BOTTOM END OF TOWERPIPE

PUMPROD SLIDES EASILY

MEASUREMENT TO PUMPROD END 670

TOWERPIPE PVC INSERT

PUMPROD CONNECTION SEE DRAWING N°6

N°	NUMBER	MATERIAL / NAME / REMARKS / MEASUREMENTS	
27	1	PUMPROD GUIDE	HARDWOOD SOAKED IN OIL
26	1	NUT	M20 (W 3/4)
25	2	BOLT. NUT. WASHERS	M12x80 (W1/2"x3")
24	4		M12x50 (W1/2"x2")
23	2		M10x100 (W3/8"x4")
22	2		M6x25 (W1/4"x1")
21	1	THREAD STUD	M24x100 (W7/8"x4")
20	1	PVC INSERT	Ø100x700
19	1	PUMPROD	GASPIPE Ø3/4"x4700
18	1	CROSS HEAD	HARDWOOD SOAKED IN OIL
17	1	FLAT IRON	30x6 (1 1/4"x1/4") LENGTH 165
16	1		220
15	1		140
14	1		650
13	4		50
12	2		147
11	2		130
10	1	CROSPIN Ø25x92 REMOVE SHARP EDGES	
9	1	CRANKBUSH	
8	2	BOLT	M20x70 (W3/4"x2 1/2")
7	2	WASHER	FOR 8
6	4	NYLON BEARING BUSH	Ø1 1/2" NYLONROD
5	2	BEARING HOUSING	
4	2	BEARING BUSH (BEST QUALITY)	
3	2	BALL BEARING (BEST QUALITY)	
2	2	BEARING HOUSING	
1	1	SHAFT	Ø50 (Ø2")x1300

4 OF 7   MOVING PARTS OF -12PU500-WINDMILL

HAMMERING PROFILES INTO THE VANE SHEET INCREASES RIGIDITY

AFTER PROFILING THE EDGES ARE BENT OVER

TOOL

GENEROUS WELDING 60 DEG.

PROPER ALIGNEMENT OF THE BEARING SUPPORTS (1) !

WEARING FLAT

WELDED TO 6

WELDED TO 6

WELDED TO T4

WELDED TO NUT

WELDED TO NUT

600

LOCK MECHANISM KEEPS WINDMILL IN SECURED POSITION (75 DEG. OUT OF THE WIND)

NO CLEARANCE !

HELPVANE SHEET IS PROFILED DIAGONALWISE AND EDGES ARE BENT OVER 45 DEG.
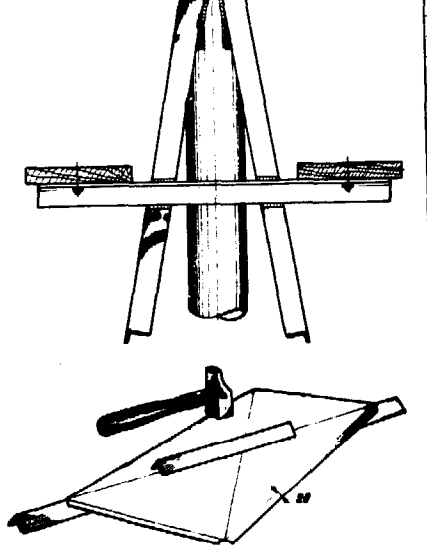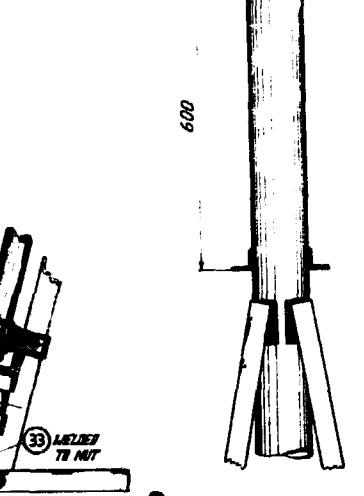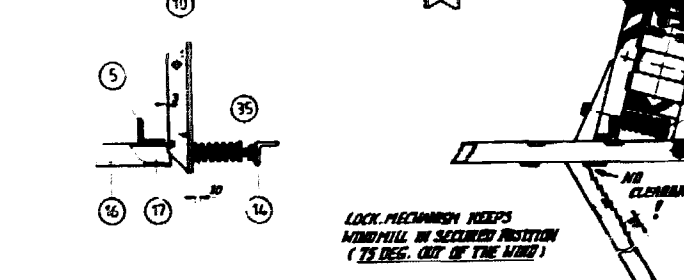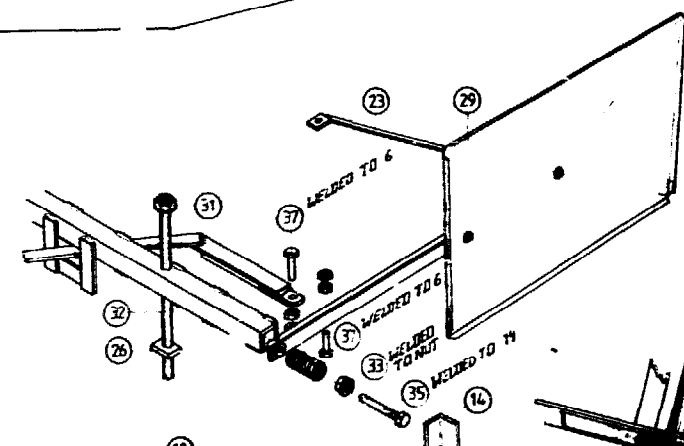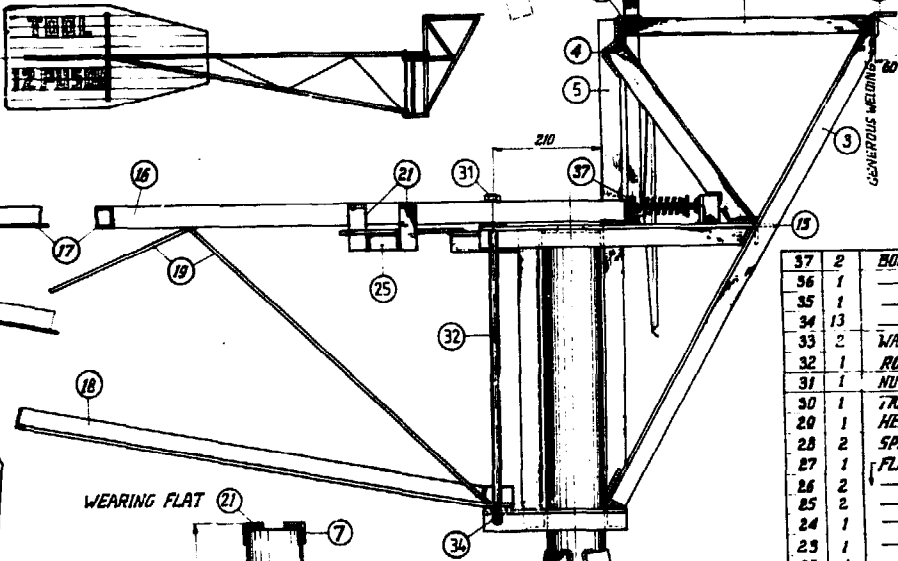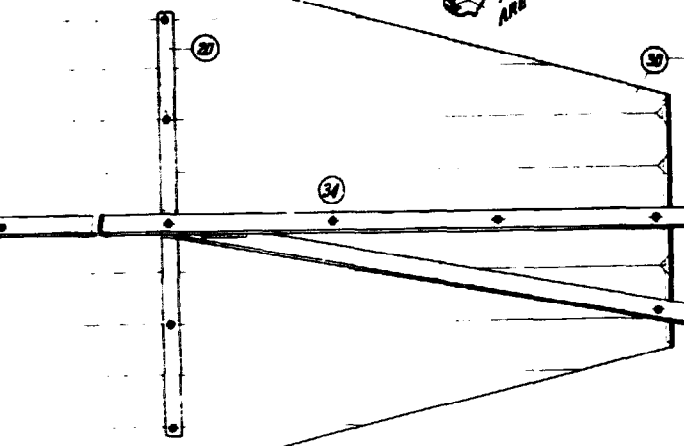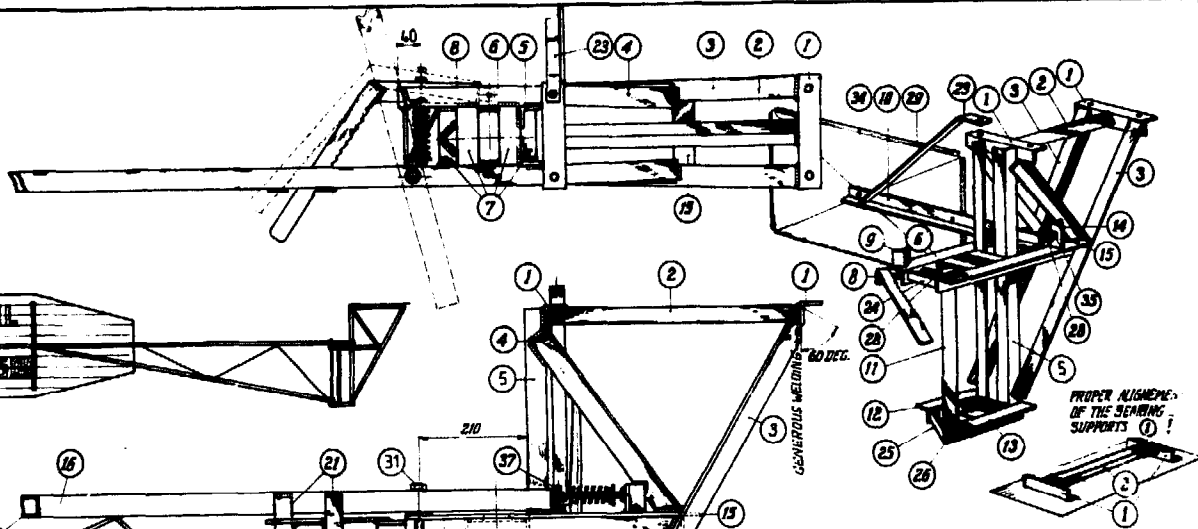
№	NUM BER	MATERIAL/NAME/REMARKS/MEASUREMENTS		
37	2	BOLT_NUT_WASHER	M 12 x 40	(W ½" x 1½")
36	1		M 10 x 150	(W ⅜" x 6")
35	1		M 12 x 100	(W ½" x 4")
34	13		M 6 x 25	(W ¼" x 1")
33	2	WASHER	⌀ 1"	(⌀ ½")
32	1	ROD	Ø 18 x 650	(Ø ¾" x 2'2")
31	1	NUT	M 20	(W ¾")
30	1	TAILVANE	1 mm SHEET	2000 x 1000
29	1	HELPVANE		1800 x 500
28	2	SPRING	APPR. Ø 30 x 100	16 4
27	1	FLAT IRON 30 x 6 (1¼" x ¼")	LENGTH 100	
26	2			50
25	2			130
24	1			115
23	1			900
22	1			360
21	4			90
20	1			830
19	3			750
18	1	ANGLE IRON 40 x 40 x 4 (1½" x 1½")	LENGTH 3000	
17	1			4000
16	1			1500
15	1			120
14	1			60
13	1			280
12	1			200
11	1			590
10	1			1000
9	1			45
8	1			180
7	4		TOWER PIPE DIA.	
6	2			54"
5	2			1000
4	2			460
3	2			1080
2	1			500
1	2	90 x 90 x 5 (2" x 2")		240

HEADCONSTRUCTION - 12.PU500 - WINDMILL

3 OF 7

2.47

35 CLAMP FITS OVER 4" PIPE AND DETERMINES PITCH OF HOLES IN ANGLES 21

TIGHT FIT IN 27
SLIDES OVER PUMP ROD

EXACT FIT IN CYLINDER IN SOAKED CONDITION
93 (062D)

28 30

18 21

PUMP ROD GUIDE WELDED IN SOCKET 8
27 20

1 30 31 32 PUMPROD CONNECTION

26

2

DETERMINATE !

BRICK MASONED WELL

30
2
8

AIRTIGHT WELDINGS !

30
2
8
11

15
9
13

13
23

RING WELDED IN CASING PROVIDES SUFFICIENT SURFACE FOR GASKET

PRESSURE AIR CHAMBER

36
4
7 5
21
35

19

8 PUMPROD GUIDE IN SOCKET

6

23
22
33
3

16
10
24
26

SUCTION AIR CHAMBER

13

24
34

15

8

6

PRESSURE SECTION

SUCTION SECTION

8 PUMPROD GUIDE IN SOCKET

22
3
←CUTTING AFTER WELDING TO 13

24
26
25
33

12
16
10

13
24
14
34
17
11
8

AIRTIGHT WELDINGS

15
11
8

BOTTOM VIEW

∗ AMOUNT OR SIZE DEPEND ON CIRCUMSTANCES

Nº	NUMBER	MATERIAL / NAME / REMARKS / MEASUREMENTS		
36	1	WOODEN PUMPROD GUIDE : 27 IN DRAWING Nº		
35	2	CLAMP	ROD Φ 12 (½")	×370
34	1	BOLT - NUT - WASHER	M12 × 20 (W½" × ¾")	
33	4	BOLT NUT WASHER	M12 × 125 (W½" × 5")	
32	∗	SPRING WASHERS	M 24 (W⅞")	
31	∗	NUTS	M 24 (W⅞")	
30	∗	THREAD STUD	M 24×100 (W⅞"×4")	
29	1	THREAD STUD	M 24×150 (W⅞" × 5")	
28	∗	GUN METAL BUSH	Φ 33 × 30	
27	1	STEEL BUSH	Φ 43 × 33	
26	1	PISTON	TEAK WOOD	
25	1	GASKET	SOLE LEATHER Φ290 × 6	
24	2	VALVE	Φ 150 × 6	
23	1	RING	ROD Φ 8 (⅜")	×870
22	8	FILLING PIECE		× 40
21	2	ANGLE IRON	40×40×4 (1⅝×1⅝)	∗
20	3	FLAT IRON	30×6 (1⅛ × ¼) ×	∗
19	∗	___		× 300
18	1	___		×450
17	1	___		× 180
16	4	___		× 60
15	12	___		× 85
14	1	FOOTVALVE	2 mm SHEET	× 170
13	2	CASING		500 × 900
12	1	CASING FLANGE		Φ290
11	2	CASING FLANGE		Φ290
10	1	CYLINDER SEAMLESS	6" GASPIPE	× 460
9	1	DELIVERY PIPE	4"	× 560
8	∗	SOCKET	FOR 4"	
7	1	T - SOCKET	4"	
6	∗	DELIVERY PIPE (EXTENSION) 4"	∗	∗
5	1	EXHAUST PIPE	4"	
4	1	TOP PIPE	4"	× 750
3	4	PIPE	½"	× 100
2	1	PUMPROD	¾"	× 950
1	∗	PUMPROD EXTENSION	¾"	× ∗

PISTONPUMP FOR 12 PU500 WINDMILL

6 OF 7

ALT 1-3-1982 W.R.
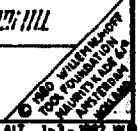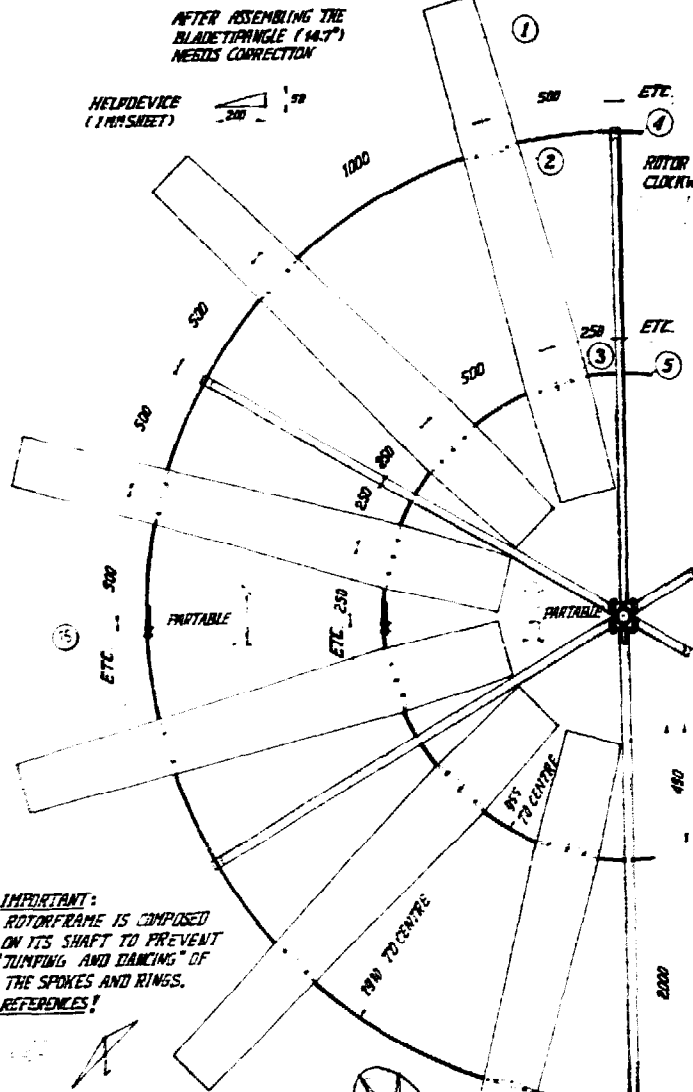
AFTER ASSEMBLING THE
BLADETIPANGLE (14.7°)
NEEDS CORRECTION

HELPDEVICE
(1MM SHEET)

ROTOR RUNS
CLOCKWISE

PARTABLE
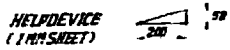
IMPORTANT:
ROTORFRAME IS COMPOSED
ON ITS SHAFT TO PREVENT
"JUMPING AND DANCING" OF
THE SPOKES AND RINGS.
REFERENCES!
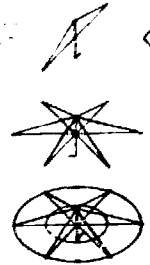
RINGS ARE POSITIONED,
CENTERED AND WELDED.

PARTS Nº 6 ARE
WELDED.

SHAFT AND
TWO ROTOR-
HALVES BELONG
TOGETHER!

SHEET IS ROLLED
IN A BENDING
RADIUS OF APPR.
900 mm
AFTER CUTTING
THE RADIUS IS
CORRECTED SO THAT
10% CURVED AIRFOILS
WILL RESULT.

2MM SHEET

1MM SHEET

R=300

R=300

HOLES 7 mm

RIGID BLADESUPPORT
IS OBTAINED AFTER
HAMMERING OVER THE
CURVED EDGE.

