contents

3 Cooling Towers 89

4 Pressure Vessels and Storage Tanks 103

5 Separators and Accumulators 153

6 Flare Stacks 166

7 Piping 188

11 Foundation Design for Towers 345

12 Insulation 363

Index 369

1. fired heaters and boilers

No industry-wide standards such as those of TEMA for heat exchangers or those of the API for rotating equipment cover process heaters. The purchaser must prepare an inquiry specification as carefully and as completely as possible without being overly restrictive to provide a guide so that the bidders can develop their quotations intelligently.

API Standard 665, Fired Heater Data Sheet, was developed to minimize problems of incomplete specifications. Its five pages provide a complete check list. Information which must be defined by the purchaser is marked with an asterisk and includes such data as complete process requirements, fuel characteristics, structural design data, coil design conditions, types of headers and terminals required and site conditions.

Heat and Material Balance

The operating company's process design section will have made heat and material balances for the plant in which the heater is required. There may even be several of these to cover different possible operating conditions, particularly in the case of a crude distillation unit. Inherent in these balances will be information on the fluid to be heated, the temperature of the substance entering and leaving the heater, the exit pressure and the enthalpies at entrance and exit, as well as the fraction in vapor form at the exit conditions. This information and the flow rate, which is best expressed in terms of mass flow per hour, will define the required heat-absorbing capacity or *duty*.

Physical Property Data

The characteristics of the fluid to be heated should be given. If there is a possibility that some of it will be vaporized in passing through the heater, equilibrium flash curves should be furnished, preferably at two or more pressures. This is particularly important if the equipment is to operate below atmospheric pressure. Accompanying these curves should be the specific gravity and the molecular weights of the vapor and liquid fractions obtained for each point on the flash distillation curves.

Maximum permissible temperature is often critical. The higher the temperature reached by the fluid, the greater the tendency to decompose or crack. Such cracking may make products off-color or otherwise objectionable and might even require expensive treatment or redistillation. In some instances, corrosion may increase rapidly after certain temperatures are passed. For this reason the maximum permissible temperature to which the fluid may be heated must be specified. Note that the highest temperature may not be at the heater exit but may be some distance back in the tubes if the rate of pressure drop is high near the end.

The temperature must be based on experience with the particular fluid. For crude petroleum and residues, temperatures of around 725 to 750°F are commonly given as maximums. Such temperature limits do not apply when the purpose of the heater is to produce a molecular rearrangement, as in thermal cracking, reforming or viscosity breaking.

Fuel and Combustion Data

The type of fuel and its heating value should be specified, preferably at the lower heating value, in *Btu per pound of fuel.* Even for gas fuel, this provides a better relationship than *heating value per cubic foot*, as it must eventually be put in terms of mass to make the computations of air-fuel ratio.

For fuel oil, the ultimate analysis in terms of hydrogen, carbon, sulfur and any other component should be given — as well as the specific gravity (or API gravity). Viscosity at two temperatures should also be given. This is needed to determine the temperature to which the fuel must be heated for good atomization.

For gas, in addition to the heating value (preferably the lower heating value in Btu per pound), an ultimate analysis is also useful although certain available correlations of heating value and specific gravity (referred to air) can be used in many instances.

The percent of sulfur by weight should be included unless it can be given as the mole fraction of H_2S. In some natural gases there may be appreciable fractions of carbon dioxide or nitrogen or both. It is important to note this, because it affects not only the combustion characteristics and heating value but also the operation of the CO_2 recorder that will be provided to control the amount of air for combustion.

Capacity

The heater is one equipment item the capacity of which cannot be readily increased after it is built — for instance by adding heat transfer area or putting larger impellers in pumps. If the furnace is the limiting element in the plant, the maximum capacity of the column will never be known. It is well to be generous here. If the uncertainty is evaluated at 15%, the unit must be designed for 115% plus or minus 15% rather than for 100% plus or minus 15%. This means that frequently the column turns out to have a capacity of 130% or more if all uncertainties turn out favorably. For this larger capacity the heater duty is correspondingly increased.

