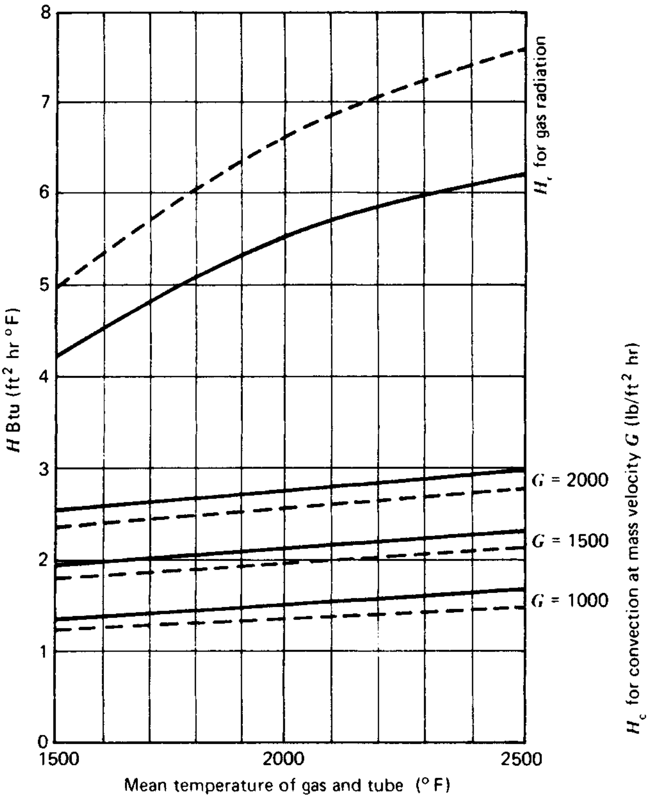

Fig. 45.31 Gas radiation (H_r) and convection (H_c) coefficients for flue gas inside radiant tubes.[1]

perature. The gas radiation factor depends on temperature and inside diameter. The effect of flame luminosity has not been considered.

45.9 FLUID FLOW

Fluid flow problems of interest to the furnace engineer include the resistance to flow of air or flue gas, over a range of temperatures and densities through furnace ductwork, stacks and flues, or recuperators and regenerators. Flow of combustion air and fuel gas through distribution piping and burners will also be considered. Liquid flow, of water and fuel oil, must also be evaluated in some furnace designs but will not be treated in this chapter.

To avoid errors resulting from gas density at temperature, velocities will be expressed as mass velocities in units of $G = \text{lb/hr} \cdot \text{ft}^2$. Because the low pressure differentials in systems for flow of air or flue gas are usually measured with a manometer, in units of inches of water column (in. H_2O), that will be the unit used in the following discussion.

The relation of velocity head h_v in in. H_2O to mass velocity G is shown for a range of temperatures in Fig. 45.32. Pressure drops as multiples of h_v are shown, for some configurations used in furnace design, in Figs. 45.33 and 45.34. The loss for flow across tube banks, in multiples of the velocity head, is shown in Fig. 45.35 as a function of the Reynolds number.

The Reynolds number Re is a dimensionless factor in fluid flow defined as $\text{Re} = DG/\mu$, where D is inside diameter or equivalent dimension in feet, G is mass velocity as defined above, and μ is viscosity as shown in Fig. 45.9. Values for Re for air or flue gas, in the range of interest, are shown in Fig. 45.36. Pressure drop for flow through long tubes is shown in Fig. 45.37 for a range of Reynolds numbers and equivalent diameters.

45.9.1 Preferred Velocities

Mass velocities used in contemporary furnace design are intended to provide an optimum balance between construction costs and operating costs for power and fuel; some values are listed on the next page:

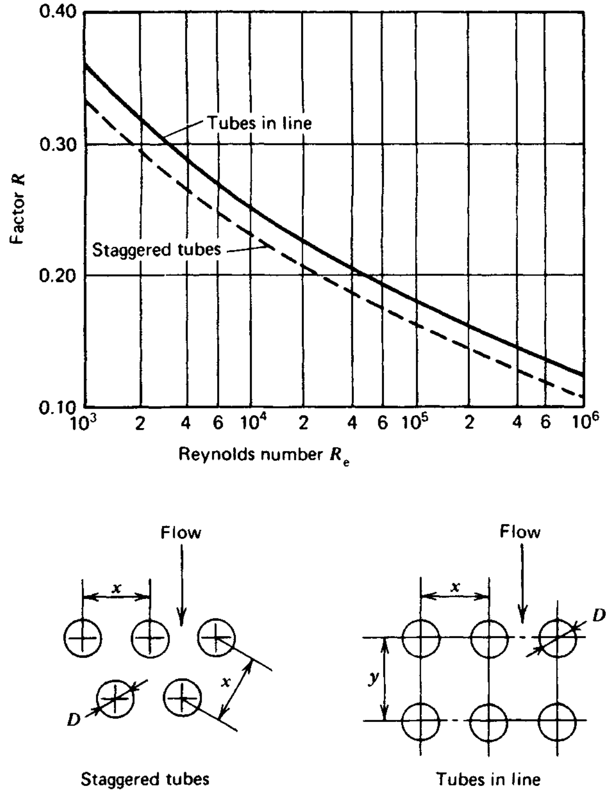

Fig. 45.32 Heat loss for flow of air or flue gas across tube banks at atmospheric pressure (velocity head) × F × R.

Medium	Mass Velocity G	Velocity Head (in. H_2O)
Cold air	15,000	0.7
800°F air	10,000	0.3
2200°F flue gas	1,750	0.05
1500°F flue gas	2,000	0.05

The use of these factors will not necessarily provide an optimum cost balance. Consider a furnace stack of self-supporting steel construction, lined with 6 in. of gunned insulation. For $G = 2000$ and $h_v = 0.05$ at 1500°F, an inside diameter of 12 ft will provide a flow of 226,195 lb/hr. To provide a net draft of 1 in. H_2O with stack losses of about 1.75 h_v or 0.0875 in., the effective height from Fig. 45.38 is about 102 ft. By doubling the velocity head to 0.10 in. H_2O, G at 1500°F becomes 3000. For the same mass flow, the inside diameter is reduced to 9.8 ft. The pressure drop through the stack increases to about 0.175 in., and the height required to provide a net draft of 1 in. increases to about 110 ft. The outside diameter area of the stack is reduced from 4166 ft^2 to $11 \times 3.1416 \times 110 = 3801$ ft^2. If the cost per square foot of outside surface is the same for both cases, the use of a higher stack velocity will save construction costs. It is accordingly recommended that specific furnace designs receive a more careful analysis before selecting optimum mass velocities.

Stack draft, at ambient atmospheric temperature of 70°F, is shown in Fig. 45.38 as a function of flue gas temperature. Where greater drafts are desirable with a limited height of stack, a jet-type stack can be used to convert the momentum of a cold air jet into stack draft. Performance data are available from manufacturers.

45.9.2 Centrifugal Fan Characteristics

Performance characteristics for three types of centrifugal fans are shown in Fig. 45.39. More exact data are available from fan manufacturers. Note that the backward curved blade has the advantage

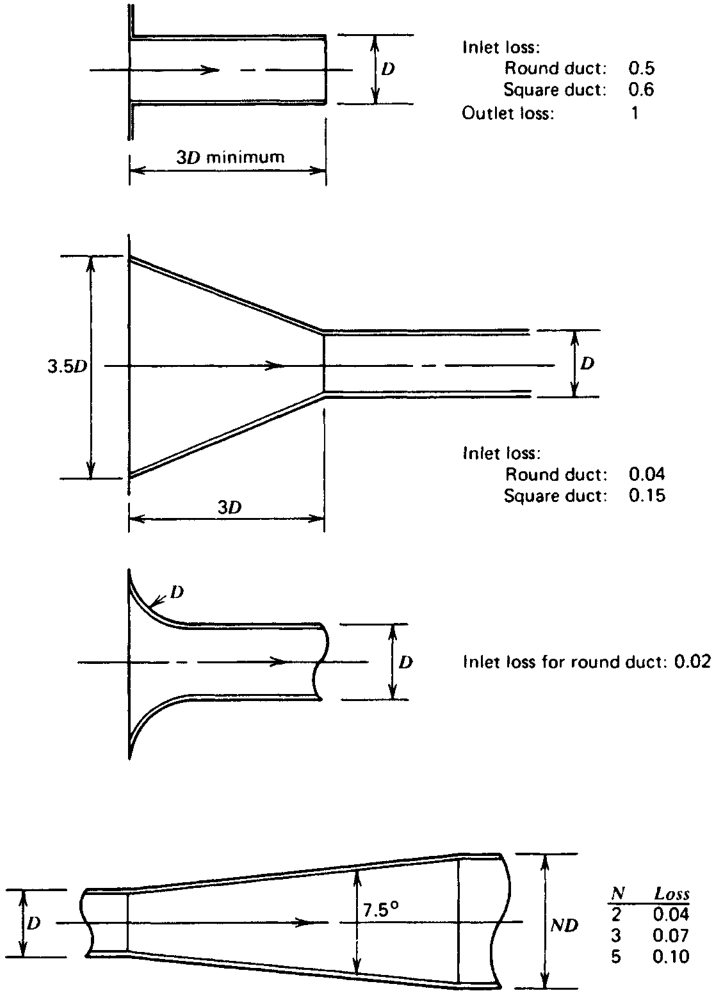

Fig. 45.33 Pressure drop in velocity heads for flow of air or flue gas through entrance configurations or expansion sections.[1]

of limited horsepower demand with reduced back pressure and increasing volume, and can be used where system resistance is unpredictable. The operating point on the pressure–volume curve is determined by the increase of duct resistance with flow, matched against the reduced outlet pressure, as shown in the upper curve.