ORIGINAL
POSITION
MARKED
BY
CENTRE
POINTS
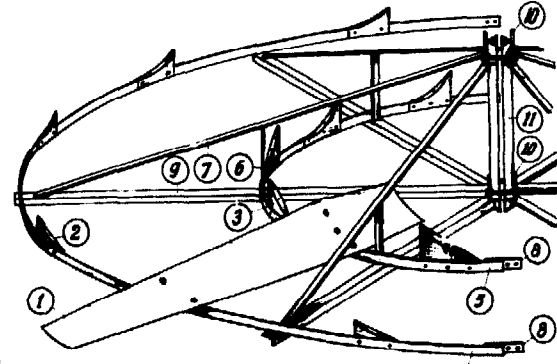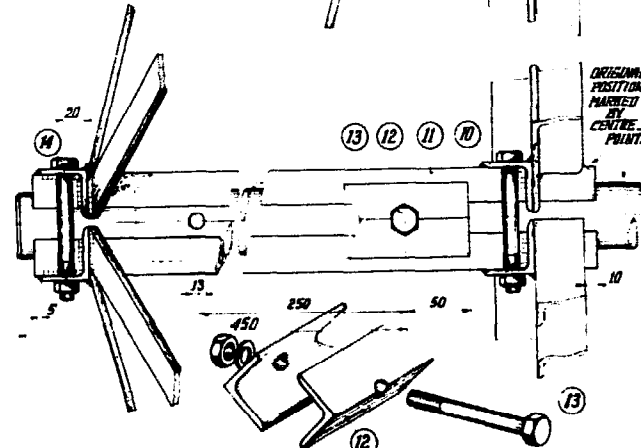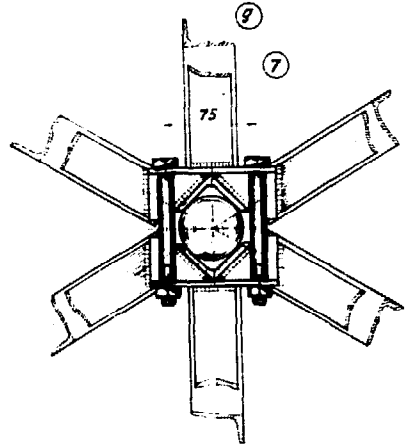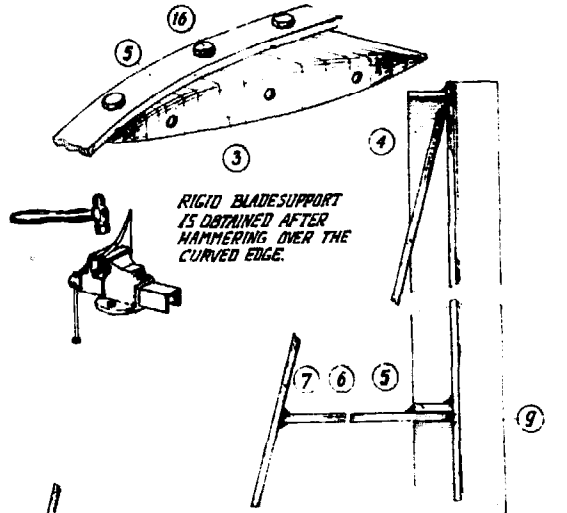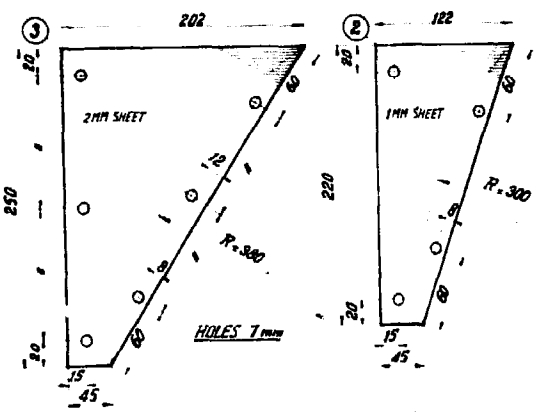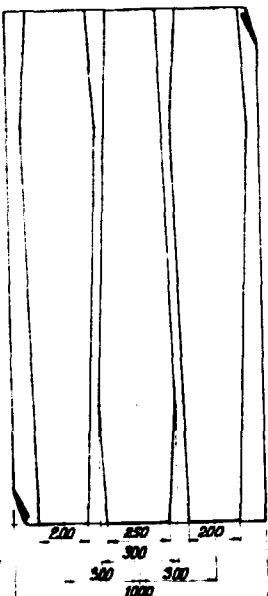
ONE OF THE BLADES IS MOUNTED
ON A WELL POSITIONED SET OF SUPPORTS
(AVERAGE MEASUREMENTS!) AND IS USED
AS A DRILLINGMOULD FOR THE OTHER
11 BLADES.

Nº	NUMBER	MATERIAL/NAME/REMARKS/MEASUREMENTS		
16	120	BOLT. NUT. WASHER	M 6 x 20	(W ¼" x ¾")
15	8		M 10 x 25	(W ⅜" x 1")
14	4		M 12 x 100	(W ½" x 4")
13	2		M 12 x 125	(W ½" x 5")
12	2	ANGLE IRON	50 x 50 x 5 (2" x 2")	x 350
11	2		40 x 40 x 4 (1⅝" x 1⅝")	x 625
10	4			x 100
9	6	FLAT IRON	30 x 6 (1¼" x ¼")	x 1900
8	4			x 120
7	6			x 1900
6	6			x 250
5	2			x 3000
4	2			x 6000
3	12	BLADE SUPPORT	2 mm SHEET	
2	12		1 mm	
1	12	BLADE	4 STANDARD SHEETS 1000 x 2000 x 1	

5 OF 7   ROTOR FOR 12.PU500 WINDMILL (λ=2)

FOR
INFORMATION:

Werkgroep en Ontwikkelingstechnieken
T.H. Twente – Wijkel 152
P.O. Box 217 Enschede
The Netherlands

THREE WAYS OF WINDMILL TRANSPORT
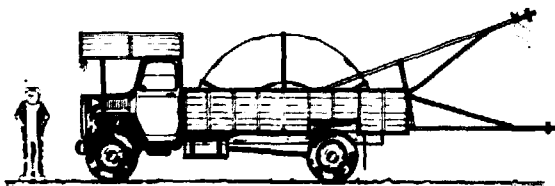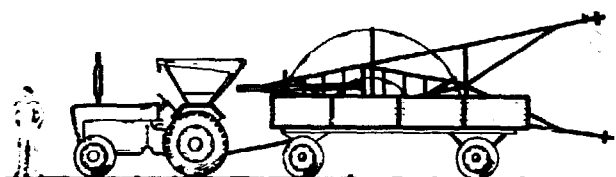
TRANSPORT BY TRUCK

TRANSPORT BY TRACTOR AND LORRY
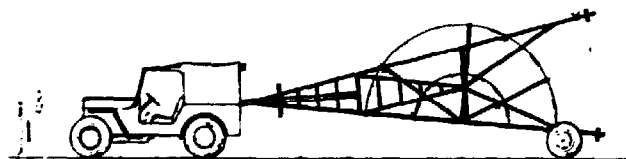
TRANSPORT BY JEEP AND ROLLING HELPDEVICE

MANPOWERED PULL ROPE

WATERLEVEL

TWO POLES (BAMBOOS OR A LADDER) FACILITATE THE ERECTION OF THE TOWER

OVERSHOOT LINES

DEEP: APPR. 40 CM
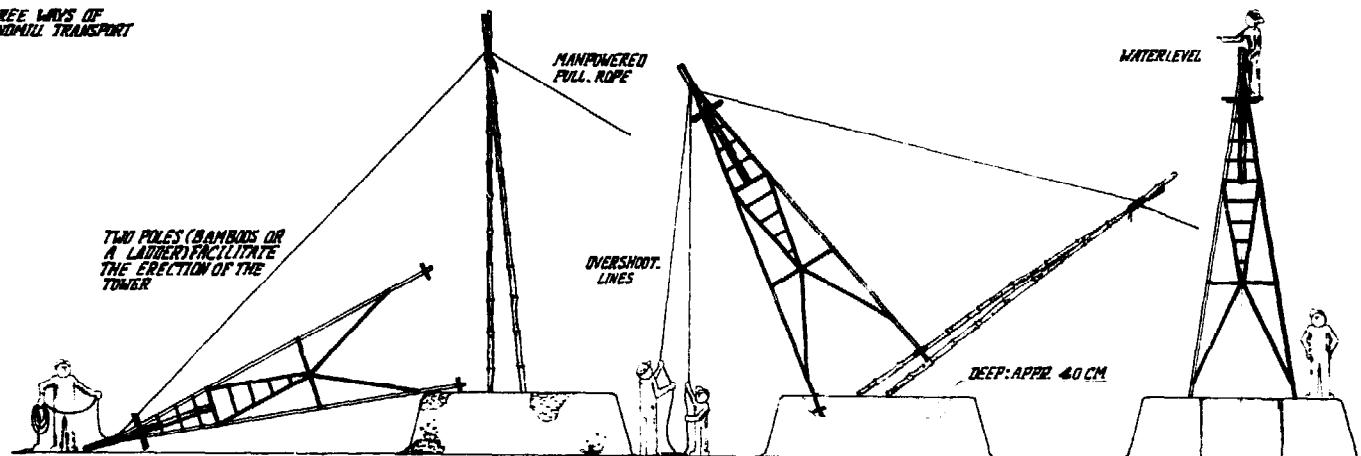
IMPORTANT: AFTER PLACING THE TOWER THE TOWERLEGS ARE POORED IN WITH CONGRETE AND AFTER APPR. 5 DAYS THE HEAD AND ROTOR CONSTRUCTION CAN BE INSTALLED SAFE AND QUICK BY MEANS OF LIFTINGDEVICES. DURING THE DRYINGPERIOD OF THE CONCRETE FUNDATION THE PISTON PUMP IS FIXED INTO THE WELL ACCORDING TO THE SPECIFICATIONS!

PUMP ROD

'718' LIFTING HELPDEVICE

LIFTING HOOK FOR HEAD CONSTRUCTION

BOLT ⑭ WELDED TO ⑤

BOLT ⑫ WELDED TO ③

20 260 120 120 120

BOLT ⑫ WELDED TO ③

ANGLE ⑦ OF TOWER (DRAWING Nº 3)

APPR. 5 DEG.

WINDDIRECTION!

TWO PERSONS STANDING ON THE PLATFORM ARE GUIDING THE HEAD AND ROTOR CONSTRUCTION DURING THEIR ASSEMBLING

HEAD IS TEMPORARILY TIED TO '718' (WIND!)

SINCE THE CENTRE OF GRAVITY OF THE HEAD CONSTRUCTION IS SITUATED ABOVE THE UPPER TAIL ANGLE A SPECIAL LIFTINGHOOK IS APPLIED. A SMOOTH SLIDING OVER THE TOWERPIPE IS ACHIEVED IF THE FRONT IS OUT OF BALANCE APPR. 5 DEGR.

ROTOR IS LIFTED WITH THE SHAFT COMPLETE WITH BEARINGS FIXED IN THE ROTORHUB IN ITS ORIGINAL POSITION (CENTRE POINTS!) AND FIXED ON THE HEAD CONSTRUCTION. NOW THE '718' CAN BE REMOVED AND THE BLADES, SAVETY DEVICE AND MOVING PARTS ARE ASSEMBLED.

Nº	NUMBER	MATERIAL/NAME/REMARKS/MEASUREMENTS	
14	2	BOLT NUT WASHER	M.12 x 75 (W ½" x 3")
13	2	"	M.12 x 150 (W ½" x 6")
12	2	"	M.12 x 50 (W ½" x 2")
11	1	ROD	Ø10 (Ø ⅜") x 200
10	1	FLAT	30 x 6 (1¼" x ¼") x 100
9	1	"	x 600
8	2	"	x 200
7	1	ANGLE IRON	40 x 40 x 4 (1½" x 1½" x 550
6	1	"	x 200
5	1	"	x 800
4	3	"	x 600
3	2	"	x 1200
2	1	GASPIPE	Ø 3" x 4500
1	1	PULLEY SHAFT AND ROPE OR STEELCABLE (30 M)	

INSTALLING THE WINDMILL — 12 PU 500 —

FOR INFORMATION: Workgroup on Development techniques
T.H. Twente — Wtkol 132
P.O. Box 217 Enschede
The Netherlands

2.51

3. Test-Rig for 12 PU 500 Windmill

In the past the development of windmills took place on the
site itself where they have been installed which created a
lot of inconvenience with regard to breakdowns and repairs.
It will be evident that such a procedure of technical develop-
ment takes a lot of time and moreover the image and populari-
ty of the technology might be degenerated due to failures.
Therefore it is advisable to simulate the practical conditi-
ons in such way that the development-process is done in a
shorter time and under heavy conditions without being de-
pendent on the fluctuating and unreliable wind. So the idea
was born to get a Test-Rig developed in which the essential
parts of the windmill can be tested thoroughly. This acti-
vity was carried out in the Research and Development section
at the Polytechnic in Allahabad.

3.1 Demanded features of the Test-Rig

Since the 12 PU 500 Windmill (12 blades Pumping Unit of 500
cm Rotordia) proved to be the most applied windmill so far
the urgency of developing its essential components is favoura-
ble. On the other hand results extracted from these tests can
be extrapolated usefully for other windmill types like 12 PU
250, 12 PU 350 and 12 PU 700 Windmills which are under develop-
ment too. Therefore it has been decided to get the Test-Rig
developed for testing the parts of the 12 PU 500 Windmill.
Considering the structure of the essential moving parts and
Pump of this particular windmill, testing of the following
parts should be carried out with regard to wear and tear:

- Main shaft bearings
- Crank Pin
- Cross Pin
- Cross head / P.V.C. - insert tower pipe
- Wooden pump-rod guide situated in the lower end of the
  towerpipe.
- Pump-rod guide (Bronze and Nylon).
- Leather sealing / pump rod
- Piston / Cylinder

- Leather valves (Piston-Foot-Valve).
- Joint between tower pipe and rotating head construction.

Actually there is no need of testing the rotor with regard to wear and tear, though fatigue cracks might occur. Field experiences showed that fatigue cracks had been developed right in the inner ring bladesupports which were made according to the first designs. The improved bladesupport proved to be sufficiently resistant with regard to fatigue and no failures appeared anymore. In this context there is no need to have the rotor tested.

Actually a normal 12 PU 500 Windmill installation provides the entire structure for a Test-Rig. Instead of the rotor a flywheel, which is driven by an electric motor, should be installed in order to obtain a regular drive to the fluctuating loaded mainshaft.

3.1.1 Drive of the Test-Rig

To estimate the required R.P.M.-number for the Test-Rig we consider the 12 PU 500 Windmill under extreme conditions. In practice the maximum R.P.M.-number does not exceed the value of 120 since the windmill is locked in secured position automatically by means of its automatic security device. This R.P.M.-number represents a reasonable value for testing pumps and the moving parts of the windmill.

Applying an electric motor for the drive a reduction of 1:12 is needed in the transmission. This is simply achieved by applying a V-belt transmission (directly) from motor pulley to the circumference of the flywheel. In practice the nominal diametres of pulley and flywheel turned out in 70 and 850 mm respectively while three V-belts (B-type) completed the transmission. The required power of the electric motor has been estimated at 4 K W.

12 PU 500
TEST RIG
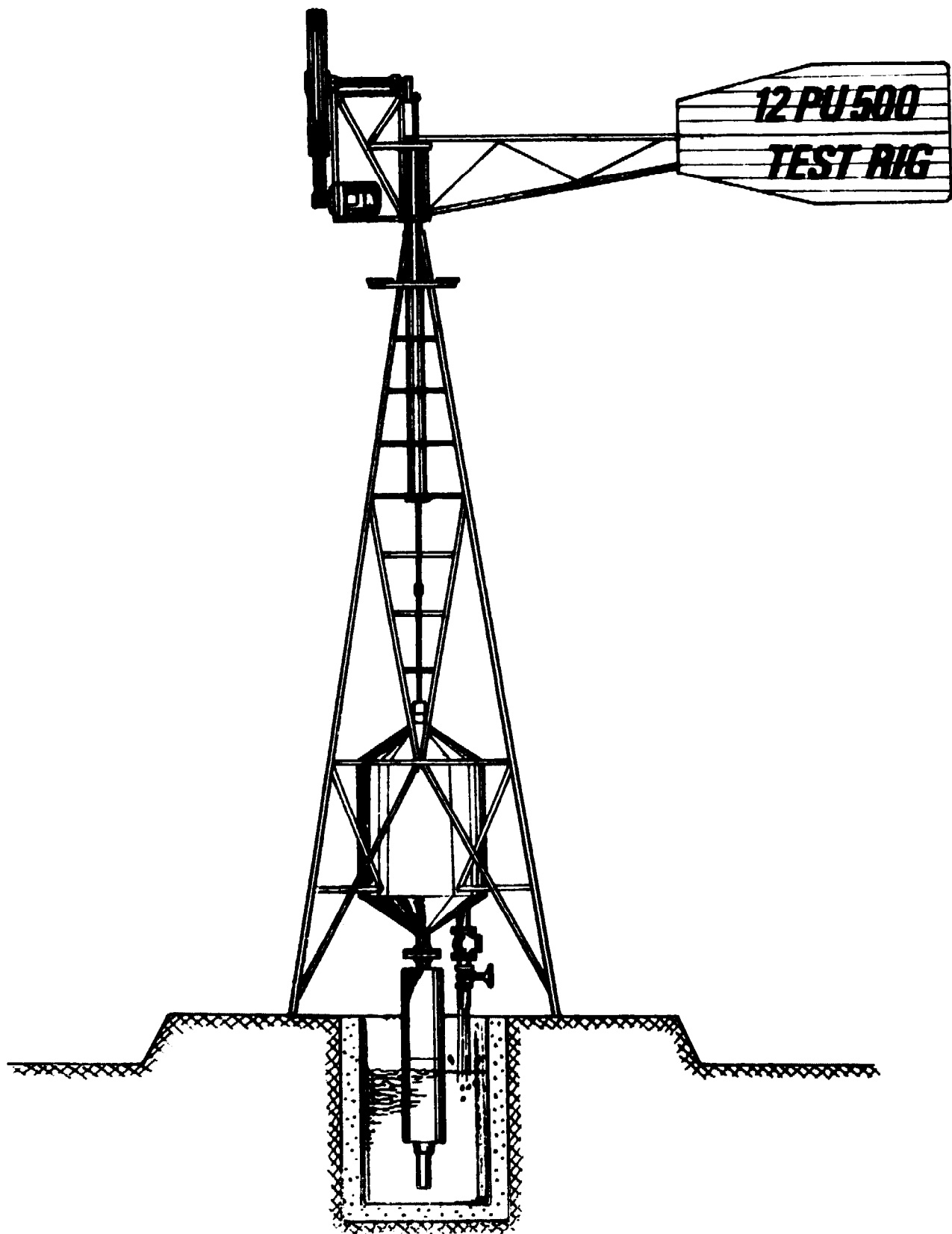
3.1.2 Selfmoving head construction

In normal condition the head just rotates if the wind-direction changes which means that in the long run wear and tear might be expected in the joint of the tower-pipe and head-construction. This rotation is made by piston and cross-head as well.