Maximum Allowable Radiant Heat Absorption Rate

Modern fired heaters absorb by radiation 60 to 70% of the total heat absorbed in the heater although this percentage is influenced by the temperature at which the oil enters the heater. The combustion gases are still quite hot (1,500 to 1,800°F). Usually a convection section is added to recover as much additional heat as is economical.

The radiant surface — including its supporting casting, refractory walls and a portion of the supporting frame and casing — costs more per square foot than does the more compact convection bank. The cheapest overall cost will be obtained if the radiant heat absorption rate is made as high as practicable.

The radiant absorption rate has been expressed in different ways by different investigators.

1. Btu per hour per square foot of projected tube area
2. Btu per hour per square foot of external tube surface (circumferential area)
3. Btu per hour per square foot on the most exposed element of the tube. This is the same as the rate on an equivalent plane surface absorbing the same heat as does the tube bank.
4. Btu per hour per square foot on the total wall surface covered by radiant absorbing tubes.

Sometimes more than one of these will be used in the same reference.

Table 1-1. Conversion of Heat Absorption Rates for Tubes Spaced Two Diameters

Multiply Number / To Obtain	Absorption on Most Exposed Element	Average Absorption on Wall Covered By Tubes	Average Absorption on Circum-Ferential Area	Average Absorption on Projected Area
Absorption on most exposed element	1.000	1.125	1.780	0.568
Average absorption on wall covered by tubes	0.8800	1.000	1.570	0.500
Average absorption on circumferential area	0.562	0.640	1.000	0.319
Average absorption on projected area	1.76	2.000	3.140	1.000

The projected tube area has little to recommend it. It does not show that the total heat absorption depends on the tube spacing. Absorption on the circumferential area is the most commonly used method. However, this does not show the effect of tube spacing either. It is, furthermore, quite misleading if there are two rows of tubes, one behind the other.

The absorption rate on the most exposed element determines the maximum absorption rate and must be used to make the radiation heat balance. This is the basic design criterion.

The absorption per square foot of wall surface behind the tubes is useful for determining the amount of furnace wall that must be covered with tubes. However, for a given spacing of tubes in terms of diameter, these rates are mutually interconvertible. Table 1-1 gives a table for tubes spaced two diameters center to center.

The maximum permissible heat absorption rate is the rate at which the thin film of oil next to the hottest side of the tube begins to crack and deposit coke. This depends on the nature of the fluid being heated, the average temperature of the fluid in the tubes and the fluid velocity.

A higher radiant heat absorption rate can be permitted with colder fluids and higher velocities. In steam boilers, rates of 60,000 Btu/hr. sq. ft. are not uncommon. Some data report up to four times this rate with exceptionally pure, silica-free water. Companies that design heaters once recommended radiant absorption rates of 12,500 to 15,000 Btu/hr. sq. ft. of circumferential tube surface. In recent years, rates up to 17,500 Btu/hr. sq. ft. have been offered. In practice, rates up to 50% higher than these are not uncommon.

Tubes

Tube Spacing

Return headers are available with minimum center-to-center distances of the tubes from 1-1/2 to 2 diameters, depending on the tube size and pressure ratings. Maximum tube spacing for standard headers is 2-1/2 or 3 diameters.

The average absorption rate per square foot of circumferential area increases with the tube spacing as shown in Table 1-2. This table shows that six tubes spaced 1-1/2 diameters on centers and covering nine diameters width of wall will absorb the same amount of heat as five tubes spaced two diameters and covering ten diameters width of wall. It is then a question of whether saving of one tube and its header amounts to more than the cost of the wall section of one-diameter width. In general it is more economical to use the two-diameter spacing.

Double rows of tubes are not economical. It was formerly quite common to have double rows of tubes on the walls of heaters. Two rows cost twice as much for tubes and headers and only absorb 12% more heat.

Tube Diameter and Length

For a given working pressure the cost of a tube varies approximately as the square of the diameter. Return headers, being geometrically similar, vary in weight and cost approximately as the cube of the diameter. For a given total surface, smaller tubes are obviously cheaper. On the other hand, the pressure drop per unit length varies as the inverse of the diameters, $1/D$. To use smaller diameter tubes and have a reasonable pressure drop, it is usually necessary to have some passes in parallel.