45.9.3 Laminar and Turbulent Flows

The laminar flow of a fluid over a boundary surface is a shearing process, with velocity varying from zero at the wall to a maximum at the center of cross section or the center of the top surface for liquids in an open channel. Above a critical Reynolds number, between 2000 and 3000 in most cases, flow becomes a rolling action with a uniform velocity extending almost to the walls of the duct, and is identified as turbulent flow.

With turbulent flow the pressure drop is proportional to D; the flow in a large duct can be converted from turbulent to laminar by dividing the cross-sectional area into a number of parallel channels. If flow extends beyond the termination of these channels, the conversion from laminar to turbulent flow will occur over some distance in the direction of flow.

Radial mixing with laminar flow is by the process of diffusion, which is the mixing effect that occurs in a chamber filled with two different gases separated by a partition after the partition is removed. Delayed mixing and high luminosity in the combustion of hydrocarbon gases can be ac-

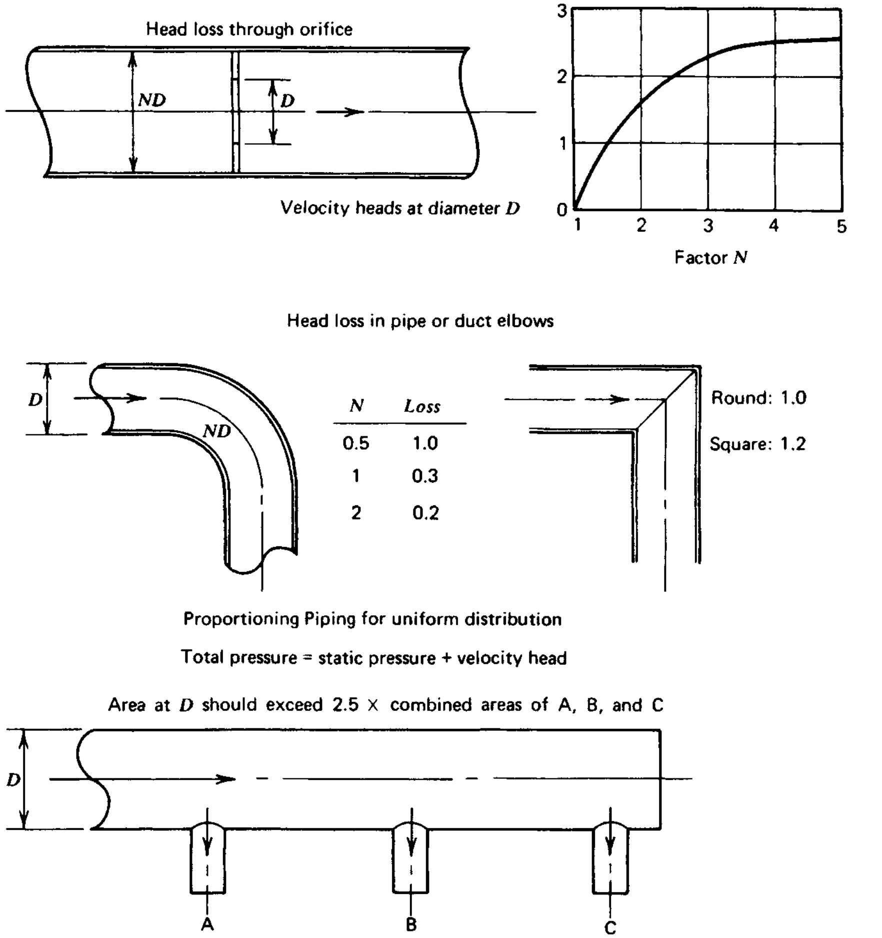

Fig. 45.34 Pressure drop in velocity heads for flow of air or flue gas through orifices, elbows, and lateral outlets.[1]

Staggered Tubes		Tubes in Line		Factor F for x/D		
x/D	Factor F	y/D	1.5	2	3	4
1.5	2.00	1.25	1.184	0.576	0.334	0.268
2	1.47	1.5	1.266	0.656	0.387	0.307
3	1.22	2	1.452	0.816	0.497	0.390
4	1.14	3	1.855	1.136	0.725	0.572
		4	2.273	1.456	0.957	0.761

complished by "diffusion combustion," in which air and fuel enter the combustion chamber in parallel streams at equal and low velocity.

45.10 BURNER AND CONTROL EQUIPMENT

With increasing costs of fuel and power, the fraction of furnace construction and maintenance costs represented by burner and control equipment can be correspondingly increased. Burner designs should be selected for better control of flame pattern over a wider range of turndown and for complete combustion with a minimum excess air ratio over that range.

Furnace functions to be controlled, manually or automatically, include temperature, internal pressure, fuel/air ratio, and adjustment of firing rate to anticipated load changes. For intermittent operation, or for a wide variation in required heating capacity, computer control may be justified to

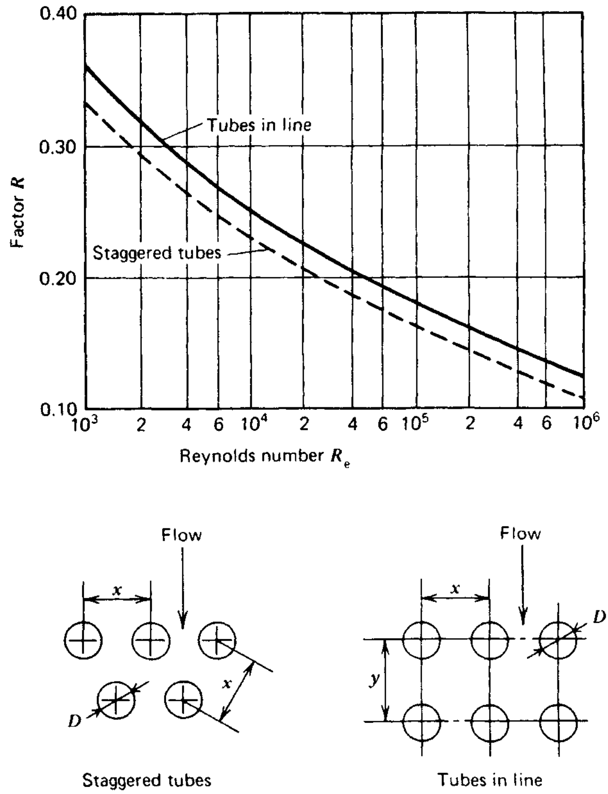

Fig. 45.35 Pressure drop factors for flow of air or flue gas through tube banks.[1]

Staggered Tubes		Tubes in Line		Factor F for x/D		
x/D	Factor F	y/D	1.5	2	3	4
1.5	2.00	1.25	1.184	0.576	0.334	0.268
2	1.47	1.5	1.266	0.656	0.387	0.307
3	1.22	2	1.452	0.816	0.497	0.390
4	1.14	3	1.855	1.136	0.725	0.572
		4	2.273	1.456	0.957	0.761

anticipate required changes in temperature setting and firing rates, particularly in consecutive zones of continuous furnaces.

45.10.1 Burner Types

Burners for gas fuels will be selected for the desired degree of premixing of air and fuel, to control flame pattern, and for the type of flame pattern, compact and directional, diffuse or flat flame coverage of adjacent wall area. Burners for oil fuels, in addition, will need provision for atomization of fuel oil over the desired range of firing rates.