However, the tests should be carried out under laboratory conditions independently from appearing winds and wind directions. This implies that the rotation of the head construction should be simulated by forcing the head to carry out this movement regularly.

In practice it is very rare that the head rotates more than 180 degrees while in most of the cases swinging movements within a certain range of some degrees occur for a certain wind direction. Actually this is the worst condition from point of view of wear and tear since still the same part of the bearing surface (here: tower-pipe) is carrying the load.

So an attractive simulation seems to be achieved by forcing the head to swing to and fro around the tower pipe within a certain range,of degrees. This cycle should be carried out not too quickly and not too slowly but in such a way that the lubrication properties of the bearing are the same as under normal field conditions. However, it is quite difficult to quantify and to extract the required movement from the latter mentioned: wind patterns are different from place to place and from time to time.

As a compromise it has been decided to swing the head construction to and fro within the range of plus and minus 45 degrees (total swing 90 degrees). In order to obtain this movement a small crank is driven by a worm-wormwheel-transmission (1:80). The latter is fed by the main shaft (120 R.P.M.) on which the worm is fitted.
The crank is linked to a joint of the tower by means of a con-

necting rod. This mechanism forces the head construction
to swing over the tower pipe. One cycle is completed in
40 seconds while this motion appears to be quite "natural".
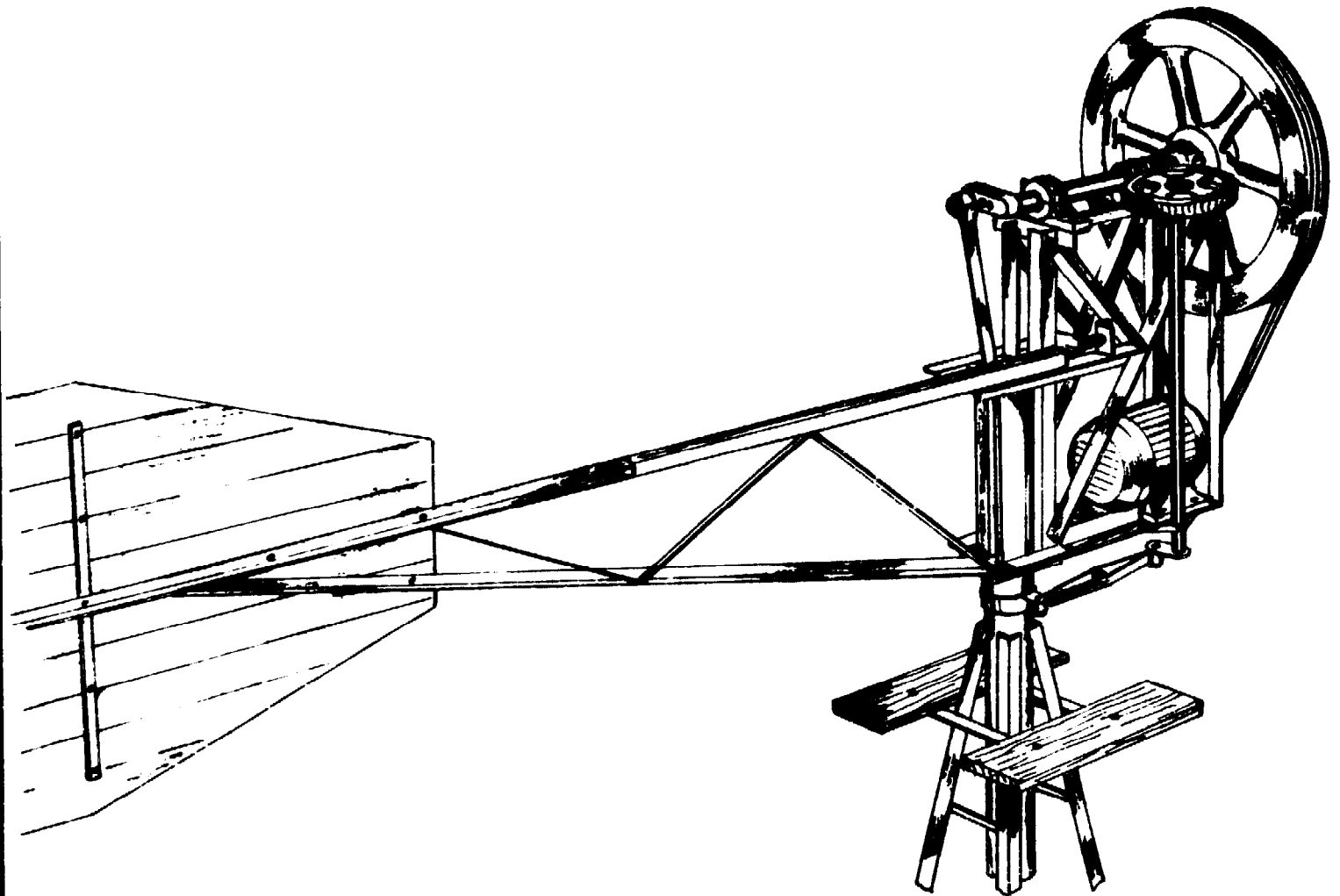
3.1.3 Simulation of the elevation head

In practice the windmills lift water over several elevation
heads up to 20 metres or even more. Then the dimension of
the pump applied depends on the actual head. Pump dimensi-
ons in the 12 PU 500 serie are: 3", 4", 5", 6" and 8"
(piston diameters.) It is expected that, as far as pump-
testing is concerned, this range of elevation heads should
be applied during these tests.

In laboratory conditions it will be possible to simulate the
elevation -heads by applying long pipe-lines leading to an
overhead tank which should be situated on several levels.

This method is expensive and it creates quite a lot of
inconvenience with regard to installation and of course it
will not be practical.

Simulation can be carried out in a simpler way by pumping
the water from a sump (bottom basin) and getting it delive-
red into an air-tight vessel where a static pressure persists.
This pressure simulates the elevation head and it is built up
by compressing the existing air which remained above the
water table in this vessel. The latter is achieved by res-
tricting the exhaust opening at its bottom-side by means of
an adjustable valve over there. Its resistance causes the
required pressure to get the actual water flow forced through
it,back into the sump.
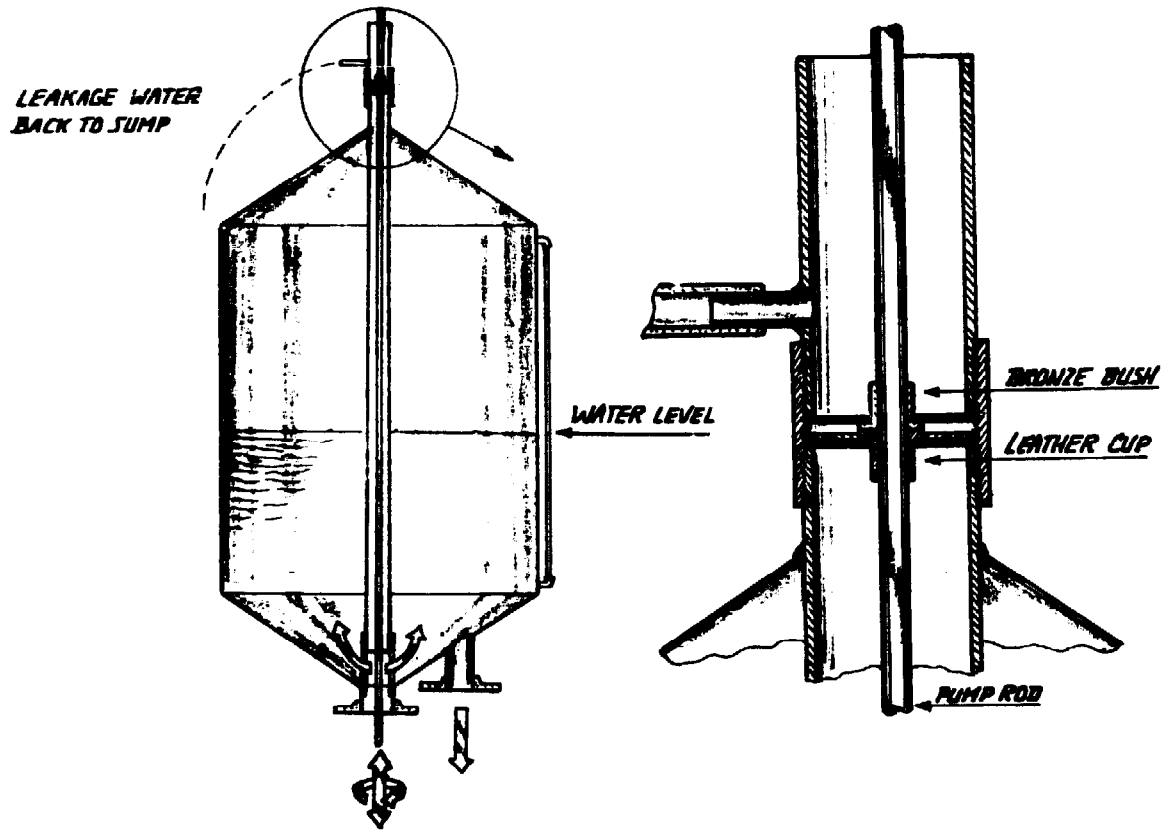
The main advantage of this system is that it is compact to be
built. However its disadvantage is that the inertia-effects
and pressure losses, which normally appear in long and
narrow pipelines,are not simulated since they are not there.
Instead the colossal vessel with compressed air is there
which provides a huge air receiver which acts as an capacitor

equalizing the irregular waterdelivery from the single acting
piston pump. Therefore it is advisable to consider this air
pressure not to be the static elevation head (geodetic head)
but the actual <u>hydraulic</u> head! The latter is known as static
head plus the pressure losses.

The pressure vessel is situated in center of the Test Rig-
tower in order to integrate it with the entire structure.
The tower provides suitable attachments for the suspension
to the several lattice joints.However a practical problem has
to be solved with regard to the leading of the pump rod
through the centre of the tank since its airtightness is
most essential. This implies the application of a <u>pump rod</u>
<u>sealing</u> which should not permit any leakage of air. The
latter is achieved by applying a pipe in centre of the tank
which bottom ends under water surface. On its topportions
the pump rod sealing is situated. In this way air leakage is
avoided in case the sealing proves not to be perfect
  since in that case only water-leakage will appear without
any effect to the functioning of the airchamber.

3.1.4 Instruments

In order to run the Test Rig in a proper and safe way and to
gather information with regard to lifetime and efficiency of
the pumps to be tested, several instruments have to be applied.
These instruments are:
- Switch and relais box
- Amps meter and KWH-meter
- Water meter
- Revolution counter
- Pressure gauge
- Water level gauge

Switch and relais box

Besides the application of a main switch box it is preferable
to have a relais box installed. Purposes of this device are:

- Switching-off the remaining phases if one of the phases is
  lacking due to power cuts. (If this is not done in time the
  electric motor will burn definitely.)

- Switching-off the electric motor in case one (or more)
  of the phase-currents exceeds an unacceptable value. (By
  means of thermo-elements.)

- To enable the operator to start and stop the Test Rig by
  a "simple push of the button". The actual switch is made
  by the relais.

The relais box is attached to the head construction of the
Test Rig in order to obtain a short multi wired connection
to the electric motor. This also means extra safety since the
Test Rig can only be started by the operator which is at the
machine itself; so the machine can not be started from ground
level while an operator is carrying out some job at the top of
the Test Rig. In case of emergency the machine can always be
stopped at ground level by using the main switch.

Amps meter / KWH-meter
It is advisable to have an Amps meter installed in order to
detect any overload. Moreover the Amps-meter provides infor-
mation to what extent the machine is loaded.
The same applies for the KWH-meter (3-phase)which counts the
amount of energy which has been consumed to drive the test
pumps under different conditions.
Till a certain extent this power consumption can be derived
for the specific pump under the specific conditions.

Water-meter
This instrument measures the total amount of water which has
been delivered during the pump tests. This reading is most
essential in order to calculate the volumetric efficiency of
the test pump. The Water-meter is situated in between the pres-
sure tank and the adjustable valve through which the water is

flown back into the sump.

Revolution counter

The revolution counter registrates each revolution of the main shaft. Its counting is essential for calculating volumetric efficiency and judging lifetime features of the test pump and moving parts as well.

Pressure gauge

As mentioned before the pressure in the tank represents the total hydraulic head, which is indicated by the pressure gauge.

Waterlevel gauge

It is advisable to indicate the actual waterlevel in the pressure vessel. This in order to detect any leakage of air which will lead to a decrease of the capacitor effect of the airchamber resulting in a "hammering" run of the machine. Under static conditions the waterlevel gauge also provides a (rough) indication of the actual pressure in the tank.

3.1.5 Way of testing

Running the Test-Rig is done in day-time (working hours) in order to have a close look at the experiments. Day by day the following readings are taken and noted:
- Water passed through water-meter.
- Number of revolutions indicated by counting device.

These readings provide the information for calculating the volumetric efficiency. (See next chapter.)

Pumps to be tested (4" - 5" - 6") are run at 15, 10 and 7 metres head respectively simulating the hydraulic heads. Adjustment of this head is achieved by tuning the adjustable valve situated in the return-flow. One should be aware that this adjustment is "tricky" for the following reasons:
- Its initial adjustment is quite a time-consuming procedure which is achieved by regularly made check - ups and re-

adjustments of the valve.

- After some time the features of the test pump change resulting in higher out-put (setting of valves) or lower output (wear) with the effect that the equilibrium of the system is going to be disturbed. Hence the regular check-ups are necessary too in order to give re-adjustments if needed.

The pumps are tested for at least one million revolutions while recording wear and tear and calculating volumetric efficiencies is done at certain intervals. The out-come of these will provide the information to base conclusions on.

3.2 Pumps to be tested

Since the completion of the Test Rig (March 1980) four types of pumps have been tested each for one million revolutions.
The test pumps were:

                        ---- 4" S 500
                        ---- 5" SS 500
                        ---- 6" P.V.C. 500
                        ---- 6" S 500

Each pump has been tested turn by turn during cycles of 250,000 revolutions loaded at hydraulic heads of 15, 10 and 7 metres respectively. At the end of each experiment of 250,000 revolutions the pumps have been checked up (and revised, if necessary).

In these check-ups the qualification of wear proved to be rather complicated. This in the sense that e.g. play of piston/cylinder and pumprod/pumprod guide are difficult to measure and to define due to unroundness of the several parts. Although initial unroundness exists in the cylinder (black pipe, P.V.C. pipe) and pump rod (gas pipe) which are acceptable (from the point of view of Appropriate Technology) wear and tear seem to cause an extra unroundness to these

particular parts. No doubt that the latter develops due
to improper alignment of the structure and slight bends in
the pumprod sections, (which is always the case in practice).
In order to judge the extent of wear of the parts in such
cases the non-measurable qualifications have to be expres-
sed in degrees.

3.2.1 The 4" S 500 (Type I, see chapter 2)

This type of pump has been designed and introduced in the
beginning of 1980 as a deepwell-pump. (Applied for the first
time at Gohri.)

It consists of two main sections as follows:

- <u>Topsection</u>: composed of 2 mm steel casing, delivery pipe,
top flange and pumprod guide.
The latter contains a brass guide bush.
The space between the delivery pipe and the
casing provides the air pressure chamber.

- <u>Bottom section</u>: Composed of 2 mm steel casing, steel cy-
linder, top- and bottom-flange, wooden foot
valve body and leather foot valve.
The wooden sealingless piston has a proper
fit in the cylinder which has been honed by
means of a simple honing device.

Though the nominal cylinder diameter is given in 4" the exact
diameter amounts 106.6 mm. (Actually such deviations also exist
for 5" and 6" pumps.)

The eight holes in piston and footvalve body are situated at
a pitch circle of 0.65 D and the diameters are 0.2 D.
This has been standardized for all the pumps.

3.2.2 The 5" SS 500

Basically this pump is of the same construction as the former though the construction of the footvalve differs and the piston has been fitted with a <u>P.V.C. piston ring</u>. These special arrangements provided the "special" label to this test pump hence an extra S has been taken up in its code number.

Its foot valve arrangement consists of a wooden footvalve body and a 2 mm steel disk which has been lined with leather. This disk contains a steel hub which slides over the central bolt of the valve arrangement and is pressed to the valve body by a spring.

Since the rigidity of this valve disk does not provide the "Umbrella-effect", if opened like the elastic leather valves have, extra "flow round" space had to be created. This has been achieved by constructing a wider bottom part (of 6") to the 5" cylinder. Hence the outside dia of the wooden foot valve body has been made at the nominal dia of 6" but of course the hole-arrangement (dia, and pitch) refers to the actual 5" cylinder dia.

As an experiment the piston has been fitted with a PVC-ring which should act as a sealing. This 8 mm wide ring has been cut from a 6"PVC-pipe and is situated in a groove in the wooden piston. This experiment could provide lifetime infor-mation of such a sealing.

3.2.3 The 6" P.V.C. 500 (Type H, see chapter 2)

This pump, known as the "combination pump", has been developed in Ghazipur (1979) and proved to be of a quite satisfactory design. It consists of a Top – and a bottom – section.

– <u>Top section</u> : Composed of a 6" PVC-pipe (1.20 m long), top flang and delivery pipe. The latter carries the pumprod guide (nylon). Here

also the space between delivery pipe and
PVC cylinder creates the airchamber. The
wooden sealingless piston runs in the PVC-
cylinder with a proper fit.

- <u>Bottom section</u>: Composed of a 6" PVC-pipe (0.75 long),
wooden footvalve body and leather valve,
PVC (4") airchamber and bottom flange.

Both sections are jointed by means of 4 anchor rods.

3.2.4 The 6" S 500. (Type J, see chapter 2)

As a matter of fact this pump is basically the same as the
other steelpumps. Only the foot valve differs:
which is made of a sheet-iron disk in stead of wood. Therefore
this pump is shorter in total length resulting in a more eco-
nomical use of sheet materials during its fabrication.

3.3 Test results

The schemes given on the next pages show results of the check-
ups made after each interval of 250,000 revolutions.
As mentioned earlier quantifying the extent of wear proves to
be quite difficult with the result that judgements are given
in degrees of wear.

The first 250.000 revolutions

During this test interval it is found that the brass pumprod
guide showed a better performance than the nylon one. The
latter already wore out in an oval shape which was already
visable. The leather valves functioned without any breakdown
and developed a good setting to their wooden counter-parts.
In contradiction with the expectation the disk-valve of the
5" SS 500 developed a good setting too, though its "steel to
steel" slide of its disk-hub over the central bolt showed
some wear. The fit of the wooden pistons in the steel cylin-
der remained proper in the sense that some jerking fit in

some positions persisted. In the PVC-pump the fit already got so loose that some play between cylinder and piston could be seen.

The PVC piston ring, applied in the 5" SS 500, wore rather seriously. The tips of the ring became sharp like a razor blade. Moreover the wear was not uniform; the thickest portion of the ring (4.0 mm) hardly showed any wear (approx. 0.2 mm) while the thinnest portion measured 1.4 mm. It has been decided to stop the experiments with the PVC ring for the time being till the cylinder surface has been polished thoroughly by the plain piston after some newly undertaken experiments. Probably a longer life of the ring would be achieved then.