If the tube length is L and the diameter D for a single pass, for n passes the length of each pass would be $L_n = L/(n)^{1/2}$ and the diameter $D_n = D/(n)^{1/2}$. The total weight of the tubes would be proportional to $1/(n)^{1/2}$ for n passes as compared with a single pass. Thus the cost of the tubes for four passes will be half that for one pass. These relations are, of course, approximate. They are based on the inside diameter for pressure drop. The weight is approximately the inside diameter plus the tube thickness. No account is taken of the

Table 1-2. Effect of Tube Spacing on Average Rate of Heat Absorption Per Square Foot of Circumferential Area

Spacing Diameters	1½	2	2½	3
Average absorption rate compared with most exposed element	0.465	0.56	0.635	0.68
Ratio to rate for 2 diameter spacing	0.83	1.00	1.135	1.217

extra thickness that may be included for corrosion allowance.

Tube Passes

Even in large heaters the total pressure drop can be kept in the range of 100 psi. The following precautions are essential for avoiding trouble.

1. The velocity of the fluid should be kept fairly high in the convection section so that a considerable fraction of the pressure drop will occur in the tubes, where no evaporation may be expected. This has a dampening effect on any tendency toward unstable flow.
2. There should be as many parallel passes in the convection bank as there are in the radiant bank at the point where they join (each pass in the convection bank may consist of two parallel passes for each one in the radiant bank).
3. There should be no further branching of passes in the radiant bank up to the point where vaporization starts. Branching into additional parallel passes may start there, as required by the increasing volume, but there should be no cross connections between the parallel passes except at the beginning of the convection bank and the end of the radiant bank where the transfer line or lines connect.
4. The practical difficulty is to arrange the passes so that each pass has as nearly as possible equal exposure to the radiation in the combustion chamber. It is easy enough to have one pass up each side wall of a heater, branching to four on the roof. By using jump headers, two parallel passes can be placed on each side of the two sidewalls, possibly branching to a total of eight on the roof. With tubes on the floor, two side walls and roof, four or possibly eight parallel passes can be installed in a rectangular heater of square cross section. Various types of cellular arrangements have also been used. Multiple passes can also be arranged readily with vertical tubes in a cylindrical combustion chamber or with vertical tubes in a cellular arrangement.

Return Tube Header

The tubes of a heater are connected in series by return bends. When cleaning will not be required, welded return bends may be used. Generally, however, each tube will have a header at one end with a removable plug for cleaning. If much cleaning is required, removable plugs will be provided at both ends of each tube.

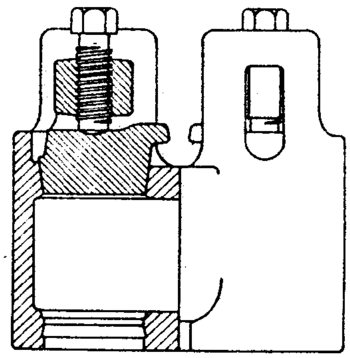

Figure 1-1*a*. The "mule ear" type return tube header is the most commonly used in the process industry.

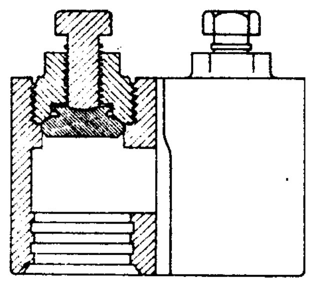

Figure 1-1*b*. A modern "screw lock" return header.

The most commonly-used header is the *mule ear* type, shown in Figure 1-1*a*.

Another type of header is the *screw lock* type, shown in Figure 1-1*b*. The plug in this header has a rather flat bevel (like a globe value) instead of a sharp conical form, and the screw is positioned so that the temperature differences are less than in the mule ear type.