The simplest type of gas burner comprises an opening in a furnace wall, through which combustion air is drawn by furnace draft, and a pipe nozzle to introduce fuel gas through that opening. Flame pattern will be controlled by gas velocity at the nozzle and by excess air ratio. Fuel/air ratio will be manually controlled for flame appearance by the judgment of the operator, possibly supplemented by continuous or periodic flue gas analysis. In regenerative furnaces, with firing ports serving alternately as exhaust flues, the open pipe burner may be the only practical arrangement.

For one-way fired furnaces, with burner port areas and combustion air velocities subject to control, fuel/air ratio control can be made automatic over a limited range of turndown with several systems, including:

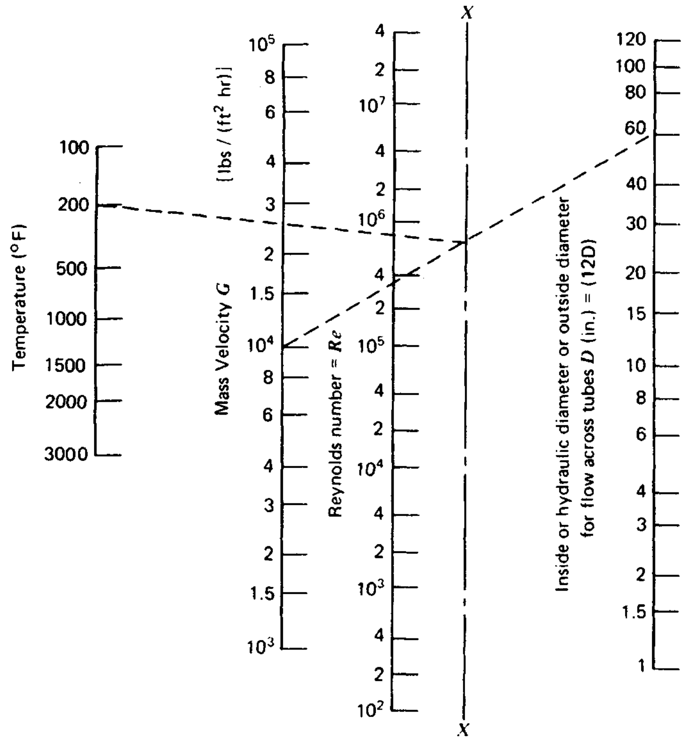

Fig. 45.36 Reynolds number (Re) for flow of air or flue gas through tubes or across tube banks.[1]

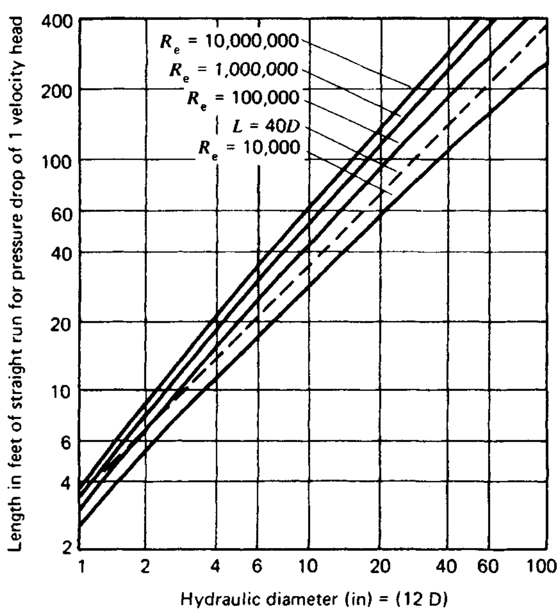

Fig. 45.37 Length in feet for pressure drop of one velocity head, for flow of air or flue gas, as a function of Re and D.[1]

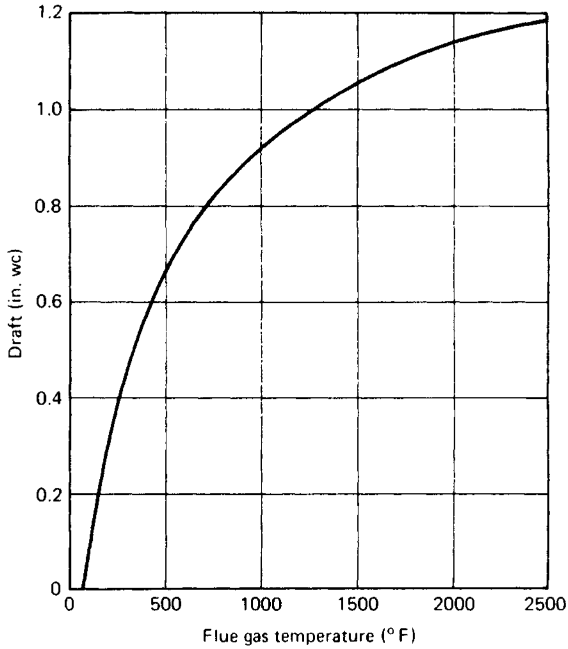

Fig. 45.38 Stack draft for ambient T_g = 70°F and psia = 14.7 lb/in.^{2}.[1]

Mixing in venturi tube, with energy supplied by gas supply inducing atmospheric air. Allows simplest piping system with gas available at high pressure, as from some natural gas supplies.

Venturi mixer with energy from combustion air at intermediate pressure. Requires air supply piping and distribution piping from mixing to burners.

With both combustion air and fuel gas available at intermediate pressures, pressure drops through adjustable orifices can be matched or proportioned to hold desired flow ratios. For more accurate control, operation of flow control valves can be by an external source of energy.

Proportioning in venturi mixers depends on the conservation of momentum—the product of flow rate and velocity or of orifice area and pressure drop. With increased back pressure in the combustion chamber, fuel/air ratio will be increased for the high pressure gas inspirator, or decreased with air pressure as the source of energy, unless the pressure of the induced fluid is adjusted to the pressure in the combustion chamber.

The arrangement of a high-pressure gas inspirator system is illustrated in Fig. 45.40. Gas enters the throat of the venturi mixer through a jet on the axis of the opening. Air is induced through the surrounding area of the opening, and ratio control can be adjusted by varying the air inlet opening by a movable shutter disk. A single inspirator can supply a number of burners in one firing zone, or a single burner.

For the air primary mixing system, a representative arrangement is shown in Fig. 45.41. The gas supply is regulated to atmospheric, or to furnace gas pressure, by a diaphragm-controlled valve. Ratio control is by adjustment of an orifice in the gas supply line. With air flow the only source of energy, errors in proportioning can be introduced by friction in the gas-pressure control valve. Each mixer can supply one or more burners, representing a control zone.

With more than one burner per zone, the supply manifold will contain a combustible mixture that can be ignited below a critical port velocity to produce a backfire that can extinguish burners and possibly damage the combustion system. This hazard has made the single burner per mixer combination desirable, and many contemporary designs combine mixer and burner in a single structure.

With complete premixing of fuel and air, the flame will be of minimum luminosity, with combustion complete near the burner port. With delayed mixing, secured by introducing fuel and air in separate streams, through adjacent openings in the burner, or by providing a partial premix of fuel with a fraction of combustion air, flame luminosity can be controlled to increase flame radiation.

In a burner providing no premix ahead of the combustion chamber, flame pattern is determined by velocity differentials between air and fuel streams, and by the subdivision of air flow into several

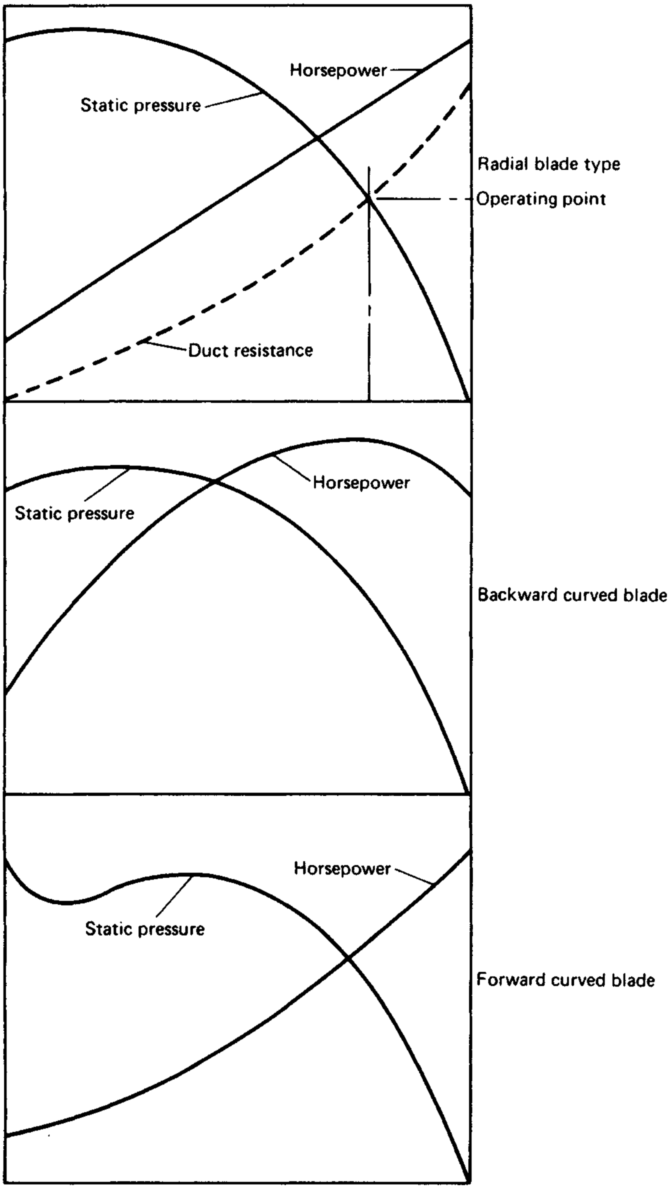

Fig. 45.39 Centrifugal fan characteristics.[1]

parallel streams. This type of burner is popular for firing with preheated combustion air, and can be insulated for that application.