After the first 500.000 revolutions

Still the brass pump-rod guides show a good sliding fit as well as the piston / steel cylinder combinations. All these moving parts showed nicely polished surfaces.
The fit of the piston in the PVC-pump showed an increased play. Though difficult to measure, its play is estimated at 2mm while a clear edge in the PVC is detected at the end and beginning of the track of the piston. Most problably due to the initial unroundness of this PVC-cylinder the piston seemed also to be worn off which is proved by a loose fit of it in an unused part of the cylinder.
The nylon pumprod guide showed increased wear. Estimated play of pumprod 1÷2 mm.
The valves continued functioning well. Though once the central nut of the leather foot-valve of the PVC-pump ran loose (due to poor fixation). Obviously the hub of the steel disk-valve as well as its central bolts showed no increase in wear. Probably its combination developed a good "run-in".

After the first 750,000 revolutions

The brass pumprod guide still shows an acceptable fit while
the nylon guide wore out in an oval shape causing an esti-
mated play of 2 mm.
All valves are in good condition except the leather foot-
valve of the PVC pump which developed cracks due to fatigue so
it had to be replaced.

The piston / steel cylinder combinations are about to show
the first wear, though not simple to quantify (say 0.5 mm
play). It is assumed that wear in the steel cylinders (if any)
is to be neglected: the inside surfaces just show nice
polishing.

The experiment with the PVC piston ring failed again : at
approx. 650,000 revolutions (so this ring just run approx.
150,000 revolutions) the ring wore off that much that it slip-
ped in between the piston and cylinder.
This caused that the wedged piston has been pulled up to its
upward position with such a force that a cleavage fracture
occurred in the cross-head and the connection rod had to
collaps during the downward stroke. (Fortunately the Test-
Rig could be stopped immediately.)

The experiments have been continued again after removing the
badly deformed ring.

After the first 1,000,000 revolutions

The brass bushes show detectable wear which is still accept-
able, but the clearance between pumprod and nylon guide was
increased up to 3 mm.

All the piston valves as well as the foot-valves were still
in good condition. However, in the 5" SS 500 the spring, ap-
plied at the top of the disk-foot valve, broke (after approx.
860,000 revolutions) and had to be replaced.

Clearance in the piston / steel cylinder combinations was
detected now and it had been estimated at approx. 1 mm. On
the other hand the clearance in the 6" PVC 500 increased to
more than 2 mm.

3.4 Volumetric efficiencies

All the pumps showed a decreasing volumetric efficiency
with increasing number of revolutions. Obviously this is due
to wear of piston and/or cylinder. Although in two cases
(5" SS 500 and 6" PVC 500) considerable wear has been de-
tected still the volumetric efficiencies exceed 75% for the
4" S 500, 80% for the 5" SS 500 and 80% for the 6" PVC 500
and 6" S 500 respectively.
In all cases the decreasing behaviour of the volumetric
efficiency does not seem to be due to malfunctioning of
valves, since their condition did not change during the 1-
million revolution tests.

It is hard to make a comparison between the pumps based upon
the registered volumetric efficiencies since relative leaka-
ge will depend upon size and pressure.
Moreover taking this in consideration, it seems that
the steel cylinders show better performance than the PVC
cylinder as far as volumetric efficiency is concerned. Next
pages show the testresults, which are calculated from the
readings during 50,000 revolutions in intervals of 100,000
revolutions.

3.5 Conclusions with regard to life time- features of the test pumps

It will be clear that pumptesting to one million of revo-
lutions just provide modest information with regard to life-
time features of the pumps.
Though further testing should be applied, yet some conclu-
sions can be drawn as far as the test results have provided
obvious information.

- The combination of a wooden piston and PVC-cylinder can not compete with the wooden piston/steel cylinder combination.

- Nylon Pumprod guides show a much quicker wear than those made of brass.

- A PVC-ring hardly covers a life of 250,000 revolutions while it creates risks for the moving parts of the windmill (here: Test Rig) if it slips in between the piston and cylinder.

- The application of a spring on top of the foot-valve (here: 5" SS 500) might fail due to fatigue (crack of the spring), causing damage to piston/cylinder combination.

- Leather valves, as applied so far, proved to be quite reliable though one should apply a good quality of processed leather (sole-leather). The main disadvantage of leather is that after it has been dried its elasticity is gone and fatigue sensitivity increases.

After testing four pumps (each during one million revolutions) the Test Rig completed another one and a half millions revolutions which should provide some significant information. As a matter of fact it drove pistons up and down covering a total distance of almost 3000 kilometres. The head construction was forced to swing to and fro more than 65,000 times in order to create the possibility to study the wear between tower pipe and head.

In this stage the following conclusions are drawn with regard to life time features of the moving components of the 12 PU 500 windmill.

- No visible wear has been noticed in radial sense in the joint of tower-pipe and head. In axial sense the head just sank approx. 0.5 mm during the run-in period. Obviously after setting of the contact surfaces no additional wear appeared.

$\eta_v \%$

$4"S\,500\ (15\ M.\ HEAD)$

$10^6 Rev.$

$\eta_v \%$

$5"SS\,500\ (10\ M.\ HEAD)$

$10^6 Rev.$

$\eta_v \%$

$6"PVC\,500\ (7\ M.\ HEAD)$

$10^6 Rev.$

$\eta_v \%$

$6"S\,500\ (7\ M.\ HEAD)$

$10^6 Rev.$

3.17
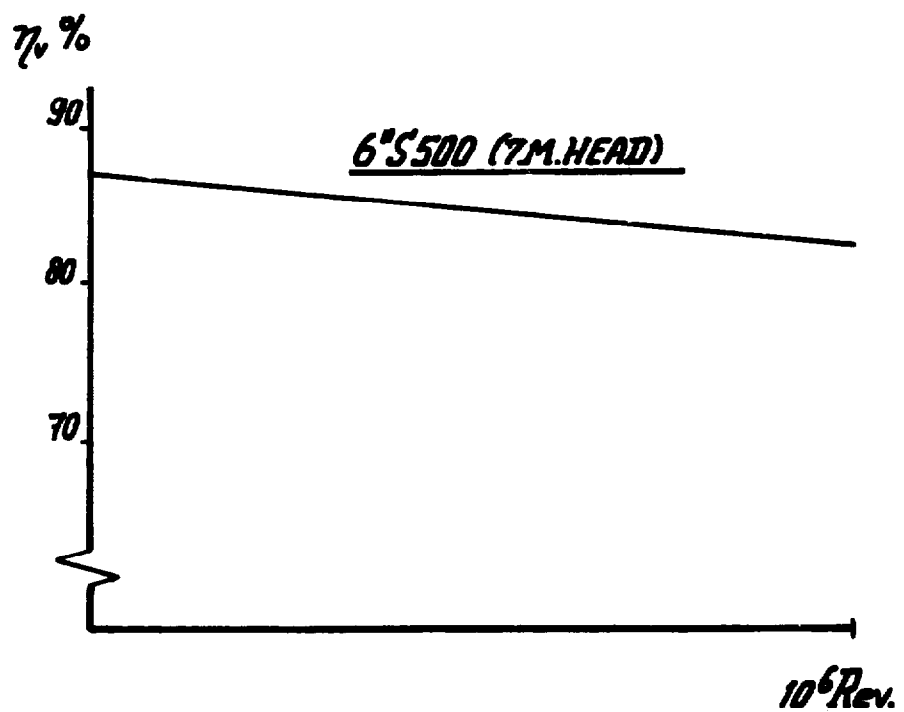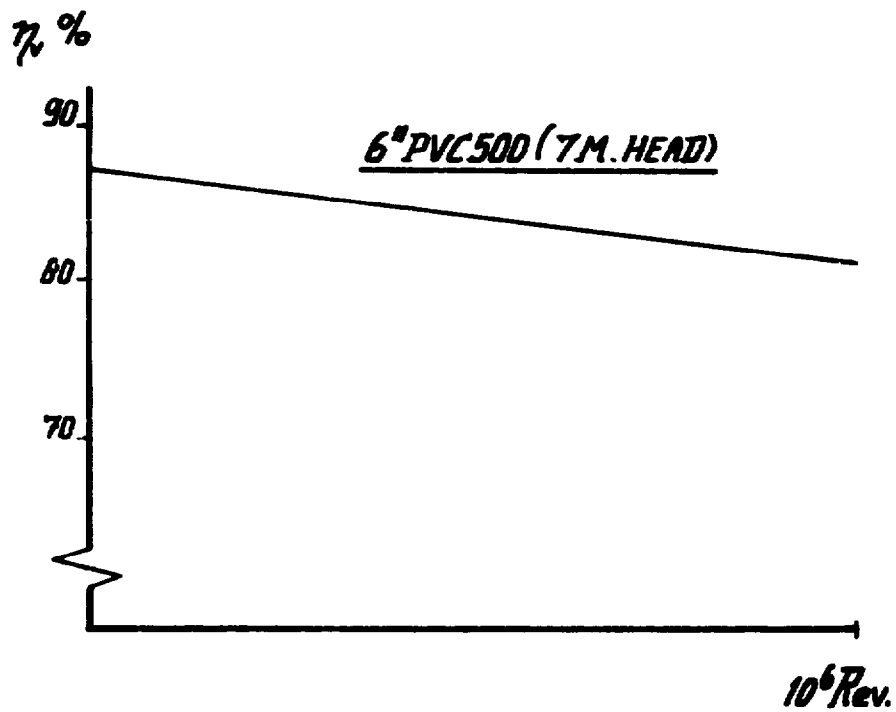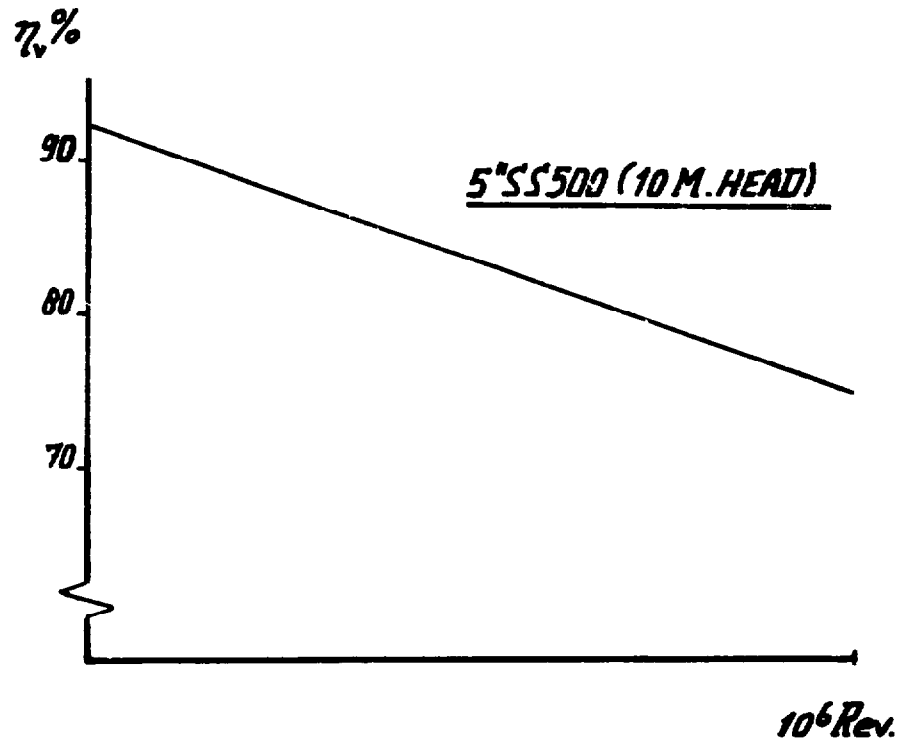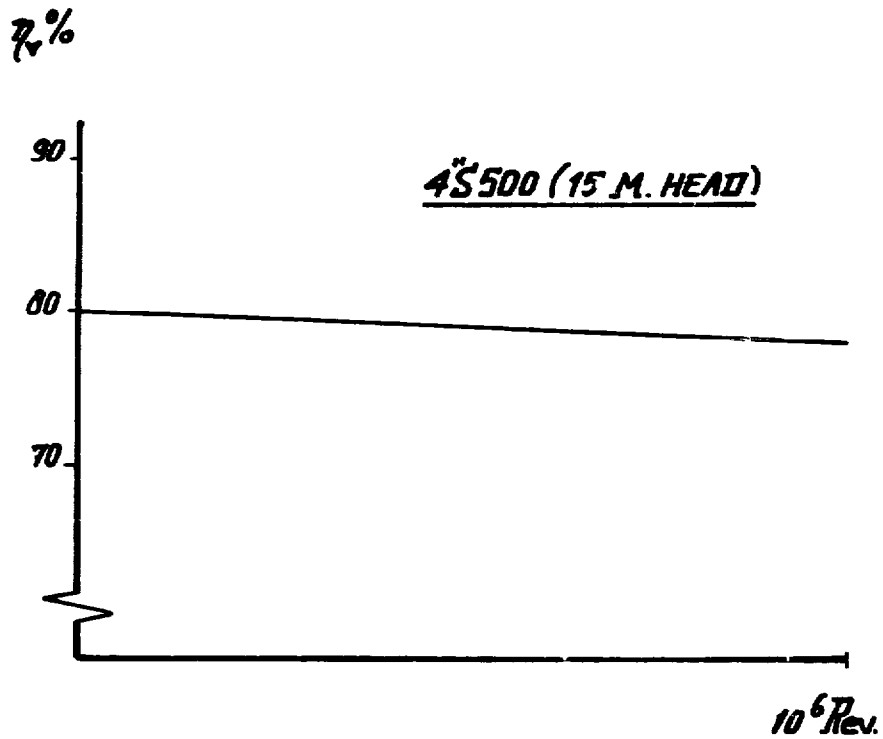
It is concluded   that the joint will serve the purpose per-
sistently.

- The bearings of the mainshaft (N.B.C.-double roller, self-
  aligning) did not show any defects or play, while no additi-
  onal lubrication was applied.

- Though once, the crosshead had to be replaced (cleavage due
  to jerking piston) its bearing properties in combination with
  the P.V.C.-towerpipe-lining seemed to be excellent. However,
  still the tests should be continued in order to find out to
  what extent wear will develop in the long run.

- The nylon bearing bushes in the connecting rod did not show
  any wear of relevance. Neither did the cross- and crank pin.

Though these results appear to be promising it is understood
that in a suitable windy area the 12 PU 500 windmill runs in
the range of 500,000 up to 1,000,000 revolutions per month.
Hence the experiences, gathered so far, just represent such a
windmill during the first three up to six months of its life-
time. In this regard these achievements should not be seen as
a milestone,but rather a beginning.

4. Properties and characteristics of the 12 PU 500.

As mentioned in the first chapter the windspeeds in Gha-
zipur are rather low and actucally the windmill should be
sufficiently adapted to these circumstances in order to
utilize the little available windpower in the most effi-
cient way. The size of the piston pump (piston dia. and
stroke) as well as the elevation head are the most impor-
tant parameters in the windmill installation which define
the starting windspeed (cut-in windspeed) and the design
windspeed. The latter is the windspeed for which the maxi-
mum overall efficiency is obtained.

4.1. Input-Output relations, volumetric- and overall-efficiency.

Considering the total windmill installation as a converting-
system the relationship of input and output is interesting
with regard to its efficiency, its capacity and the economics.
The applied measurement equipment at the R.T.I.-windmill was
not suitable for the definition of the input-output relation-
ship since the windspeed is monitored interval-wise while the
R.P.M.-number and output of the windmill was fluctuating
continuously due to changings in the windspeed.
Accurate readings can just be taken if the circumstances are
stationary which means that the windspeed as well as the
R.P.M.-number are constant while the flow measurement is taken.
These data will provide a relation between input and output and
the required instruments are:
- anemometer
- R.P.M. counter
- flowmeter

Although these instruments were available the above mentioned
measuring could not be carried out satisfactorily. In particular
the flow measurements proved to be rather inaccurate due to the
irregularity of the waterdelivery of the single-acting piston
pump.

Therefore some efforts have been undertaken in developing a
simple flowmeter and finally two flowmeters resulted. The
first one consisted of a simple 1-gallon oilcan. The flowing
water enters the open bottomside and is released through the
screw-top-opening. The static waterlevel in the can will repre-
sent the actual flow which is indicated by a "gauge-glass"
simply made from a plastic hose. These oilcans are standar-
dized so a similar flowmeter can be made in any place of the
world, but its use is restricted to a maximum flow of approx.
7 m^3/hr. Moreover an irregular flow causes considerable errors
since this instrument reacts rather slow and airbulbs spoil
the readings depending on the way of entrance of the water.
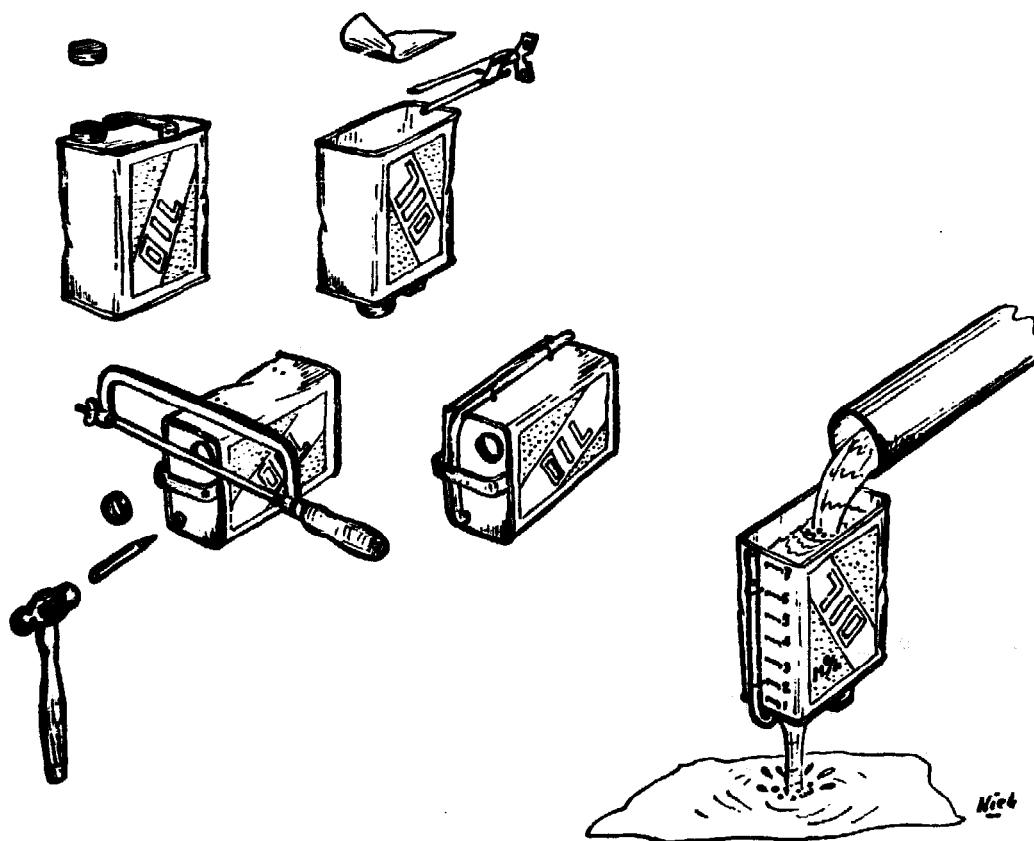
The second device of flowmeasuring consisted of a shallow basin
in which the flow enters the back-side and leaves this basin
on the other side via a V-notch. The static waterlevel in the
basin also represents the actual flow according to several theo-
retically defined tables. Unfortunately this equipment did
not serve the purpose very well. The phenomenon of slowness,
due to the large volume, caused that this device acts too late
in order to determine the actual output.