Radiant Absorption Rate

The temperatures at various places in the heater are computed step-by-step as follows:

1. Take as a basis one square foot of radiant absorbing surface in the heater. This should be a square foot of the equivalent plane surface. The rate of heat absorption on this surface will be the same as that on the most exposed element of the

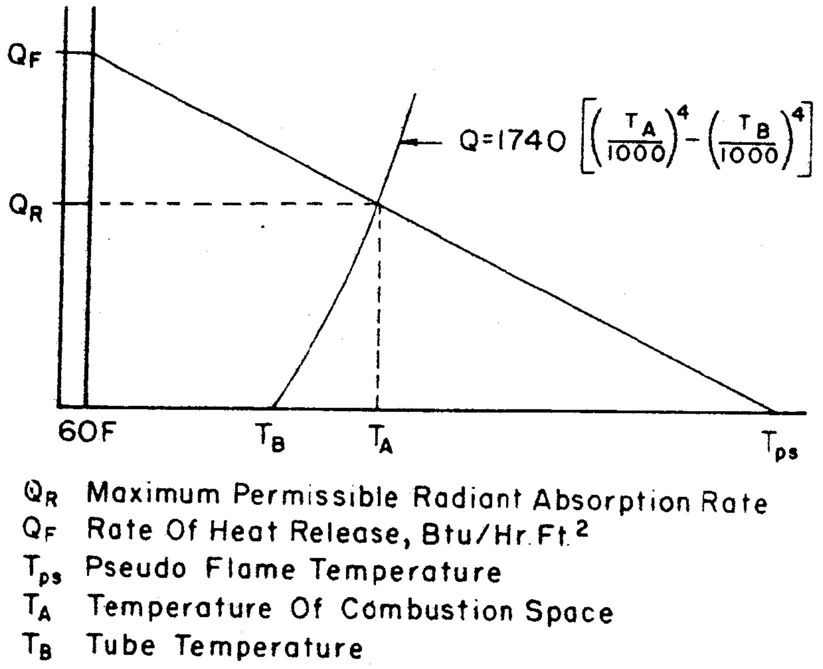

Figure 1-2. Use this chart to find the temperature of the combustion space.

tube. On the basis of the fluid heated and its temperature, decide the maximum permissible radiant absorption rate for the design. Call this Q_r.

2. Estimate the temperature on the outside tube surface. This will be the fluid temperature plus the temperature difference through the tube walls plus whatever fouling resistance experience indicates. This should be computed for the most exposed element.

3. For this tube surface temperature, plot a curve (Figure 1-2) showing the furnace temperatures required to give particular rates of heat transfer by radiation using the Stefan-Boltzmann law:

$$Q_r = 1740\left[\left(\frac{T_a}{1000}\right)^4 - \left(\frac{T_b}{1000}\right)^4\right]$$

where:

Q_r = Btu/hr.sq.ft. on most exposed surfaces
T_a = Furnace temperature, °R
T_b = Outside tube surface temperature, °R

(Note that the absolute temperature is used in this computation, but it will be convenient to plot it in terms of °F.)

For the usual box-type heater, you can get a good correlation of temperatures and absorption rates by using the Stefan-Boltzmann curve, neglecting the heat absorbed by convection suggested by Lobo and Evans and taking ϕ = 1.0 (in effect, ignoring this coefficient for the difference in radiation from gases and black bodies). This makes a considerably easier computation, and the gases in it combine to act as though they constituted a black body at the temperature of the outgoing gases.

4. On Figure 1-2, plot the maximum heat absorption rate chosen for design, expressed in terms of Btu/hr. ft. on the most exposed surface. The two coordinates of this point are the combustion chamber temperature and the heat absorption rate in Btu per hour per square foot on the most exposed surface.

5. For this temperature and for the lower heating value of the fuel and the amount of combustion air, including excess chosen, compute the pseudo-flame temperature and plot this on the zero ordinate. Draw a straight line back through this point and the heat absorption value plotted on the curve until it intersects the line of atmospheric temperature (conventionally, 60°F). This will give the Btu per hr. per sq. ft. that must be released. Call this Q_f. Then Q_r/Q_f is the fraction of the heat released per square foot per hour that is absorbed by radiation.