Partial premix can be secured by dividing the air flow between a mixing venturi tube and a parallel open passage.

With the uncertainty of availability of contemporary fuel supplies, dual fuel burners, optionally fired with fuel gas or fuel oil, can be used. Figure 45.42 illustrates the design of a large burner for firing gas or oil fuel with preheated air. For oil firing, an oil-atomizing nozzle is inserted through the gas tube. To avoid carbon buildup in the oil tube from cracking of residual oil during gas firing, the oil tube assembly is removable.

Oil should be atomized before combustion in order to provide a compact flame pattern. Flame length will depend on burner port velocity and degree of atomization. Atomization can be accom-

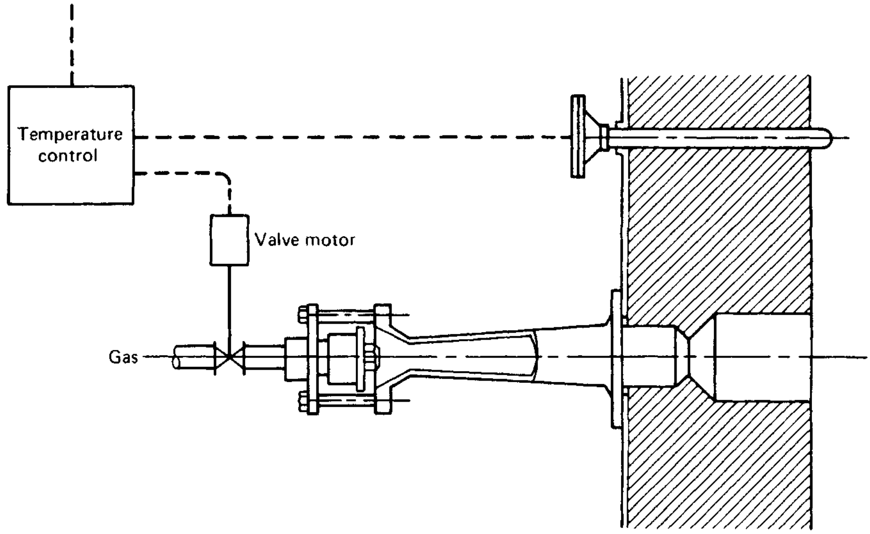

Fig. 45.40 Air/gas ratio control by high-pressure gas inspirator.[1]

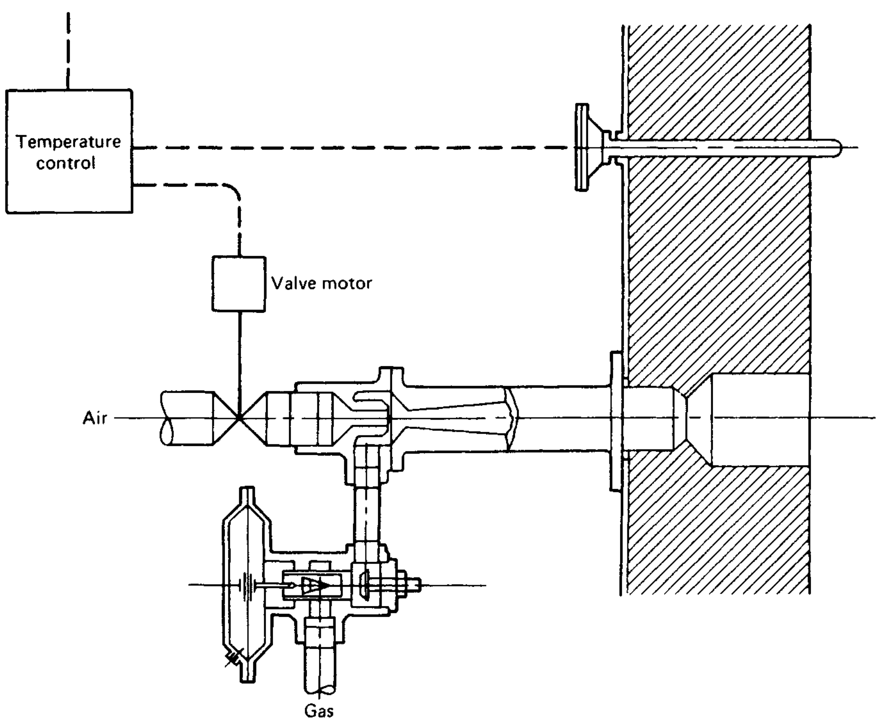

Fig. 45.41 Air/gas ratio control by air inspirator.[1]

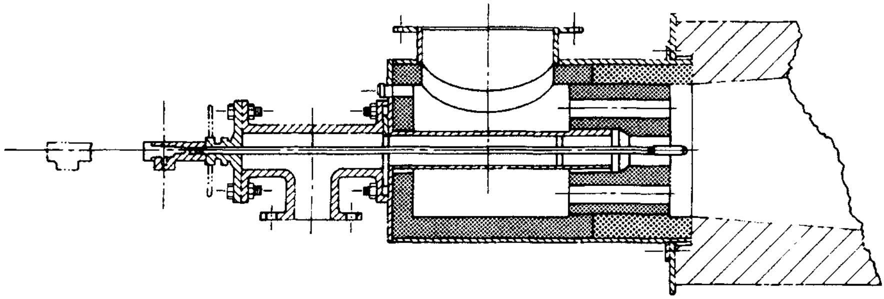

Fig. 45.42 Dual fuel burner with removable oil nozzle.[1] (Courtesy Bloom Engineering Company.)

plished by delivery of oil at high pressure through a suitable nozzle; by intermediate pressure air, part or all of the combustion air supply, mixing with oil at the discharge nozzle; or by high-pressure air or steam. For firing heavy fuel oils of relatively high viscosity, preheating in the storage tank, delivery to the burner through heated pipes, and atomization by high-pressure air or steam will be needed. If steam is available, it can be used for both tank and pipe heating and for atomization. Otherwise, the tank and supply line can be electrically heated, with atomization by high-pressure air.

45.10.2 Burner Ports

A major function of fuel burners is to maintain ignition over a wide range of demand and in spite of lateral drafts at the burner opening. Ignition can be maintained at low velocities by recirculation of hot products of combustion at the burner nozzle, as in the bunsen burner, but stability of ignition is limited to low port velocities for both the entering fuel/air mixture and for lateral drafts at the point of ignition. Combustion of a fuel/air mixture can be catalyzed by contact with a hot refractory surface. A primary function of burner ports is to supply that source of ignition. Where combustion of a completely mixed source of fuel and air is substantially completed in the burner port, the process is identified as "surface combustion." Ignition by contact with hot refractory is also effective in flat flame burners, where the combustion air supply enters the furnace with a spinning motion and maintains contact with the surrounding wall.

Burner port velocities for various types of gas burners can vary from 3000 to 13,000 lb/hr · ft^2, depending on the desired flame pattern and luminosity. Some smaller sizes of burners are preassembled with refractory port blocks.

45.10.3 Combustion Control Equipment

Furnace temperature can be measured by a bimetallic thermocouple inserted through the wall or by an optical sensing of radiation from furnace walls and products of combustion. In either case, an electrical impulse is translated into a temperature measurement by a suitable instrument and the result indicated by a visible signal and optionally recorded on a moving chart. For automatic temperature control, the instrument reading is compared to a preset target temperature, and the fuel and air supply adjusted to match through a power-operated valve system.