So both measuring devices failed because the slowness in indicating made these "instruments" unsuitable for direct flow registration.

Due to this another way of flowmeasuring has been developed: in a period of constant windspeed and constant R.P.M.-number (audible) a 20 l. drum is shifted under the output opening of the pistonpump during the downward -stroke (no output!).
At the same time a stopwatch is switched on and from that moment on the number of strokes is counted (e.g. four or five).
Then during the downwards-stroke again, the drum is removed quickly and its contents will amount to the delivery of four (or five) strokes. The timespan, registrated by the stopwatch, the windspeed and the amount of water delivered will provide sufficient information in order to calculate respectively the input-output relation, the volumetric efficiency and the overall efficiency.

The above mentioned method proved to be the most accurate and reproductive way of measuring.

In the following table the results (screened) are printed of the first measurement which have been carried out in May 1978.

Dia. pump: 132 mm.
Stroke: 254 mm. (10")
Elevation head: 6.5 m.
Pumptype: A (see pistonpumps)

V (m/sec.)	n (R.P.M.)	$\emptyset$ (dm^3/sec.)	η_v —	η_{tot} —
2.5	14	0.6	0.74	0.21
3.0	25	1.2	0.83	0.24
3.5	31	1.6	0.90	0.20
4.0	37.5	2.0	0.93	0.16
4.5	43	2.4	0.96	0.14
5.0	49	2.8	0.98	0.12
5.5	55	3.2	1.00	0.11
6.0	61	3.6	1.02	0.09
6.5	68	4.0	1.01	0.08
7.5	76	4.4	1.00	0.07

The maximum overall-efficiency amounts to 0.24 for a wind speed
of 3 m/sec. For higher values of the windspeed the overall-
efficiency is decreasing up to a few per cent only.
Improvements have been obtained later when the blades of the
rotor were replaced by the narrow ones and the pumpsize was
increased from 132 mm. to 150 mm. New measurements were carried
out in May 1979. The condition of the piston was rather bad;
wear of the piston caused a considerably affected output due
to internal leakage, but in spite of that higher overall-
efficiences have been extracted from these measurements:

Dia. pump: 150 mm. (clearance appr. 3.5 mm.)
Stroke: 240 mm.
Elevation head: 6.5 m.
Pumptype: E (see pistonpumps)

V (m/sec.)	n (R.P.M.)	ϕ (dm^3/sec.)	η_v —	η_{tot} —
2.5	21	1.0	0.67	0.32
3.0	26	1.3	0.71	0.24
3.5	31	1.5	0.68	0.18
4.0	36	1.7	0.67	0.13
4.5	42	2.0	0.67	0.11
5.0	47	2.5	0.75	0.10
5.5	55	3.0	0.77	0.09
6.0	63	3.5	0.78	0.08
6.5	70	4.0	0.81	0.07
7.0	78	4.5	0.82	0.07
7.5	86	5.0	0.82	0.06
8.0	96	5.5	0.81	0.05
8.5	105	6.0	0.81	0.05

Although these results are not quite representative
(low volumetric efficiency) one can conclude that overall-
efficiencies of acceptable value can be obtained even if the
pistonpump is not in proper condition. After replacing the
piston new measurements have been carried out with respective-
ly 0.5 mm. and 1.0 mm clearance between piston and cylinder.
The difference in these measuring data proved to be in-
significant so both data have been combined:

Dia. pump: 150 mm. ($0.5 \div 1$ mm. clearance)

Stroke: 240 mm.

Elevation head: 6.5 m.

Pumptype: E (see pistonpumps)

"Cut-in" windspeed 2.5 m/sec.

V (m/sec.)	n (R.P.M.)	$\emptyset$ (dm^3/sec.)	η_v —	η_{tot} —
3.0	25	1.5	0.85	0.30
3.5	31	1.8	0.82	0.23
4.0	36	2.2	0.87	0.19
4.5	42	2.6	0.88	0.15
5.0	47	3.0	0.90	0.13
5.5	54	3.5	0.92	0.11
6.0	60	4.0	0.94	0.10
6.5	67	4.6	0.97	0.09
7.0	77	5.2	0.96	0.08

Decreasing the stroke from 240 to 160 mm. reduces the output and "cut-in" windspeed.

Dia. pump: 150 mm. ($0.5 \div 1$ mm. clearance)

Stroke: 160 mm.

Elevation head: 6.5 m.

Pumptype: E (see pistonpumps)

Cut-in windspeed 2.2 m/sec.

V (m/sec.)	n (R.P.M.)	$\emptyset$ (dm^3/sec.)	η_v —	η_{tot} —
2.5	26	0.9	0.74	0.31
3.0	31	1.1	0.76	0.22
3.5	37	1.3	0.75	0.16
4.0	43	1.5	0.74	0.13
4.5	49	1.7	0.74	0.10
5.0	55	1.9	0.74	0.08
5.5	61	2.2	0.77	0.07
6.0	69	2.4	0.74	0.06
6.5	76	2.7	0.76	0.05
7.0	85	3.0	0.75	0.05

In the next paragraphs extra attention will be paid to the windmill-pump-combination.

4.2. The powercharacteristics of the windmill.

By means of the former described way of measuring several
properties of the windmill-pump combination can be defined
but still the power characteristics of the windmill as well as
the pump are completely unknown. In order to investigate this
also torque-measurements must be carried out. In practice
many complications will be met in order to realize a reliable
torque-measuring because the actual torque will fluctuate
considerably due to the irregular drive of the pistonpump.
Knowing the characteristics of the rotor the conversion of
windpower into useful mechanical power can be calculated
then for any windspeed and R.P.M.-numbers. This characteristic
is known as the $c_P - \lambda$ curve; c_P is here the power coefficient
(the rate of power conversion) while λ stands for the tip-speed
ratio. Both dimensionless ratios are defined as follows:

$$c_P = \frac{P}{\frac{1}{2} \rho V^3 \pi R^2}$$

$$\lambda = \frac{\Omega R}{V}$$

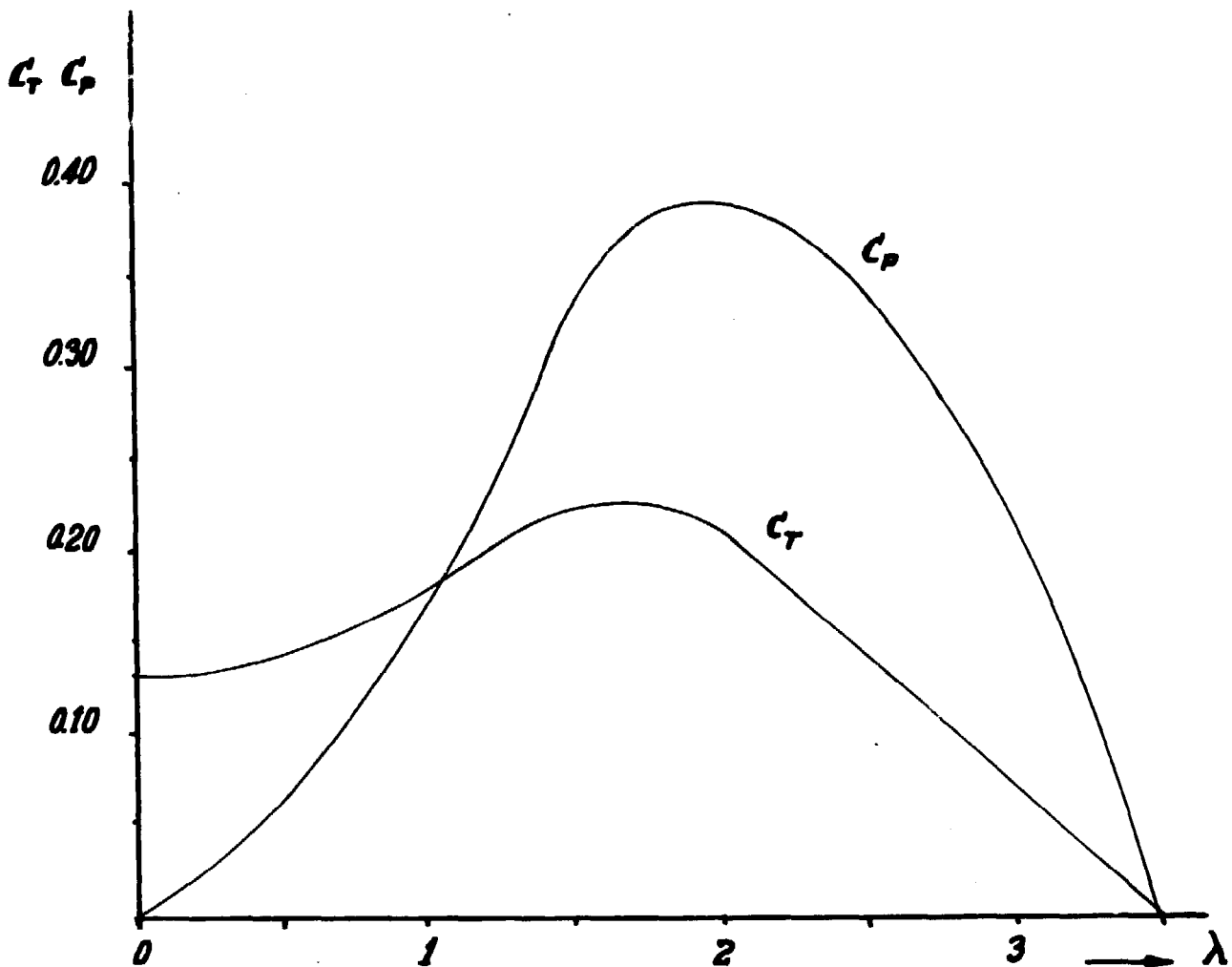Besides a dimensionless torque-coefficient is defined as
follows:

$$c_T = \frac{T}{\frac{1}{2} \rho V^2 \pi R^3}$$

The symbols are representing:

P   = useful converted power   [Watts]
T   = useful converted torque   [Nm]
ρ = density of air   [1.2 Kg./m^3]
V   = windspeed   [m/sec.]
R   = radius of the rotor   [m]
Ω = rotor angular velocity   [rad/sec]

Both the C_T - λ curve and the C_P - λ curve will represent
the main properties of the rotor in a certain range of
Reynolds numbers (Re). Since 10% arched steel plate airfoils
are applied the curves will be valid if Reynolds-numbers
appear which exceed the <u>critical</u> Re-number of 10^4 since the
applied airfoil properties, which determined the blade design,
are just valid under this condition $(RE > 10^4)$. By the help of
some SWD-members (SWD= Steering Committee for Windenergy in
Developing Countries) the C_T-λ and C_P-λ curves of a scale-model
(1.5 m. dia.) resulted from windtunnel-tests. In these tests
the influence of the presence of several disturbing parts
in the rotorframe has been investigated. Also the presence
of the helpvane of the safety-device proved to affect the
   C_P - curve resulting in lower efficiencies.
The assumption that this C_P-λ curve will represent the 5 m.
dia. windmill provides a lot of information with regard to
the actual generated power and torque. Also the powercharac-
teristic(s) of the pistonpump can be derived then. It is

12 PU 500
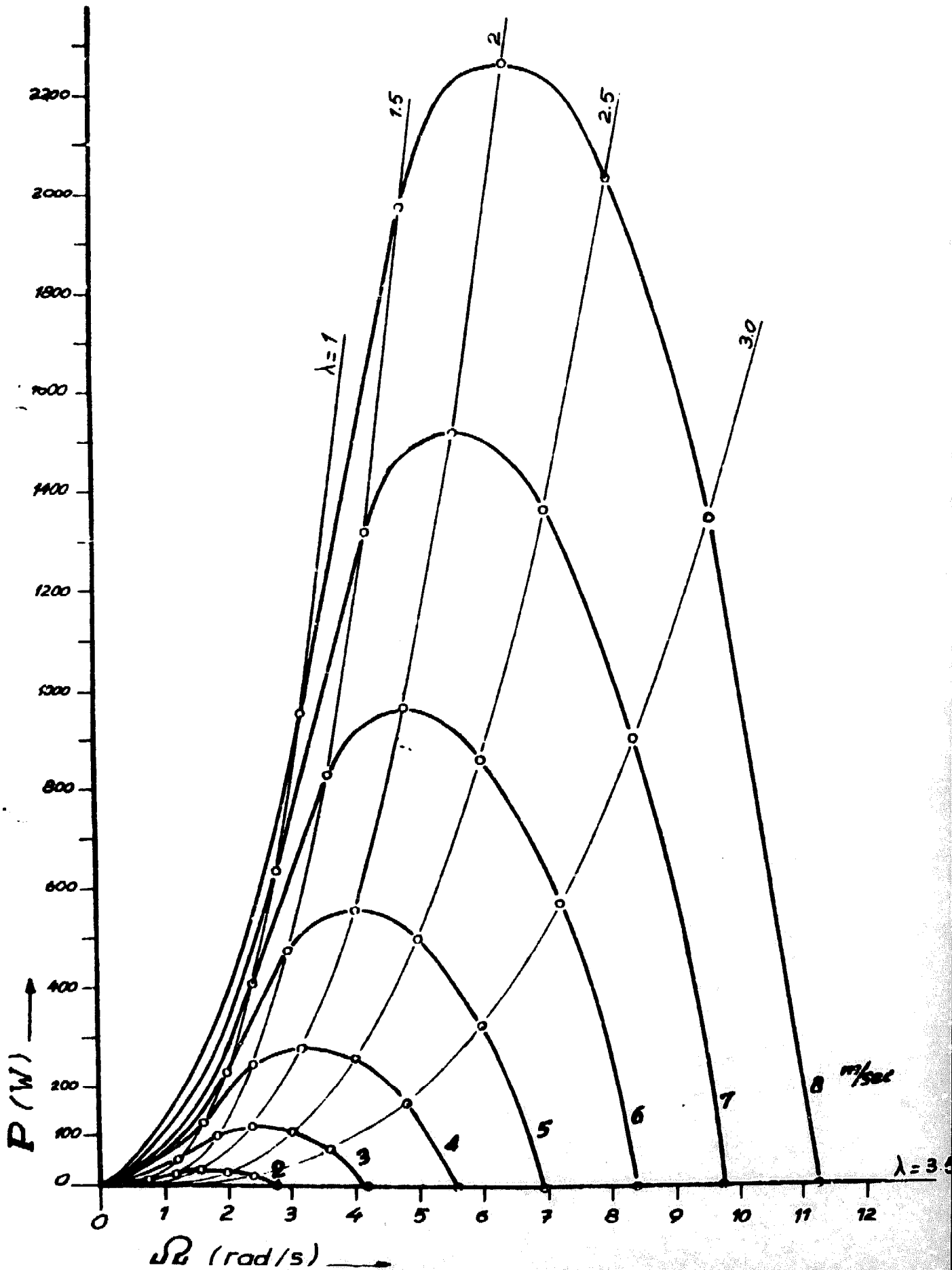
evident that the mechanical losses of power must be taken
into account. An acceptable value of a mechanical efficiency
will be 0.90. Probably a part of mechanical loss is related
to the angular velocity of the system,but is supposed to be
constant in a certain range of R.P.M.-numbers .These assump-
tions led to the approximation of the power-curves of the 5 m.
dia. windmill. The cubic influence of the windspeed is clearly
visible while the most attractive load-curve should cover
the cube-curve for $\lambda = 2$.

4.3. Analysis of the windmill-pistonpump-combination.

Since the R.P.M.-number in relation with the windspeed is
known, the load characteristics of the pump(s) can be drawn
into the former derived powercurves. Considering this it can
be concluded that the transmitted power to the pump depends on
the pump-performance and the crank radius. The considerable
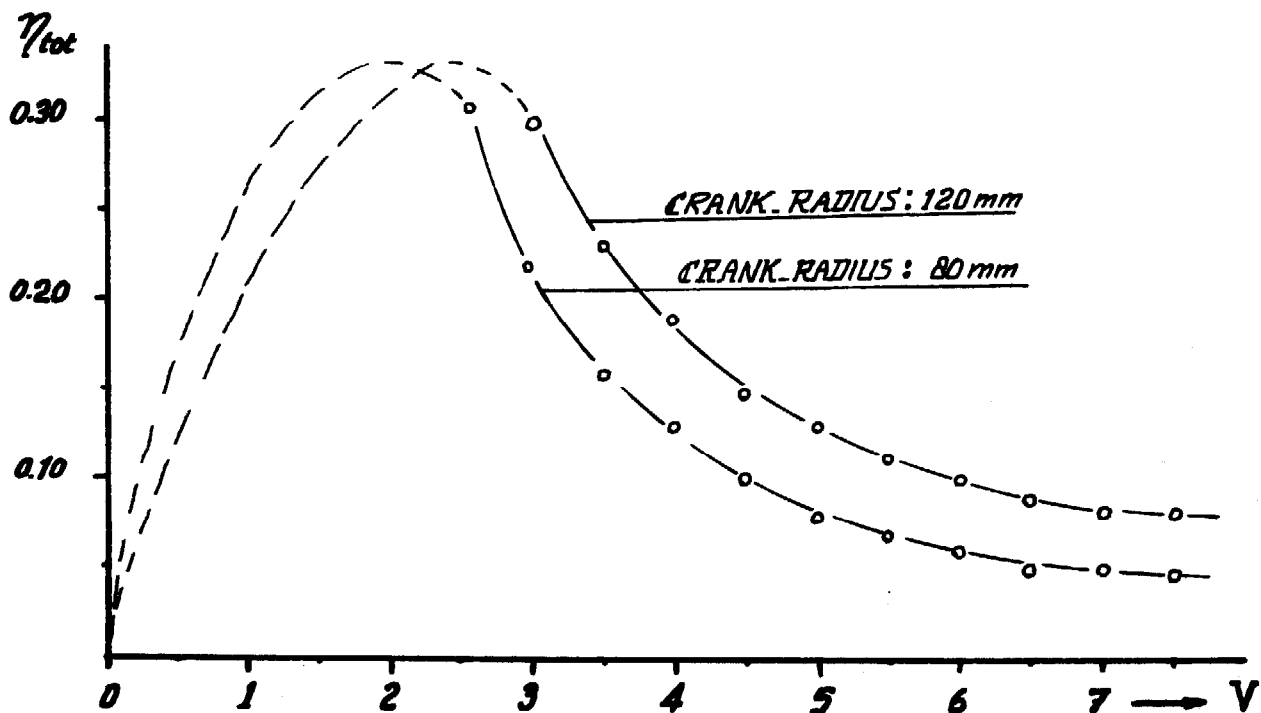
reduction of the overall-efficiency can be explained as
follows: for increasing windvelocities the combination ope-
rates at increasing values of the tip speed ratio so the rate
of the windpowerconversion drops. The mechanical efficiency
and the efficiency of the pistonpump affects the overall-
efficiency as follows:

$$\eta_{tot} = C_{P(\lambda)} \cdot \eta_{mech.} \cdot \eta_{pump}$$

Since the mechanical efficiency is fixed at 0.9 and the C_p-value does not reach values below $0.15 \div 0.20$ the low total efficiencies (e.g. 0.05 for $V = 8$m/sec.) must be caused by a low efficiency of the pump. This assumption can be proved simply by considering the power-curve of the pistonpump.
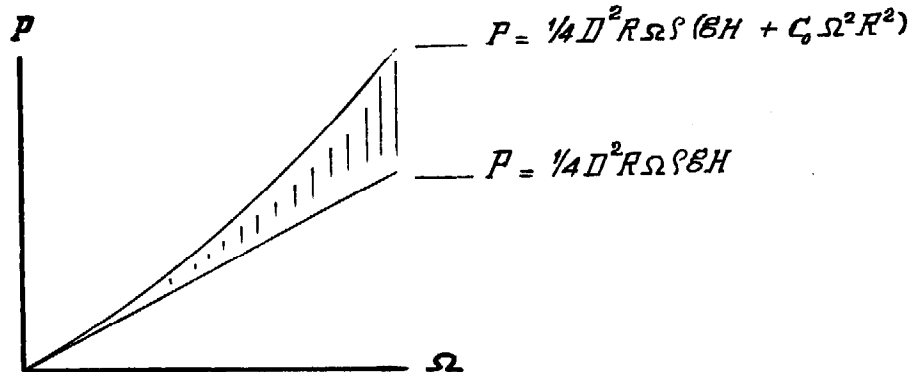
In case of higher angular velocities the losses in power increase considerably due to friction and all kinds of hydraulic losses.