6. The temperature of the combustion chamber found in Step 4 will usually be in the range of

1,600 to 1,800°F. This still leaves about half the heat of the fuel in the gases. It is usually economical to add a convection section to recover some of this heat. This must be determined by an economic balance, taking into account the value of the heat, the transfer factor, the cost of the surface per square foot (including a proportionate cost of the headers) and the payout period. It may be economical to recover 70 to 80% of the heat available from the range of the combustion chamber temperature down to the entering oil temperature. Call this amount of heat Q_c. The total heat absorbed in both radiant and convection sections is then $Q_r + Q_c$ and the overall furnace efficiency is $E = (Q_r + Q_c)/Q_f$.

For each square foot of radiant section plus the convection section, the heat absorbed per hour is $Q_r + Q_c$. The surface thus found is the equivalent plane surface having the same radiant absorption as the most exposed element of the tubes. This can be converted into other forms by applying the factors from Table 1-1. After finding the overall efficiency, the fuel rate and the weight of the combustion gases can be computed. The convection bank can then be designed by conventional methods.

Convection Bank

It is often difficult to get a convection bank arrangement that will have the relatively high gas velocity required to give a reasonably high heat transfer factor. The centers of the headers for the convection bank, should be as short as they can be and still permit nesting the headers at 60° angles. Usually this requires at least 1-1/2-diameter spacing. Smaller tubes may make the bundle more compact with two or more tubes in parallel for each pass facing the radiant bank. This arrangement will also be cheaper for a given area. Generally, the convection bank should be deep in the direction of the gas flow and narrow transverse to the flow. To give a general idea of the gas velocity required, a velocity of 25 feet per second will give a transfer factor of about 6.3 Btu/hr sq. ft. °F with 4-inch tubes spaced on 8 3/8-inch centers and staggered.

Burners

The type of burners used depends on the fuel, whether oil or gas or both. Furnace capacity is often limited by flames licking the tube, causing hot spots. A short flame is always desirable because of this problem. For short flames the velocity through the burner register must be high — around 40 to 60 feet per second. This may require the use of a forced-draft fan even though there is no air preheater.

The burners should not be installed too close to the tubes. Depending on the burner size, 3 to 5 feet should be provided between the centers of burners and tubes on the same wall. If burners are placed on the end walls, they should be 5 feet from the side walls.

Fuel oil must be atomized to burn. Steam in the burner tip may be used or the oil can be brought in at high pressures to furnish the energy for atomization by so-called mechanical burners.

A gas burner may have a central tip that is interchangeable with an oil burner or it may have a ring with numerous holes surrounding the burner refractory orifice. The latter type is often used for combination oil and gas burners.

If there is an air preheater, the combustion air will be brought in through an insulated duct to avoid heat loss and to protect the operators. It is important that such ducts be provided with expansion bellows and that the duct be properly tapered to insure an even supply of air to each burner.

Structural Arrangements

Usually the heater has a structural steel frame to which tube support castings, hangers and end tubesheets are attached. The support castings must be designed to permit expansion — perhaps 3/4 inch in a five-foot casting. Sheet steel casings are frequently used. Where the climate is mild, a saving may be made by omitting the casing. Roof tube, hang wall tube supports should preferably hold not more than two tubes each to avoid stresses and warping caused by unequal temperatures in different parts of the hangers.

Side wall tube supports should be of the open hook type. This is better than having a web on the hot side, since the web gets hotter than the portion behind the tubes and tends to warp. Because the material must have high strength at high temperatures, 25-12 chrome nickel is commonly used. Intermediate hangers spaced about 10 feet apart are close enough. Typical designs are shown in Figures 1-3 and 1-4.