Control may be on–off, between high and low limits; three position, with high, normal, and off valve openings; or proportional with input varying with demand over the full range of control. The complexity and cost of the system will, in general, vary in the same sequence. Because combustion systems have a lower limit of input for proper burner operation or fuel/air ratio control, the proportioning temperature control system may cut off fuel input when it drops to that limit.

Fuel/air ratios may be controlled at individual burners by venturi mixers or in multiple burner firing zones by similar mixing stations. To avoid back firing in burner manifolds, the pressures of air and gas supplies can be proportioned to provide the proper ratio of fuel and air delivered to individual burners through separate piping. Even though the desired fuel/air ratio can be maintained for the total input to a multiple burner firing zone, errors in distribution can result in excess air or fuel being supplied to individual burners. The design of distribution piping, downstream from ratio control valves, will control delayed combustion of excess fuel and air from individual burners.

In batch-type furnaces for interrupted heating cycles, it may be advantageous to transfer temperature control from furnace temperature to load temperature as load temperature approaches the desired level, in order to take advantage of higher furnace temperatures in the earlier part of the heating cycle. An example is a furnace for annealing steel strip coils. Because heat flow through coil laminations is a fraction of that parallel to the axis of the coil, coils may be stacked vertically with open coil separators between them, to provide for heat transfer from recirculated furnace atmosphere to the end surfaces of coils. For bright annealing, the furnace atmosphere will be nonoxidizing, and the load will be enclosed in an inner cover during heating and cooling, with the atmosphere recirculated by a centrifugal fan in the load support base, to transfer heat from the inner cover to end faces of coils. There will also be some radiation heat transfer from the inner cover to the cylindrical surface of the coil stack.

Inner covers are usually constructed of heat-resisting alloy, with permissible operating temperatures well above the desired final load temperature. A preferred design provides for initial control of furnace inside wall temperature from a thermocouple inserted through the furnace wall, with control switched to a couple in the support base, in control with the bottom of the coil stack, after load temperature reaches a present level below the desired final temperature.

To avoid leakage of combustion gases outward through furnace walls, with possible overheating of the steel enclosure, or infiltration of cold air that could cause nonuniform wall temperatures, control of internal furnace pressure to slightly above ambient is desirable. This can be accomplished by an automatic damper in the outlet flue, adjusted to hold the desired pressure at the selected point in the furnace enclosure. In furnaces with door openings at either end, the point of measurement should be close to hearth level near the discharge end. A practical furnace pressure will be 0.01–0.05 in. H_2O.

With recuperative or regenerative firing systems, the preferred location of the control damper will be between the waste-heat recovery system and the stack, to operate at minimum temperature. In high-temperature furnaces without waste-heat recovery, a water-cooled damper may be needed.

With combustion air preheated before distribution to several firing zones, the ratio control system for each zone will need adjustment to entering air temperature. However, if each firing zone has a separate waste-heat recovery system, the zone air supply can be measured before preheating to maintain the balance with fuel input.

The diagram of a combustion control system in Fig. 45.43 shows how these control functions can be interlocked with the required instrumentation.

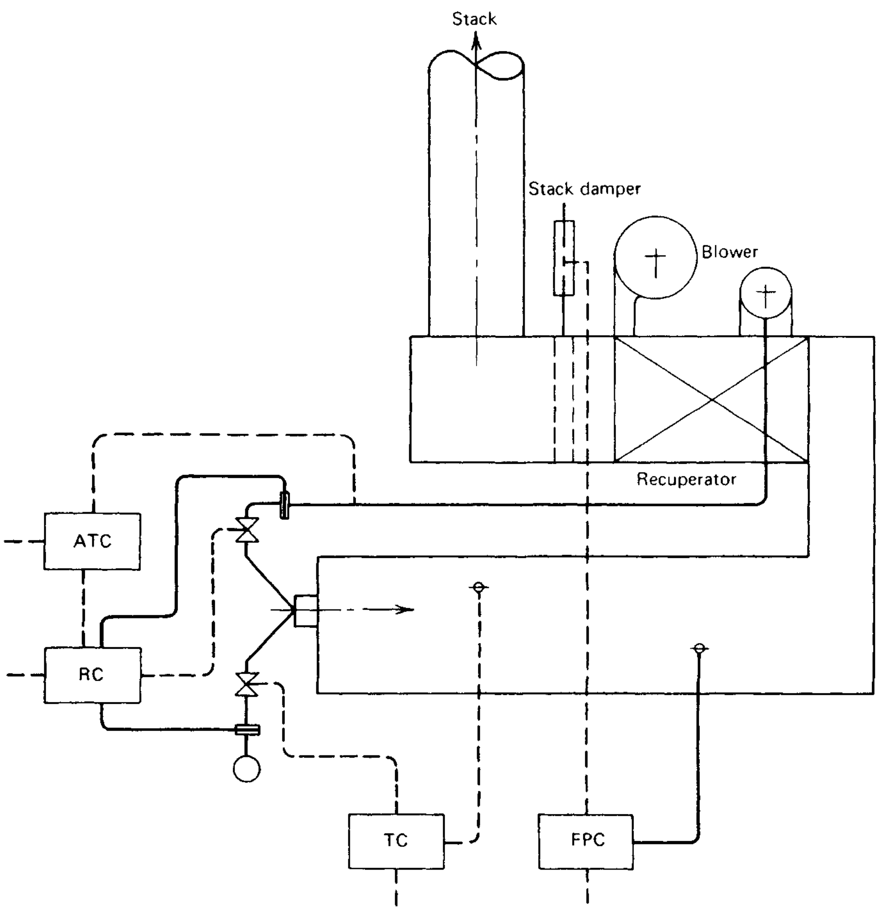

Fig. 45.43 Combustion control diagram for recuperative furnace.[1]

For automatic furnace pressure control to be effective, it should be used in combination with proportioning-type temperature control. With on–off control, for example, the control of furnace pressure at zero firing rate cannot be accomplished by damper adjustment, and with a continuous variation in firing rate between maximum and minimum limits, or between maximum and off, the adjustment of damper position to sudden changes in firing rate will involve a time-lag factor that can make control ineffective.

An important function of a furnace control system is to guard against safety hazards, such as explosions, fires, and personal injury. Requirements have been well defined in codes issued by industrial insurers, and include provision for continuous ignition of burners in low-temperature furnaces, purging of atmosphere furnaces and combustion of hydrogen or carbon monoxide in effluent atmospheres, and protection of operating personnel from injury by burning, mechanical contact, electrical shock, poisoning by inhalation of toxic gases, or asphyxiation. Plants with extensive furnace operation should have a safety engineering staff to supervise selection, installation, and maintenance of safety hazard controls and to coordinate the instruction of operating personnel in their use.

45.10.4 Air Pollution Control

A new and increasing responsibility of furnace designers and operators is to provide controls for toxic, combustible, or particulate materials in furnace flue gases, to meet federal or local standards for air quality. Designs for furnaces to be built in the immediate future should anticipate a probable increase in restrictions of air pollution in coming years.

Toxic contaminants include sulfur and chlorine compounds, nitrogen oxides, carbon monoxide, and radioactive wastes. The epidermic of "acid rain" in areas downwind from large coal-burning facilities is an example.

Combustible contaminants include unburned fuel, soot, and organic particulates from incinerators, and the visible constituents of smoke, except for steam. Other particulates include suspended ash and suspended solids from calcination processes.

Types of control equipment include:

1. Bag filters or ceramic fiber filters to remove suspended solids. Filters require periodic cleaning or replacement, and add to the pressure drop in flue gases leaving the system.
2. Electrostatic filters, in which suspended particles pass through a grid to be electrically charged, and are collected on another grid or on spaced plates with the opposite potential. Smaller units are cleaned periodically by removal and washing. Large industrial units are cleaned in place. A possible objection to their use is a slight increase in the ozone content of treated air.
3. Wet scrubbers are particularly effective for removing water-soluble contaminants such as sulfur and chlorine compounds. They can be used in place of filters for handling heavy loads of solid particulates such as from foundry cupola furnaces, metal-refining processes, and lime kilns. Waste material is collected as a mud or slurry, requiring proper disposal to avoid solid-waste problems.
4. Combustible wastes, such as the solvent vapors from organic coating ovens, may be burned in incinerator units by adding combustion air and additional fuel as required. Fuel economy may be improved by using waste heat from combustion to preheat incoming gases through a recuperator. The same system may be used for combustible solid particulates suspended in flue gases.
5. Radioactive wastes from nuclear power plants will usually be in the form of suspended solids that can be treated accordingly if suitable facilities for disposal of collected material are available, or as radioactive cooling water for which a suitable dumping area will be needed.