The determinated powercurves can suitably be approached by the following expression:

$$P = \frac{1}{4} D^2 R \Omega \rho (gH + C_o \Omega^2 R^2)$$

Symbols:

P = Power   [Watt]
D = Diameter of piston   [m]
R = Crank radius   [m]
Ω = Angular velocity   [rad/sec]
ρ = Density of water   [kg/m^3]
g = Gravity-acceleration [m/sec^2]
H = Elevation head   [m]

The symbol C_0 stands for the coefficient of loss. As far as just the hydraulic losses are concerned the hydraulic efficiency is defined by:
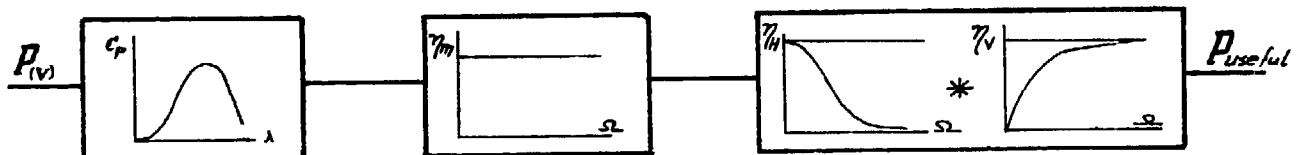
$$\eta_H = \frac{\phi \cdot \rho g H}{\phi \rho (gH + C_o \Omega^2 R^2)} = \frac{1}{1 + C_o \dfrac{\Omega^2 R^2}{gH}}$$

while the total efficiency of the pistonpump is found by:

$$\eta_p = \eta_H * \eta_v$$

(η_v = volumetric efficiency)

In general the total windmill-pistonpump-installation can be represented schematically in case of fixed parameters like elevation head,pistondiameter and crank radius.


From this scheme it can be concluded that the total efficiency is badly affected by the influence of hydraulic efficiency since the measured tipspeed ratio (λ) in practice seldom exceeds the value of 3.0 while C_p maintains reasonable values.

From the two last mentioned measuring-data the value for C_0 can be derived so an approach of powercharacteristics in case of different elevationheads and crank radia will result. For C_0= 200,rather "pessimistic"but acceptable powercurves can be drawn through the powercurves of the windmill in order to define the meetingpoints and the actual angular velocity of the system. Assuming the volumetric efficiency be 0.8 the input-output relation is found by the calculated output for all combinations of elevationhead and crank radia.

Although this determination is not quite valid the results will provide a rather good impression concerning the output which can be expected. One can conclude that in case of higher elevation the output is hardly affected,but the starting wind speed is higher.
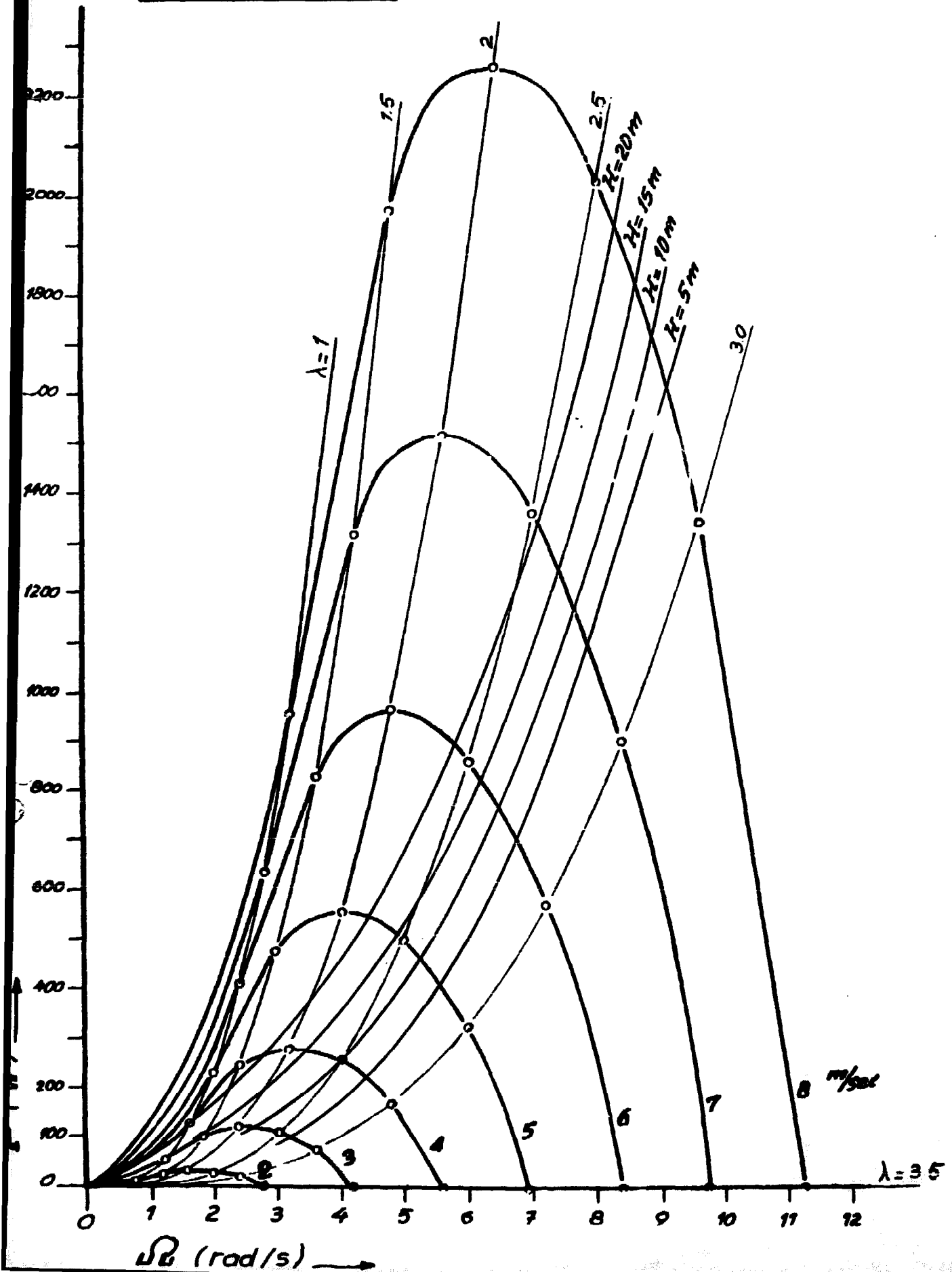
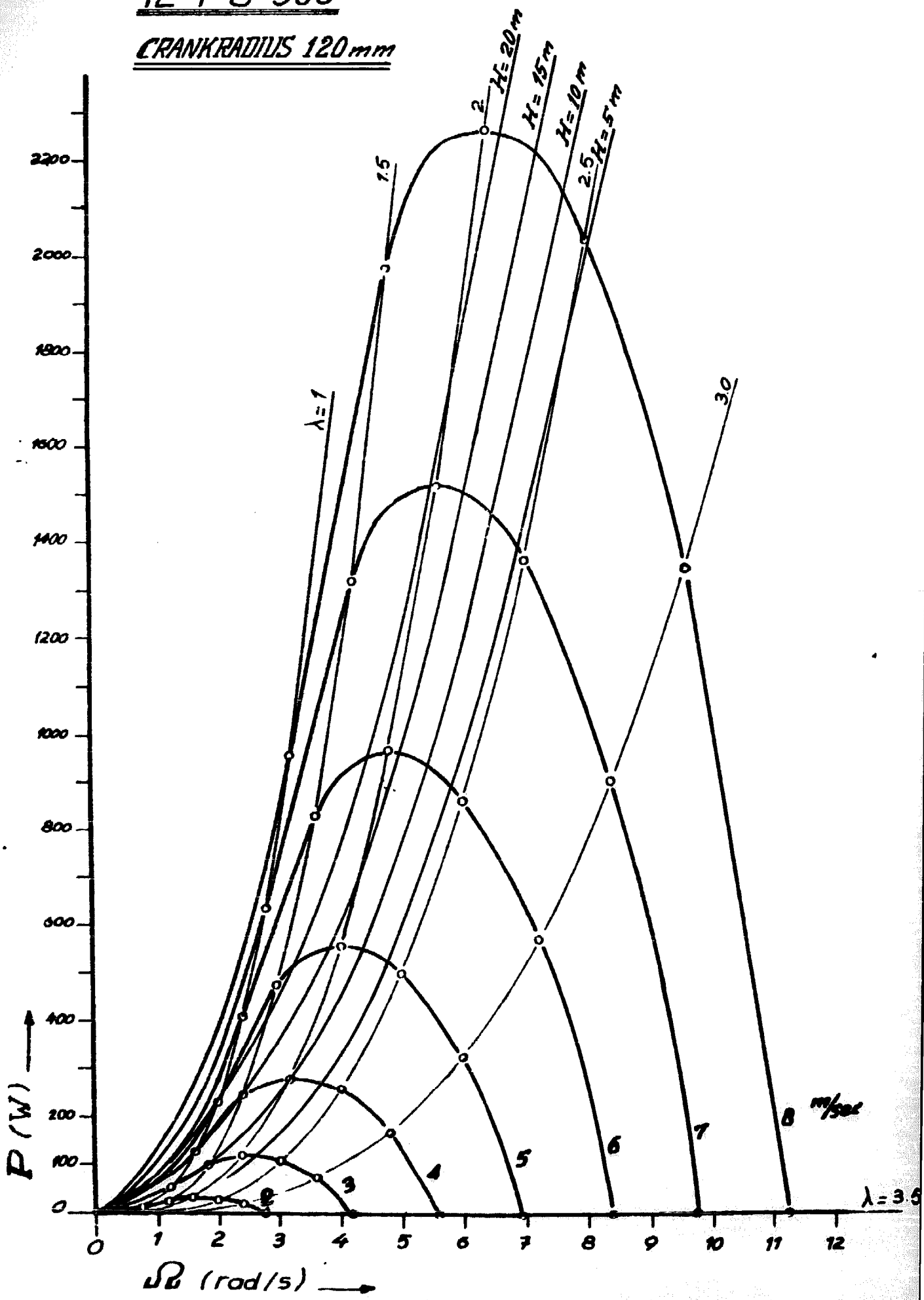4.

12 PU 500

CRANKRADIUS 60 mm.
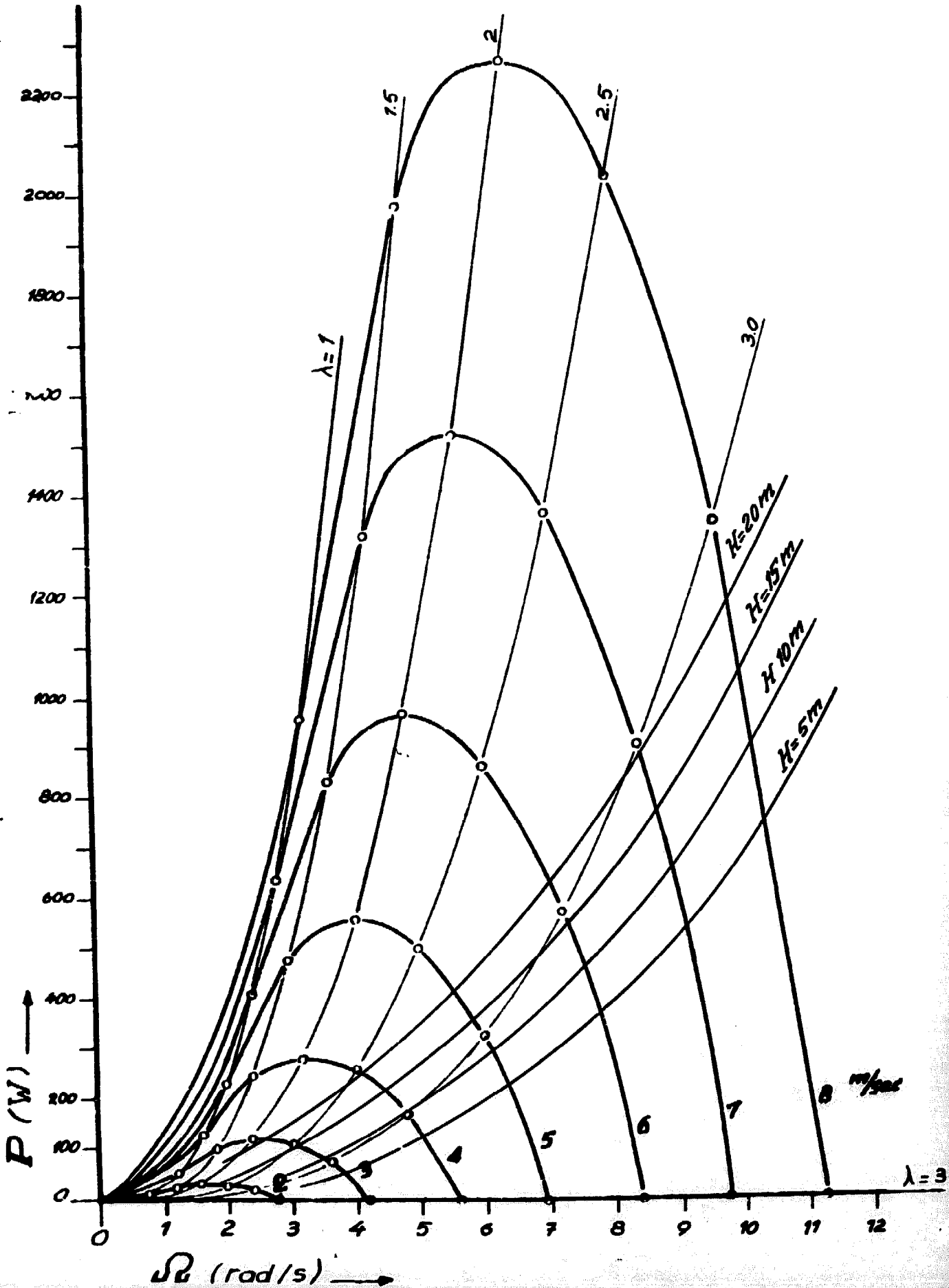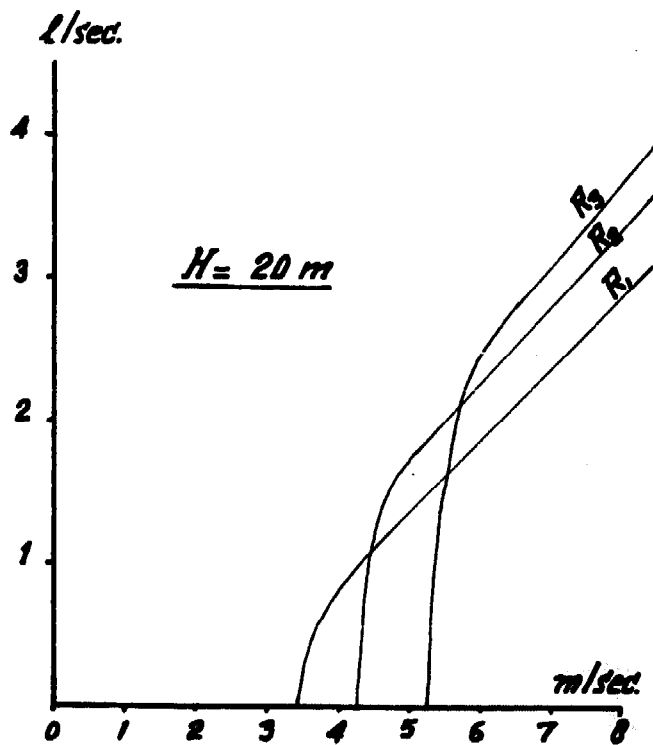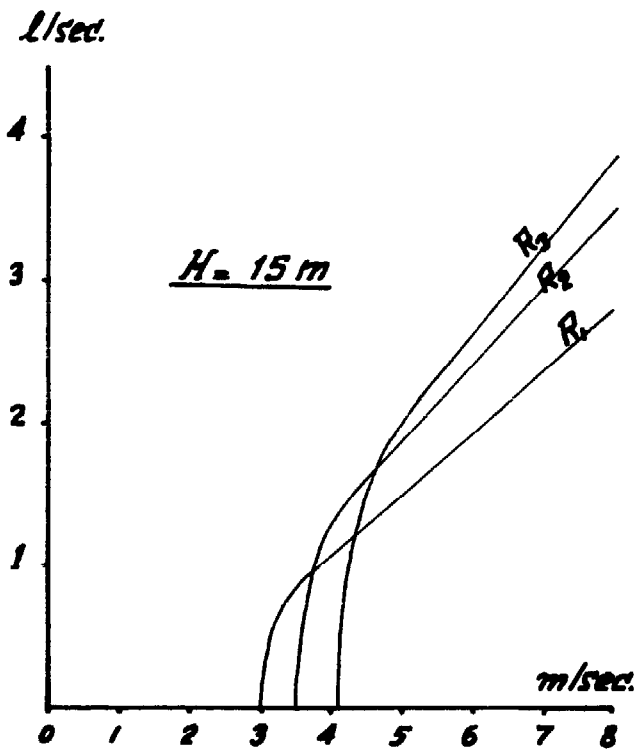
12 PU 500
CRANKRADIUS 90mm