At the ends of the heaters, tubesheets are provided through which the tubes pass outside the combustion chamber to a header compartment. Tubesheets are generally cast iron with plastic refractory facing on the fire side in sections holding five or six tubes and are bolted to the

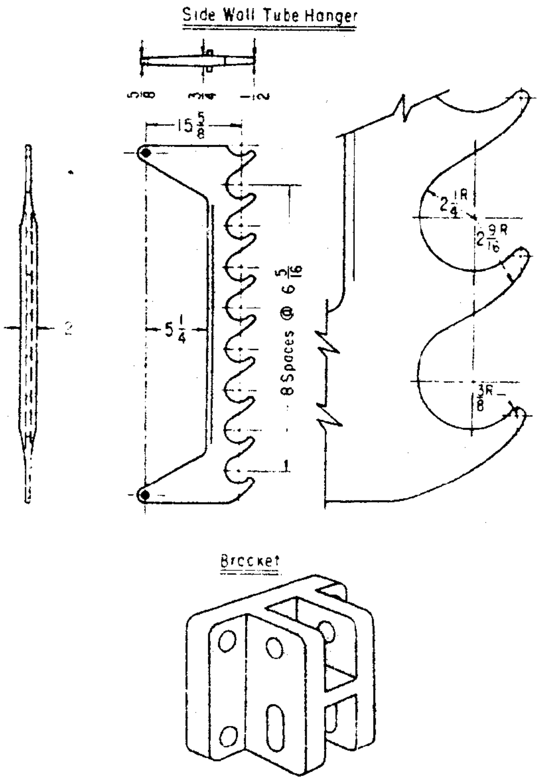

Figure 1-3. Typical side wall tube hangers and brackets for 3½ in. tubes are spaced approximately 10 ft. on centers.

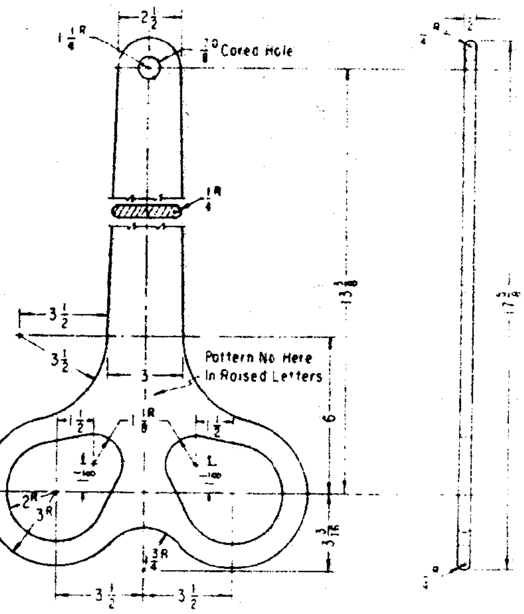

Figure 1-4. Typical roof tube hangers are also spaced 10 ft. on centers.

structural frame by ribs or flanges. These connections should also have slotted holes to permit expansion.

Convection bank intermediate tube supports are also usually of heat resisting materials but are not necessarily as high in alloying elements. It is best not to use large sheets holding many tubes, because the portions near the hottest gases get hotter than the rest and the castings tend to warp. A series of relatively narrow plates holding three rows of tubes, with lugs for stacking, makes a satisfactory arrangement. These may be supported in channel-shaped guides on the side walls of the convection bank enclosure (Figure 1-5).

End tube supports for convection banks may be made larger. They can have refractory facings on the hot side and do not get as hot as the intermediate supports.

Refractories

Compared to those of large steam boilers, process heater temperatures are relatively low.

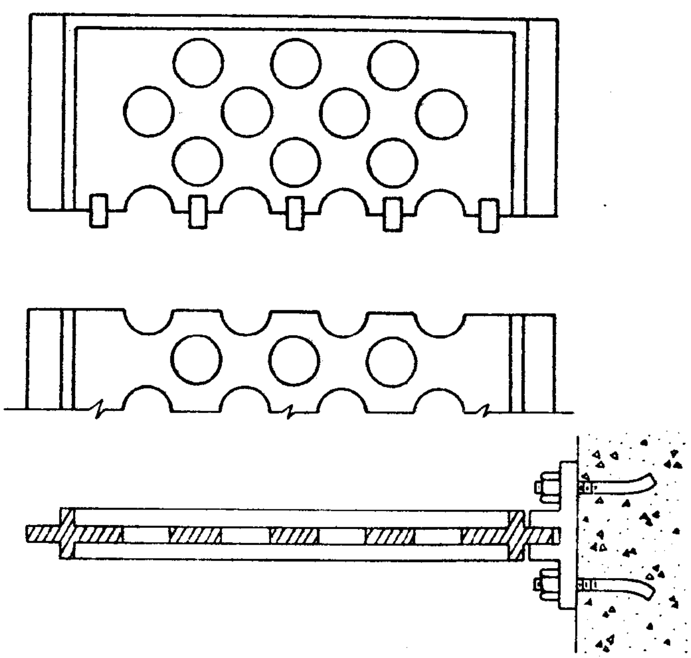

Figure 1-5. Channel shaped guides are used for convection bank intermediate tube supports.