45.11 WASTE HEAT RECOVERY SYSTEMS

In fuel-fired furnaces, a fraction of the energy from combustion leaves the combustion chamber as sensible heat in waste gases, and the latent heat of evaporation for any water vapor content resulting from the combustion of hydrogen. Losses increase with flue gas temperature and excess air, and can reach 100% of input when furnace temperatures equal theoretical flame temperatures.

Waste heat can be recovered in several ways:

1. Preheating incoming loads in a separate enclosure ahead of the furnace.
2. Generating process steam, or steam for electric power generation. Standby facilities will be needed for continuous demand, to cover interruptions of furnace operation.
3. Preheating combustion air, or low-Btu fuels, with regenerative or recuperative firing systems.

45.11.1 Regenerative Air Preheating

For the high flue gas temperatures associated with glass- and metal-melting processes, for which metallic recuperators are impractical, air may be preheated by periodical reversal of the direction of firing, with air passing consecutively through a hot refractory bed or checker chamber, the furnace combustion chamber, and another heat-storage chamber in the waste-gas flue. The necessary use of the furnace firing port as an exhaust port after reversal limits the degree of control of flame patterns and the accuracy of fuel/air control in multiple port furnaces. Regenerative firing is still preferred, however, for open hearth furnaces used to convert blast furnace iron to steel, for large glass-melting furnaces, and for some forging operations.

A functional diagram of a regenerative furnace is shown in Fig. 45.44. The direction of flow of combustion air and flue gas is reversed by a valve arrangement, connecting the low-temperature end of the regenerator chamber to either the combustion air supply or the exhaust stack. Fuel input is reversed simultaneously, usually by an interlocked control. Reversal can be in cycles of from 10 to 30 min duration, depending primarily on furnace size.

45.11.2 Recuperator Systems

Recuperative furnaces are equipped with a heat exchanger arranged to transfer heat continuously from outgoing flue gas to incoming combustion air. Ceramic heat exchangers, built up of refractory tubes or refractory block units arranged for cross flow of air and flue gas, have the advantage of higher temperature limits for incoming flue gas, and the disadvantage of leakage of air or flue gas between passages, with leakage usually increasing with service life and pressure differentials. With the improvement in heat-resistant alloys to provide useful life at higher temperatures, and with better control of incoming flue gas temperatures, metallic recuperators are steadily replacing ceramic types.

Metal recuperators can be successfully used with very high flue gas temperatures if entering temperatures are reduced by air dilution or by passing through a high-temperature waste-heat boiler.

Familiar types of recuperators are shown in the accompanying figures:

Figure 45.45: radiation or stack type. Flue gases pass through an open cylinder, usually upward, with heat transfer primarily by gas radiation to the surrounding wall. An annular passage is provided between inner and outer cylinders, in which heat is transferred to air at high velocity by gas radiation and convection, or by solid-state radiation from inner to outer cylinders and convection. The radiation recuperator has the advantage of acting as a portion of an exhaust stack, usually with flue gas and air counterflow. Disadvantages are distortion and resulting uneven distribution of air flow, resulting from differential thermal expansion of the inner tube, and the liability of damage from secondary combustion in the inner chamber.

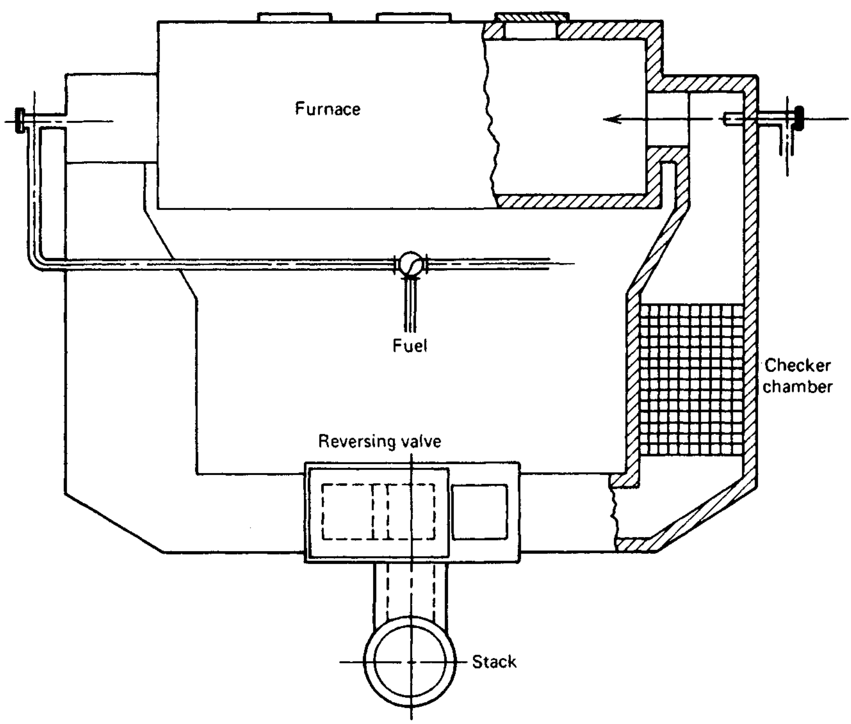

Fig. 45.44 Regenerative furnace diagram.[1]

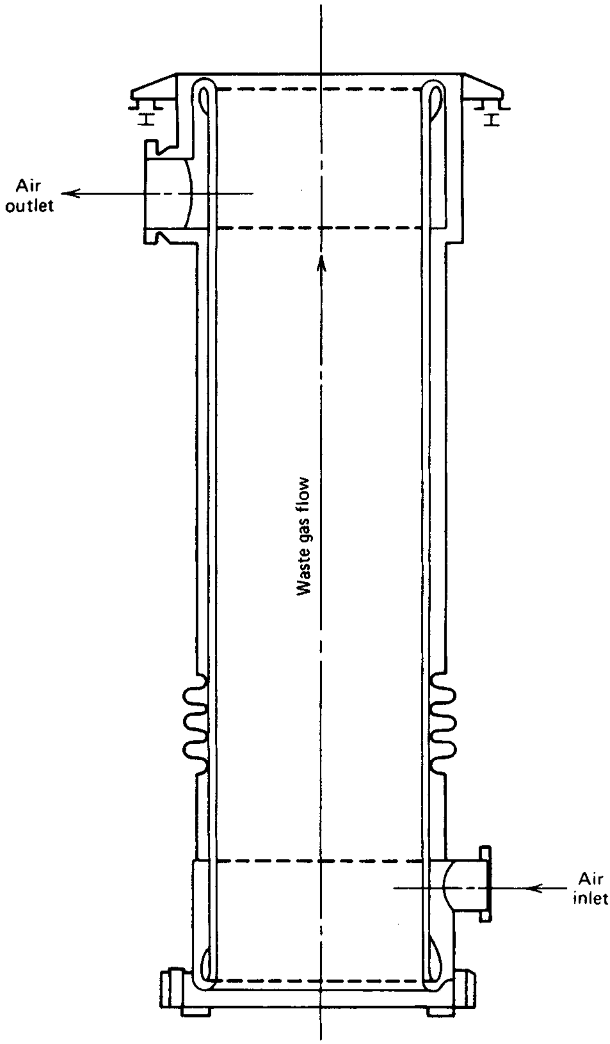

Fig. 45.45 Stack-type recuperator.[1] (Courtesy Morgan Engineering Company.)

Figure 45.46: cross-flow tubular type. By passing air through a series of parallel passes, as in a tube assembly, with flue gas flowing across tubes, relatively high heat-transfer rates can be achieved. It will ordinarily be more practical to use higher velocities on the air side, and use an open structure on the flue gas side to take some advantage of gas radiation. Figure 45.46 shows a basic arrangement, with air tubes in parallel between hot and cold air chambers at either end. Some problems may be introduced by differential thermal expansion of parallel tubes, and tubes may be curved to accommodate variations in length by lateral distortion.

A popular design avoids the problems of thermal expansion by providing heat-exchange tubes with concentric passages and with connections to inlet and outlet manifolds at the same end. Heat transfer from flue gas to air is by gas radiation and convection to the outer tube surface, by convection from the inner surface to high-velocity air, and by solid-state radiation between outer and inner tubes in series with convection from inner tubes to air. Concentric tube recuperators are usually designed for replacement of individual tube units without a complete shutdown and cooling of the enclosing flue. The design is illustrated in Fig. 45.47.