4.13

12 PU 500
CRANKRADIUS 120mm

INPUT - OUTPUT - RELATIONS (150 mm PUMP)
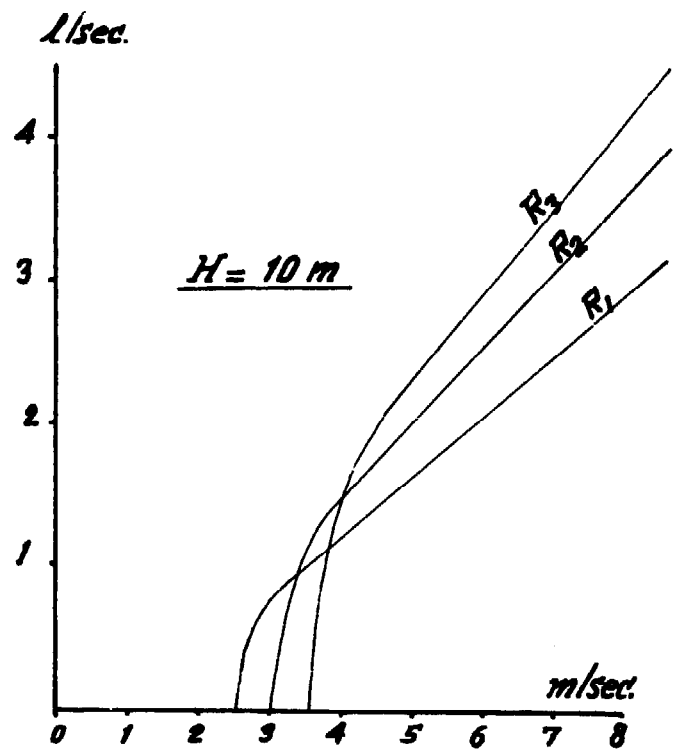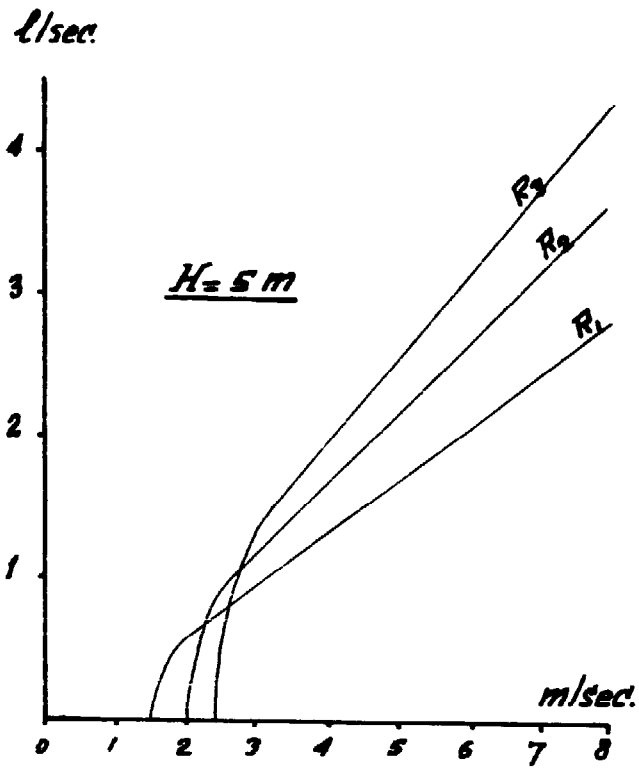
H = ELEVATION HEAD

R₁ = 60 mm
R₂ = 90 mm
R₃ = 120 mm } CRANK RADIUS

5. Defining optimal starting windspeed, windmill- and pump dimensions

In the former chapter the "mis-match" of the windmill and single-acting piston pump has been fully described for the 12 PU 500 windmill. As a matter of fact the same applies for the other members of 12 PU-family (250, 350 and 700) for their $C_p-\lambda$ curves are the same.

Apart from the problem which windmill type should be applied also the starting windspeed to be chosen is most essential with regard to the way the windmill is going to perform. To illustrate this we consider the following extreme situations which often occur in practice:

Too low starting windspeed:

Too small a pump applied. Windmill is almost always running (psychologically attractive). Poor output: too low quantity in the long run.

Too high starting windspeed:

Too big a pump applied. Windmills is almost never running (psychologically unattractive.) Impressive output, but too low a quantity in the long run.
Somewhere in between an optimum exists for which the most optimal starting windspeed occurs for which the best windmill-pump combination can be composed. To derive this optimal combination there is a need for a simple and systematic procedure by which the following can be calculated:

- starting windspeed
- rotor diameter (250-350-500-700)
- pump dimension (3"-4"-5"-6"-8")

Whether such optimizing contradicts for instance social acceptance or required storage capacity cannot be foreseen as yet. However, as all components are available

in a finite number of sizes a certain deviation occurs. The latter provides a margin to create a compromise if necessary.

5.1 Deriving the optimal starting windspeed:

The essential point in power conversion by the windmill, combined with the single acting piston pump, is that the highest efficiency (η) occurs for the optimal (design) windspeed V_0. The starting windspeed V_s is found slightly less. Further the usefull output of the system is almost proportional to the windspeed and as a matter of fact the total output curve can be suitably approached as follows:
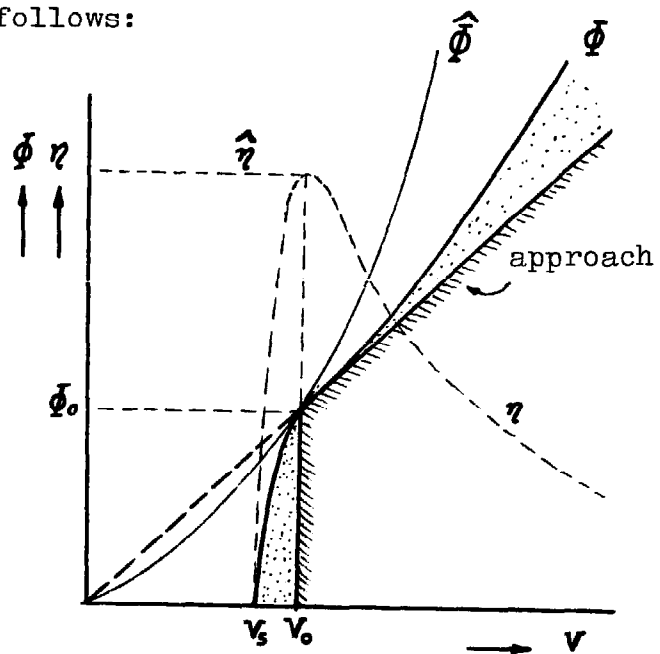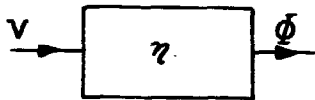
$$V_s \approx V_0$$

$$V < V_0 : \phi = 0$$

$$V = V_0 : \phi = \phi_0$$

$$V \geqslant V_0 : \phi = \frac{\phi_0}{V_0} * V$$

Of course some deviations occur particularly about the interval V_s-V_0 and in case of higher windspeeds but these are supposed to be neglected.

Symbols:

A = rotor area $[m^2]$

Φ = flow (water delivery) $[m^3/s]$

V = actual windspeed $[m/s]$

V_0 = optimal (design) windspeed $[m/s]$

H = elevation head $[m]$

$\hat{\eta}$ = peak-efficiency $[-]$

P = power $[W]$

ρ_a = specific density of air $[kg/m^3]$

ρ = specific density of water $[kg/m^3]$

g = gravity acceleration $[m/s^2]$

P (windmill) $= \eta * \tfrac{1}{2}\rho_a V_0^2 A$

P (pump) $= \rho g H * \Phi_0$

After equalization:

$$\hat{\eta} * \tfrac{1}{2}\rho_a V_0^2 A = \rho g H * \Phi_0$$

for $V = V_0$:

$$\boxed{\Phi_0 = \frac{\hat{\eta} * \tfrac{1}{2}\rho_a V_0^3 * A}{\rho g H}} \quad [m^3/s] \qquad\qquad (5.1)$$

for $V > V_0$:

$$\boxed{\Phi_0 = \frac{\hat{\eta} * \tfrac{1}{2}\rho_a V_0^2 A * V}{\rho g H}} \quad [m^3/s] \qquad\qquad (5.2)$$

for:

$\hat{\eta} = 0.30$ (12 PU Serie)

$\rho_a = 1.3$ kg/m^3

$\rho = 10^3$ kg/m^3

$g = 9.81$ m/s^2

$$\boxed{\Phi = 2.10^{-5}\frac{A}{H}V_0^2 * V} \quad [m^3/s]$$

$$(V \geq V_0)$$

In this expression the influence of elevation head, rotor area and windspeeds are quite evident.

We are interested in a maximum total water delivery during the time period considered $t_1 - t_2$: (e.q. January or a critical season):

$$Q = \int_{t_1}^{t_2} \Phi\,dt = \int_{t_1}^{t_2} 2.10^{-5}\frac{A}{H}V_0^2 * V\,dt = 2.10^{-5}\frac{A}{H}V_0^2\int_{t_1}^{t_2}V\,dt$$

$$(V \geq V_0)$$

It is noted that as far as optimizing is concerned the expression $V_0^2 \int_{t_1}^{t_2} V\,dt$ $(V \geqslant V_0)$ should reach a maximum. However, the actual windspeed is not presented in continuous functions although Histograms provide time durations of windspeeds belonging to certain ranges. Therefore it is advisable to rewrite the expression in order to facilitate an interval-wise calculation as follows:

$$\boxed{V_0^2 \int_{t_1}^{t_2} V\,dt} \qquad \Longrightarrow \qquad \boxed{V_0^2 \sum_{t_1}^{t_2} V_i\, t_i}$$

Windspeed V_i represents the mean windspeed of a certain interval (e.g. 5-6; $V_i = 5.5$) and t_i is the time of which windspeeds, belonging to the interval concerned, actually occur during the time span:
Note that the expression indicates the extent of useful energy conversion. For convenience it is written in short as E.

As a matter of fact the optimizing problem can be suitably shown by means of this picture:
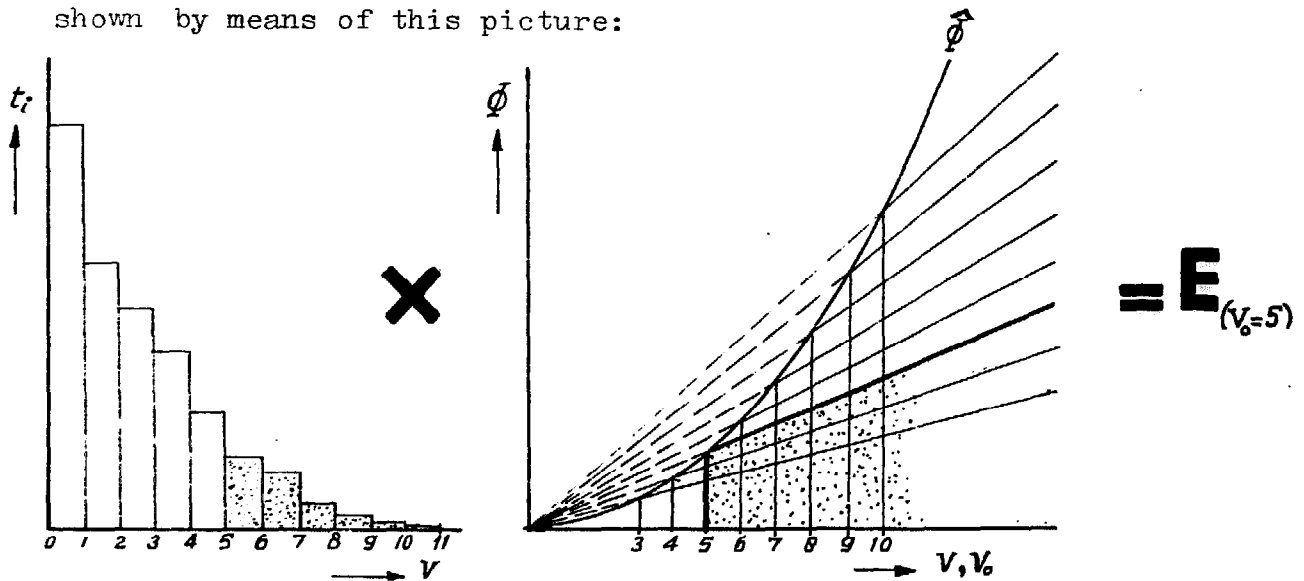
Many boring multiplications carried out again and again for an other starting windspeed V_0 (as a parameter) are needed

to figure out for which value of V_0 a maximum is found for the expression:

$$E = V_0^2 \leq V_i * t_i \quad (V \geq V_0)$$

Such a process can suitably be computerized, but a systematical calculation is simply carried out in a few minutes only without using any advanced calculation machines. This is done table-wise as follows:

V_0	V_i	t_i	$V_i * t_i$	$\leq V_i * t_i$	V_0^2	E
0						
1						
2						
3						
4						
5						
6						
7						
8						
9						
10						
11						
12						

V_0-column

Here the whole numbers of the potential starting windspeeds V_0 are filled out, while these correspond with each start value of the intervals given by the histogram.

The required length of this column depends on relevant duration of intervals of high windspeeds and whether the windmill still functions as the security mechanism might have switched off the windmill. Generally no relevant contribution to the total addition is found for windspeeds which exceed values of 10 up to 12 m/sec.

V_i-column

As mentioned before V_i represents the mean value of the windspeeds which are represented in the interval concerned.

t_i-column

Time duration of windspeeds represented in the interval are preferably filled out in hours.

$V_i * t_i$ -column

Values of the two foregoing columns are multiplied and printed here.

$\sum V_i * t_i$ -column

Here starts a tricky procedure! This column is filled out starting from the bottom adding the value of the foregoing column.

V_o^2-column

Print values of the first column in square.

$E = V_o^2 \sum V_i * t_i$-column

Multiply the figures of the last two columns to find a maximum and so the optimal starting windspeed V_o as a round figure.

Of course a real optimum is found for a value of V_o which is slightly more or less but as mentioned before the components of the windmill installation are bound to certain series of fixed dimensions.
Therefore better continue the calculations with the "round figure" found sofar but knowing where the real optimum exists. (This saves time and calculations.)

real optimum

V_0, starting windspeed
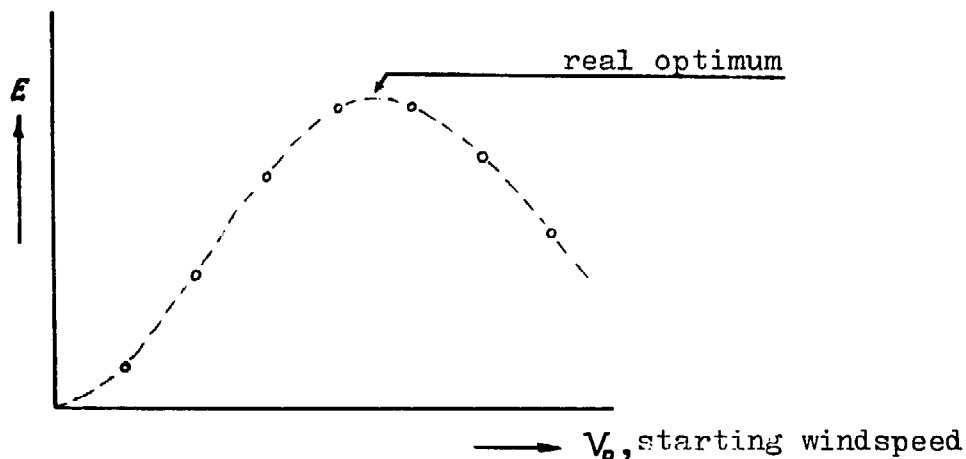
The expected total delivery is calculated as follows:

$$Q_{t_1 \div t_2} = 2 \cdot 10^{-5} \frac{A}{H} * E \quad [m^3] \qquad (5.3)$$

which is only valid if time (t_1) has been expressed in
seconds. In practice winddata are given in hours so far the
practical case the foregoing formule turns into:

$$Q_{t_1 \div t_2} = 7.2 \cdot 10^{-2} \frac{A}{H} * E \quad [m^3] \qquad (5.4)$$

5.2 Deriving required rotor dia:

Depending on water requirement and actual elevationhead
the minimum rotor area is simply derived by applying the
formula:

$$A = \frac{QH}{7.2 \, 10^{-2} \, E} \quad [m^2] \qquad (5.5)$$

But in practice only a few windmill designs are at our
disposal which means that each choice to be made will be
a compromise:

type	rotor area	power
12 PU 250	4.9	P
12 PU 350	9.6	2P
12 PU 500	19.6	4P
12 PU 700	38.5	8P

One should realize that a generous rotor area always pro-
vides some margins which, of course, is more attractive
than a very critically chosen windmill which will hardly
cover the need.

5.3 Pump dimensions

So far we are able to extract the optimal starting windspeed
and windmill type to be applied under certain circumstances
(winds, waterdepth, water requirement). The final match of
pump and windmill should prove the demanded properties of
starting behaviour and output.