45.11.3 Recuperator Combinations

To provide preheated air at pressure required for efficient combustion, without excessive air leakage from the air to the flue gas side in refractory recuperators, the air pressure can be increased between

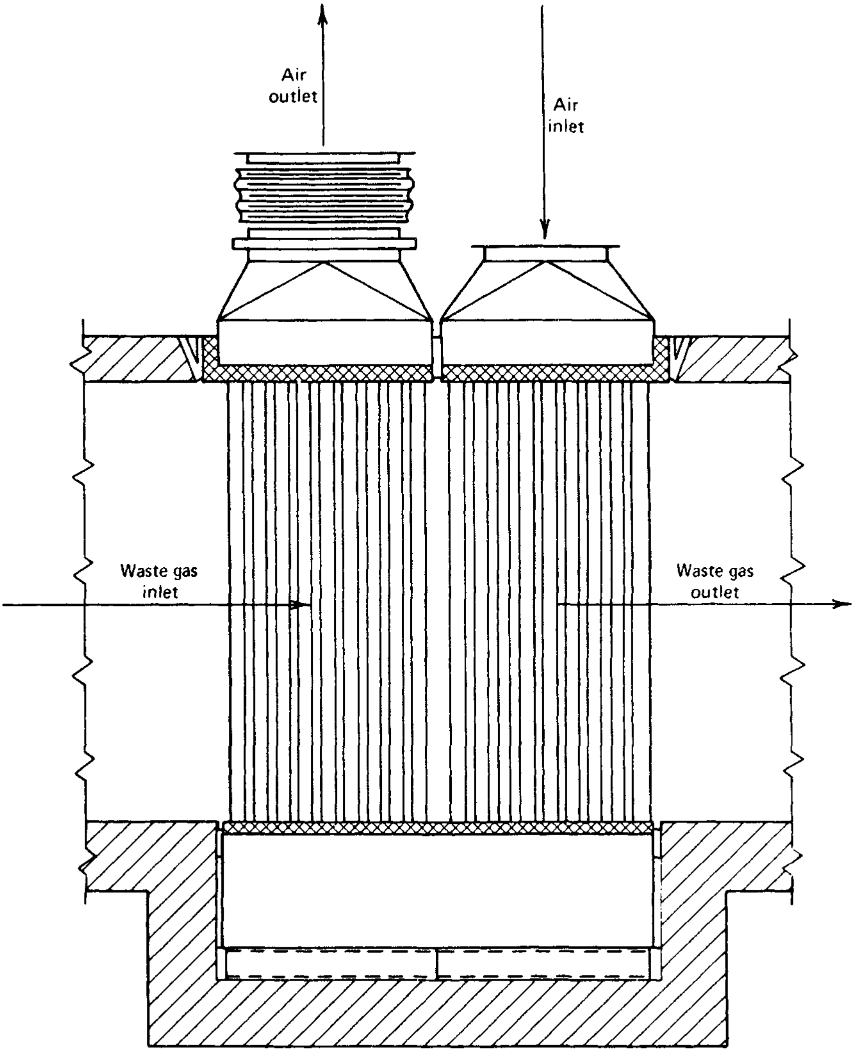

Fig. 45.46 Cross-flow-type recuperator in waste-gas flue.[1] (Courtesy Thermal Transfer Corporation.)

the recuperator and burner by a booster fan. Top air temperatures will be limited by fan materials. As an alternative, air temperatures can be boosted by a jet pump with tolerance for much higher temperatures.

In a popular design for recuperator firing of soaking pits, flue gases pass through the refractory recuperator at low pressure, with air flowing counterflow at almost the same pressure. Air flow is induced by a jet pump, and, to increase the jet pump efficiency, the jet air can be preheated in a metal recuperator between the refractory recuperator and the stack. Because the metal recuperator can handle air preheated to the limit of the metal structure, power demand can be lowered substantially below that for a cold air jet.

Radiant tubes can be equipped with individual recuperators, as shown in Fig. 45.48. Some direct-firing burners are available with integral recuperators.

45.12 FURNACE COMPONENTS IN COMPLEX THERMAL PROCESSES

An industrial furnace, with its auxiliaries, may be the principal component in a thermal process with functions other than heating and cooling. For example, special atmosphere treatment of load surfaces, to increase or decrease carbon content of ferrous alloys, can be accomplished in a furnace heated by

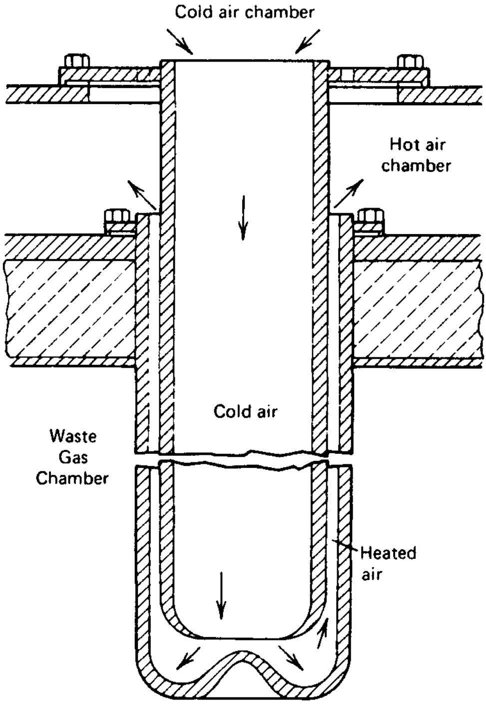

Fig. 45.47 Concentric tube recuperator, Hazen type.[1] (Courtesy C-E Air Preheater Division, Combustion Engineering, Inc.)

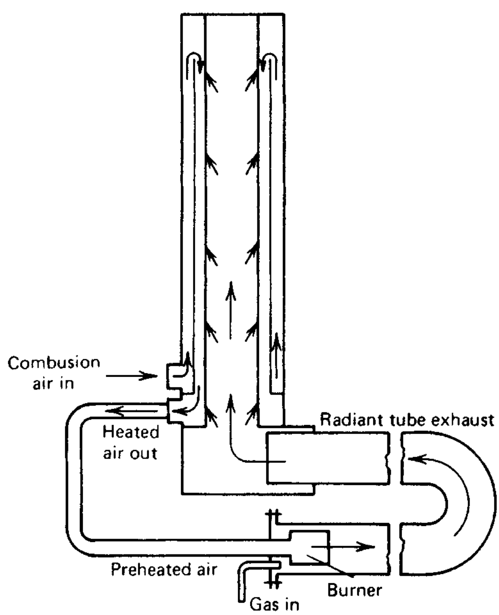

Fig. 45.48 Radiant tube recuperator.[1] (Courtesy Holcroft Division, Thermo-Electron Corp.)

radiant tubes or electrical heating elements or by electric induction. A source of the required controlled atmosphere is usually part of the furnace process equipment, designed and supplied by the furnace manufacturer.

Continuous heat treatment of strip or wire, to normalize or anneal ferrous materials, followed by coating in molten metal, such as zinc or aluminum, or electroplating can be accomplished by one of two arrangements for furnace coating lines. One arrangement has a sequence of horizontal passes, with a final cooling zone to regulate strip temperature to the approximate temperature of the coating bath, and an integral molten-metal container. Strip is heat treated in a controlled atmosphere to avoid oxidation, with the same atmosphere maintained to the point of immersion in molten metal. The second arrangement is for higher velocities and longer strands in heating and cooling passes. In this arrangement, strip may be processed in a series of vertical strands, supported by conveyor rolls.

Furnace lines designed for either galvanizing or aluminum coating may be designed with two molten-metal pots, with the entry strand arranged to be diverted to either one, and with the cooling zone adjustable to discharge the strand to either pot at the required temperature.

Thermal processing lines may include furnace equipment for heating the load to the temperature required for annealing, normalizing, or hardening, a quench tank for oil or water cooling to develop hardness, a cleaning station to remove quench oil residues, and a separate tempering furnace to develop the desired combination of hardness and toughness. Loads may be in continuous strand form, or in units carried by trays or fixtures that go through the entire process or carried on a series of conveyors. The required atmosphere generator will be part of the system.

Where exposure to hydrogen or nitrogen in furnace atmospheres may be undesirable, as in heat treatment of some ferrous alloys, heating and cooling can be done in a partial vacuum, usually with heat supplied by electrical resistors. Quenching can be done in a separate chamber with a controlled atmosphere suitable for brief exposure.

Systems for collecting operating data from one or more furnaces, and transmitting the data to a central recording or controlling station, may also be part of the responsibility of the furnace supplier.

45.13 FURNACE CAPACITY

Factors limiting the heating capacity of industrial furnaces include building space limitations, available fuel supplies, limited temperature of heat sources such as electric resistors or metal radiant tubes, and limits on final load temperature differentials. Other factors under more direct control by furnace designers are the choice between batch and continuous heating cycles; time–temperature cycles to reach specified final load temperatures; fuel firing arrangements; and control systems for furnace temperature, furnace pressure, and fuel/air ratios. In addition, the skills and motivation of furnace operating personnel, as the result of training, experience, and incentive policies, will directly affect furnace efficiency.

45.14 FURNACE TEMPERATURE PROFILES

Time–temperature patterns can be classified as uniform wall temperature (T_w), uniform combustion gas temperature (T_g), or variable T_w and T_g designed to secure the best combination of heating capacity and fuel efficiency.

In a batch-type furnace with fairly massive loads, the temperature control system can be arranged to allow firing at the maximum burner capacity until a preset wall temperature limit is reached, adjusting firing rate to hold that wall temperature, until load temperature approaches the limit for the heated surface, and reducing the wall temperature setting to hold maximum load temperature T_s while the minimum T_c reaches the desired level.

In continuous furnaces, control systems have evolved from a single firing zone, usually fired from the discharge end with flue gas vented from the load charge end, to two or three zone firing arranged for counterflow relation between furnace loads and heating gases.

Progress from single to multiple zone firing has improved heating rates, by raising furnace temperatures near the charge end, while increasing fuel demand by allowing higher temperatures in flue gas leaving the preheat zone. Load temperature control has been improved by allowing lower control temperatures in the final zone at the discharge end.

With multiple zone firing, the control system can be adjusted to approach the constant-gas-temperature model, constant wall temperature, or a modified system in which both T_g and T_w vary with time and position. Because gas temperatures are difficult to measure directly, the constant-gas-temperature pattern can be simulated by an equivalent wall temperature profile. With increasing fuel costs, temperature settings in a three-zone furnace can be arranged to discharge flue gases at or below the final load temperature, increasing the temperature setting in the main firing zone to a level to provide an equilibrium wall and load temperature, close to the desired final load temperature, during operating delays, and setting a temperature in the final or soak zone slightly above the desired final load surface temperature.

45.15 REPRESENTATIVE HEATING RATES

Heating times for various furnace loads, loading patterns, and time–temperature cycles can be calculated from data on radiation and non-steady-state conduction. For preliminary estimates, heating times for steel slabs to rolling temperatures, with a furnace temperature profile depressed at the entry end, have been estimated on a conservative basis as a function of thickness heated from one side and final load temperature differential and are shown in Fig. 45.26. The ratios for heating time required for square steel billets, in various loading patterns, are shown in Fig. 45.25. For other rectangular cross sections and loading patterns, heating times can be calculated by the Newman method.

Examples of heating times required to reach final load temperatures of T_s = 2300°F and T_c = 2350°F, with constant furnace wall temperatures, are:

1. 12-in.-thick carbon steel slab on refractory hearth with open firing: 9 hr at 54.4 lb/hr · ft^2.
2. 4-in.-thick slab, same conditions as 1: 1.5 hr at 109 lb/hr · ft^2.
3. 4 in. square carbon steel billets loaded at 8 in. centers on a refractory hearth: 0.79 hr at 103 lb/hr · ft^2.
4. 4 in. square billets loaded as in 3, but heated to T_s = 1650°F and T_c = 1600°F for normalizing: 0.875 hr at 93 lb/hr · ft^2.
5. Thin steel strip, heated from both sides to 1350°F by radiant tubes with a wall temperature of 1700°F, total heating rate for both sides: 70.4 lb/hr · ft^2.
6. Long aluminum billets, 6 in. diameter, are to be heated to 1050°F. Billets will be loaded in multiple layers separated by spacer bars, with wind flow parallel to their length. With billets in lateral contact and with wind at a mean temperature of 1500°F, estimated heating time is 0.55 hr.
7. Small aluminum castings are to be heated to 1000°F on a conveyor belt, by jet impingement of heated air. Assuming that the load will have thick and thin sections, wind temperature will be limited to 1100°F to avoid overheating thinner sections. With suitable nozzle spacing and wind velocity, the convection heat-transfer coefficient can be H_c = 15 Btu/hr · ft^2 and the heating rate 27 lb/hr · ft^2.

45.16 SELECTING NUMBER OF FURNACE MODULES

For a given heating capacity and with no limits on furnace size, one large furnace will cost less to build and operate than a number of smaller units with the same total hearth area. However, furnace economy may be better with multiple units. For example, where reheating furnaces are an integral part of a continuous hot strip mill, the time required for furnace repairs can reduce mill capacity unless normal heating loads can be handled with one of several furnaces down for repairs. For contemporary hot strip mills, the minimum number of furnaces is usually three, with any two capable of supplying normal mill demand.

Rolling mills designed for operation 24 hr per day may be supplied by batch-type furnaces. For example, soaking-pit-type furnaces are used to heat steel ingots for rolling into slabs. The mill rolling rate is 10 slabs/hr. Heating time for ingots with residual heat from casting averages 4 hr, and the time allowed for reloading an empty pit is 2 hr, requiring an average turnover time of 6 hr. The required number of ingots in pits and spaces for loading is accordingly 60, requiring six holes loaded 10 ingots per hole.

If ingots are poured after a continuous steelmaking process, such as open hearth furnaces or oxygen retorts, and are rolled on a schedule of 18 turns per week, it may be economical at present fuel costs to provide pit capacity for hot storage of ingots cast over weekends, rather than reheating them from cold during the following week.

With over- and underfired slab reheating furnaces, with slabs carried on insulated, water-cooled supports, normal practice has been to repair pipe insulation during the annual shutdown for furnace maintenance, by which time some 50% of insulation may have been lost. By more frequent repair, for example, after 10% loss of insulation, the added cost of lost furnace time, material, and labor may be more than offset by fuel savings, even though total furnace capacity may be increased to offset idle time.

45.17 FURNACE ECONOMICS

The furnace engineer may be called on to make decisions, or submit recommendations for the design of new furnace equipment or the improvement of existing furnaces. New furnaces may be required for new plant capacity or addition to existing capacity, in which case the return on investment will not determine the decision to proceed. Projected furnace efficiency will, however, influence the choice of design.

If new furnace equipment is being considered to replace obsolete facilities, or if the improvement of existing furnaces is being considered to save fuel or power, or to reduce maintenance costs, return on investment will be the determining factor. Estimating that return will require evaluation of these factors:

Projected service life of equipment to be improved

Future costs of fuel, power, labor for maintenance, or operating supervision and repairs, for the period assumed

Cost of production lost during operating interruptions for furnace improvement or strikes by construction trades

Cost of money during the improvement program and interest available from alternative investments

Cost of retraining operating personnel to take full advantage of furnace improvements

45.17.1 Operating Schedule

For a planned annual capacity, furnace size will depend on the planned hours per year of operation, and fuel demand will increase with the ratio of idle time to operating time, particularly in furnaces with water-cooled load supports. If furnace operation will require only a two- or three-man crew, and if furnace operation need not be coordinated with other manufacturing functions, operating costs may be reduced by operating a smaller furnace two or three turns per day, with the cost of overtime labor offset by fuel savings.

On the other hand, where furnace treatment is an integral part of a continuous manufacturing process, the provision of standby furnace capacity to avoid plant shutdown for furnace maintenance or repairs may be indicated.

If furnace efficiency deteriorates rapidly between repairs, as with loss of insulation from water-cooled load supports, the provision of enough standby capacity to allow more frequent repairs may reduce overall costs.

45.17.2 Investment in Fuel-Saving Improvements

At present and projected future costs of gas and oil fuels, the added cost of building more efficient furnaces or modifying existing furnaces to improve efficiency can usually be justified. Possible improvements include better insulation of the furnace structure, modified firing arrangements to reduce flue gas temperatures or provide better control of fuel/air ratios, programmed temperature control to anticipate load changes, more durable insulation of water-cooled load supports and better maintenance of insulation, proportioning temperature control rather than the two position type, and higher preheated air temperatures. For intermittent furnace operation, the use of a low-density insulation to line furnace walls and roofs can result in substantial savings in fuel demand for reheating to operating temperature after idle periods.

The relative costs and availability of gas and oil fuels may make a switch from one fuel to another desirable at any future time, preferably without interrupting operations. Burner equipment and control systems are available, at some additional cost, to allow such changeovers.

The replacement of existing furnaces with more fuel-efficient designs, or the improvement of existing furnaces to save fuel, need not be justified in all cases by direct return on investment. Where present plant capacity may be reduced by future fuel shortages, or where provision should be made for increasing capacity with fuel supplies limited to present levels, cost savings by better fuel efficiency may be incidental.

Government policies on investment tax credits or other incentives to invest in fuel-saving improvements can influence the return on investment for future operation.

REFERENCE

1. C. Cone, *Energy Management for Industrial Furnaces,* Wiley, New York, 1980.