To find out the optimal pump dimension the equilibrium is to
be considered for

Windmill: $\quad \Omega_o = \dfrac{\lambda_o V_o}{R}$ $\qquad$ (5.6)

Pump: $\qquad \phi_o = \dfrac{\Omega_o}{2\pi}\, 2r\, \dfrac{\pi}{4} d^2 \eta_v$ $\quad$ (5.7)

$\qquad \Omega_o$ = angular velocity for $V = V_o$ $\qquad \left| 1/s \right|$
$\qquad r$ = crank radius $\qquad \left| m \right|$
$\qquad d$ = pump dia $\qquad \left| m \right|$
$\qquad \eta$ = volumetric efficiency (0,85) $\quad \left[- \right]$
$\qquad \lambda_o$ = tipspeed ratio (2) $\qquad \left[- \right]$
$\qquad R$ = rotor radius $\qquad \left| m \right|$

(5.6) substituted into (5.7):

$$\boxed{\phi_o = \tfrac{1}{2} V_o \dfrac{r}{R} d^2 \eta_v} \quad \left[m^3 \right] \qquad (5.8)$$

In paragraph 5.1 was found for $V = V_o$: $\quad \phi_o = \dfrac{\eta\, \tfrac{1}{2} \rho_a V_o^3 A}{\rho g H}$ $\qquad$ (5.1)

after equalization of : (5.8) and (5.1)

$$\boxed{H = 4.68 * 10^{-5} \dfrac{A}{d^2} \cdot \dfrac{R}{r} * V_o^2} \ \left[m \right] \qquad (5.9)$$

It will be useful to create a graphical representation of
the forementioned equation which is achieved as follows:

H is proportional to V_o^2 while A, d^2 and $\dfrac{R}{r}$ (which are fixed
figures) can be applied as parameters. As well known the
12 PU - windmills have three crank radia(r) so the ratio
leads to:

$$\frac{R}{r_1} = 41.66$$
$$\frac{R}{r_2} = 27.78$$
$$\frac{R}{r_3} = 20.83$$

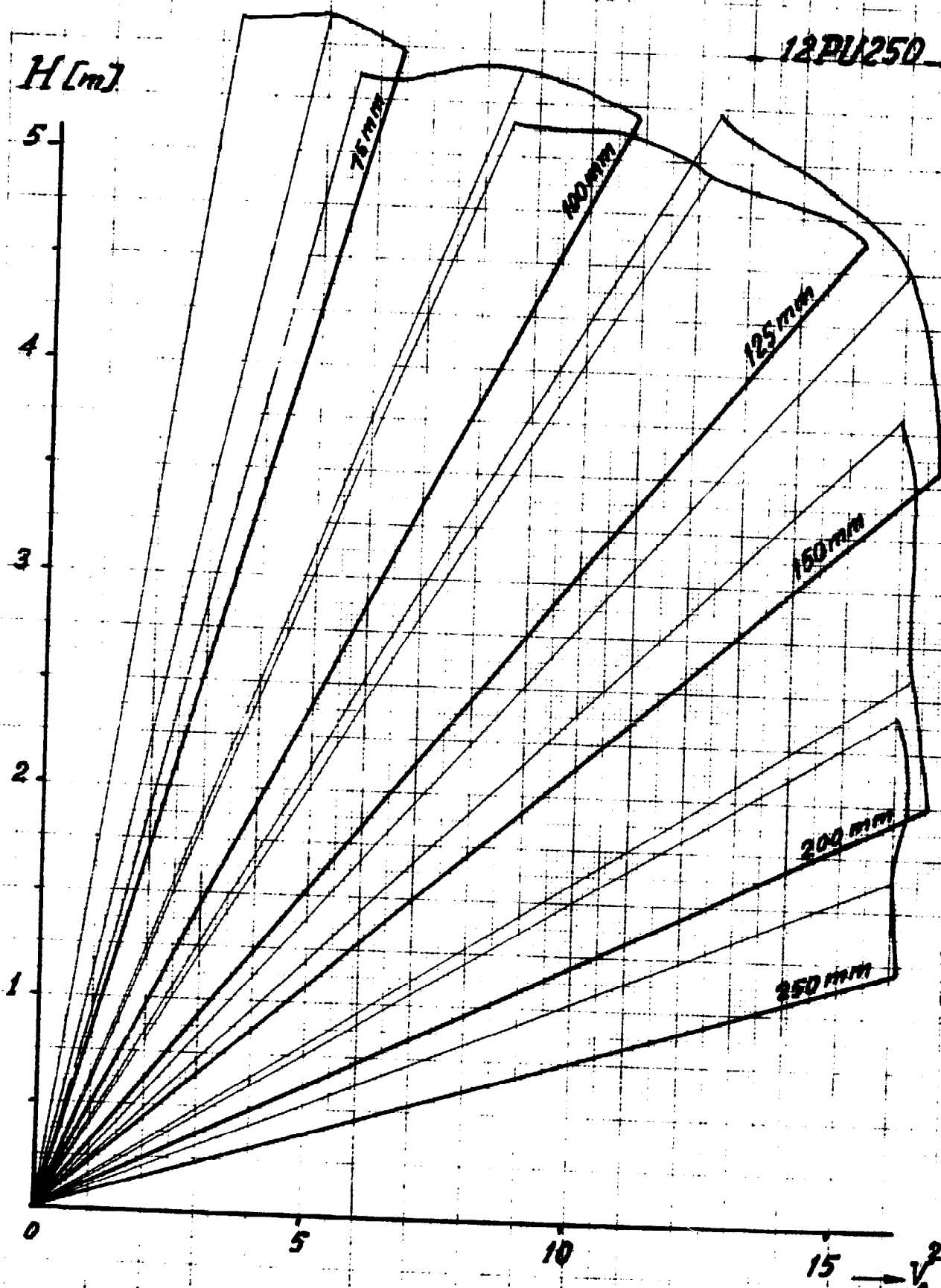The graphical representation will be:
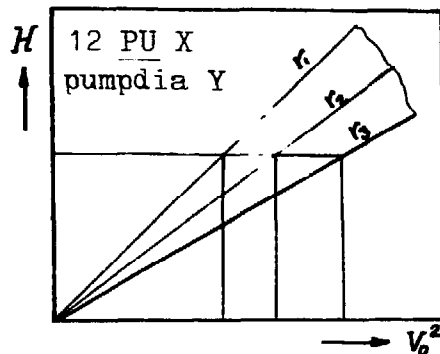
More pump types can be involved,so the H-V_0-plane will contain several overlapping fan-shaped figures representing the pump size and crank-radius concerned.
Though in principle the choice is made for the biggest radius
r_3 an extra possibility is available to give an additional change in the starting windspeed. This is done by re-adjustment of the crank which might be necessary during critical periods.

Next pages show the graphs which are valid for the 12 PU 250-350 and 500 windmills respectively. By means of these graphs the most optimal pump is chosen and of course, most probably, a compromise has to be made. However, in many cases an overcapacity of the windmill will be applied which allows us to deviate from the round figure of V_0 , which was found so far. Preferably the "match point" is chosen which lies just on left side of the V_0 which should be obtained. The reasons for this are:
- A lower starting windspeed proved to be more attractive with regard to social acceptance of the windmill.
- As the windmill is running during a higher percentage of time its delivery will be more regular, which requires a smaller storage tank in case of regular consumption.

5.11

- 12 PU 350 -
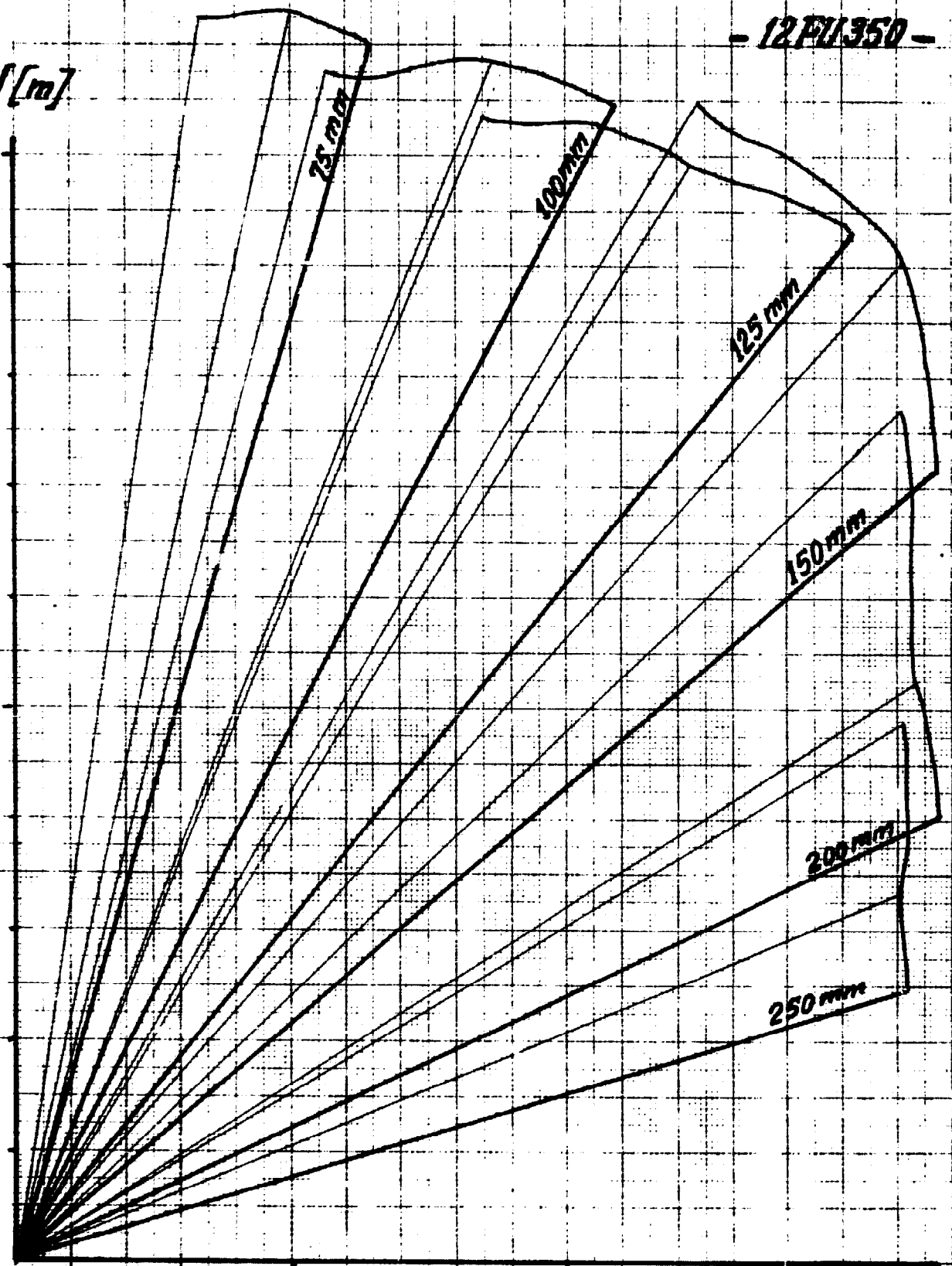
H [m]

10

5

0   5   10   15   → V_o^2

75 mm

100 mm

125 mm

150 mm

200 mm

250 mm

5.12

12 PU500

H [m]

75 mm
100 mm
125 mm
150 mm
200 mm
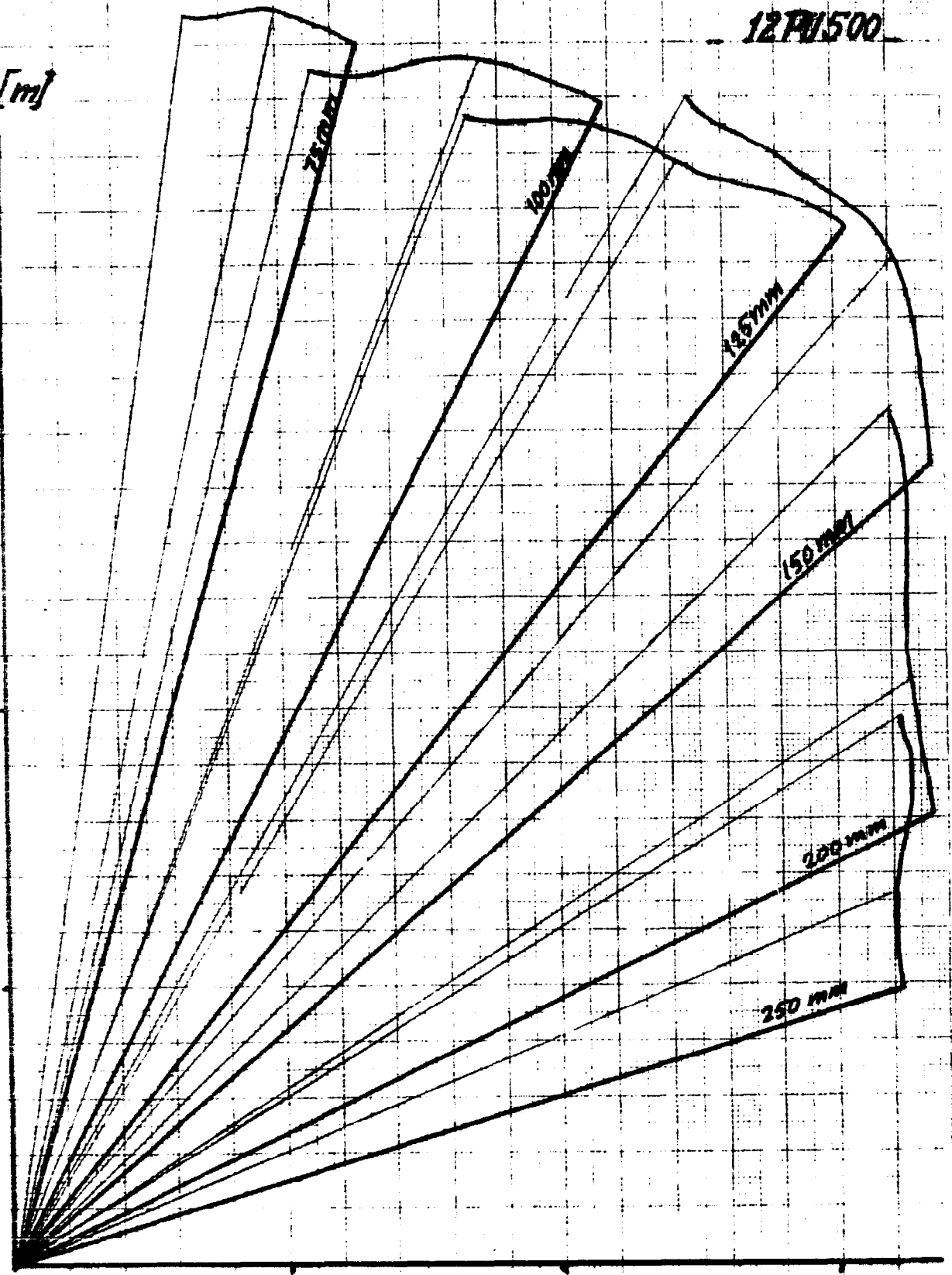250 mm

20

15

10

5

0   5   10   15   → V₀²

- The forces transmitted acting on the moving parts are less
  in case of lower starting windspeeds. To a certain extent
  a less quicker wear of such components and a lower probabi-
  lity of breakdowns occur.
  This might seem contradictional for the windmill has a
  higher degree of running, but such indications have been
  noticed in practice.

5.4 Numerical example

In a certain project area the following winds appear during
the most critical period of water requirement.
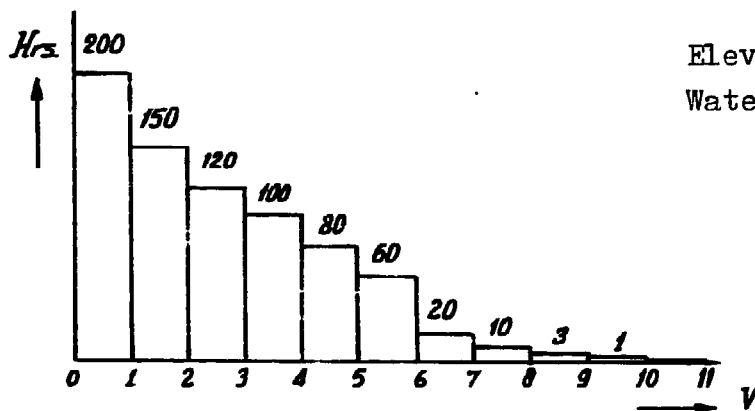
Elevation head = 10 m

Water requirement = 1,500 m^3

1. What is the optimal starting windspeed (round figure)?
2. Which windmill is applied to cover the water requirement?
3a. Which pump dia should be applied to approach the maximum
    output possible?
3b. What will be the starting windspeed?
3c. What is the maximum output to be expected?
3d. How many running- hours and -percentage are expected?
4a. Which minimum pump dia is possible to obtain at least
    the required output and what will be the starting wind-
    speed then?
4b. How many running- hours and -percentage are expected?

1.

V_o	V_i	t_i	$V_i * t_i$	$\sum V_i * t_i$	V_o^2	E
0	0.5	200	-	-	-	-
1	1.5	150	225	1805	1	1,805
2	2.5	120	300	1580	4	6,320
3	3.5	100	350	1280	9	11,520
4	4.5	80	360	930	16	14,880
5	5.5	60	330	570	25	14,250
6	6.5	20	130	240	36	8,640
7	7.5	10	75	110	49	5,390
8	8.5	3	25.5	35	64	2,240
9	9.5	1	9.5	9.5	81	770
10	10.5	-	-	-	100	-
11	11.5	-	-	+	121	-

⟵ maximum (row 4)

The optimal starting windspeed amounts to 4 (round figure)

2. $Q = 7.2 * 10^{-2} * \frac{A}{H} * E = 7.2 * 10^{-2} * \frac{A}{10} * 14880 = 1500 \implies A = 14.0 \ m^2$

$\implies 12PU500$

3a/3b.
By graphs: H = 10m

(pump dia = 150) $V_o^2 = 11.7 \longrightarrow V_o = 3.42 \ m/s$

(pump dia = 200) $V_o^2 = 21^* \longrightarrow V_o = 4.58 \ m/s$

* Actually this figure is not represented at the V_o^2-scale, but the graph reads half the value of V_o^2 for half the value of H.

Though the second match (pump dia = 200) is not desirable as explained before it is considered here to illustrate the consequences of such a choice.

3c. Recalculation of E by means of proportional interpolation.
$E_{(150)} \geqslant 11520 + (14880-11520)(3.42-3) = 12931.2$
$E_{(200)} \geqslant 14250 + (14880-14250)(5-4.58) = 14514.6$
Obviously the second match is closest to the optimum.
$Q_{(150)} \geqslant 7.2 * 10^{-2} * \frac{19.6}{10} * 12931.2 = 1825 \ m^3$
$Q_{(200)} \geqslant 7.2 * 10^{-2} * \frac{19.6}{10} * 14514.6 = 2048 \ m^3$

3d. <u>Total hours of functioning:</u> (from table)

$T_{(150)}$ = 1+3+10+20+60+80+(4-3.42)100 = 232 *hrs.*

$T_{(200)}$ = 1+3+10+20+60+(5-4.38)80 = 127.6 *hrs.*

<u>Percentages:</u>

$P_{(150)} = \frac{232}{744}$ * 100% = 31%

$P_{(200)} = \frac{127.6}{744}$ * 100% = 17%

4a. $Q = 7.2 * 10^{-2} * \frac{A}{H} * E = 7.2 * 10^{-2} * \frac{19.6}{10} * E = 1500$

$E = 10629 \implies V_o^2 \approx 9$ (see table)

pump dia = 125 mm (see graphs)

with V_o^2 = 8.2 and V_o = 2.86 m/s

<u>Recalculation for E:</u> (from table)

$E_{(125)} \geqslant 6320 + (11520-6320)(2.86-2) = 10792$

This is slightly more than was demanded:

$Q_{(125)} \geqslant 7.2 * 10^{-2} * \frac{19.6}{10} * 10792 = 1523$ m^3

A 125 mm pump will serve the purpose so the starting windspeed of 2.86 m/s is confirmed.

4b. <u>Running hours:</u> (from table)

$T_{(125)}$ = 1+3+10+20+60+80+100+120*(3-2.86) = 288.4 hrs

<u>Running percentage:</u>

$P_{(125)} = \frac{288.4}{744}$ * 100% = 39%