Refractories suitable for the very highest temperatures are not necessary. Insulating refractories with a much higher resistance to heat flow than solid firebrick are available. They permit the use of thinner and lighter walls and roofs. Plastic insulation is placed over the outside of the refractory walls and roof, partly to act as a seal to prevent the lower pressure inside the combustion chamber from drawing air through the walls. The amount of such insulation must be limited so that the castings supporting the walls and roof are not exposed to excessive temperature. If there is no steel casing, this plastic may in turn have a seal coat containing asphalt, generally included as an emulsion.

Foundations

The heater usually rests on a concrete foundation. This may be only a slab if the soil bearing pressure is good. Alternatively, the foundation may be supported by piling and the slab by concrete beams. If the floor of the combustion chamber is practically at ground level, some means of ventilating and cooling the top of the slab should be provided. No amount of insulation under a refractory floor will keep the slab from becoming overheated — with resultant calcination of the concrete — unless there is a means for actually removing the heat that passes through the refractory floor. A layer of hollow tile laid end to end to form air ducts under the floor will do this. Care must be taken that the ends of these ducts are not blocked where the columns are anchored to the slab. The end nearest the stack can have a duct connected to the stack or to the forced-draft fan inlet to produce the necessary air current through the floor ducts.

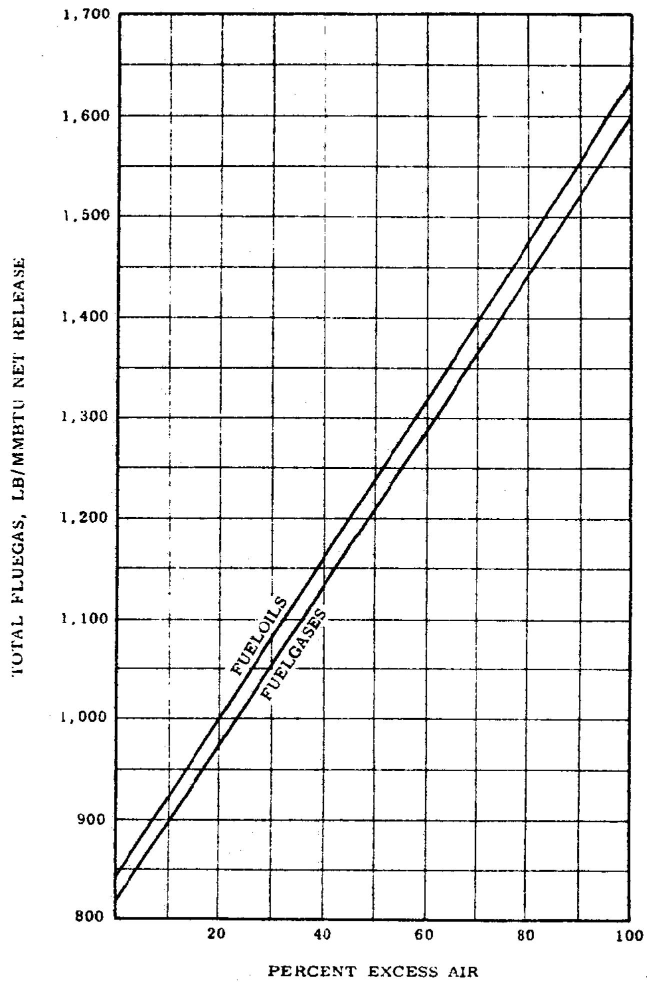

Figure 1-6. Flue gas rates.

Rating a Furnace

First, calculate the net heat release using the specified absorption duty and efficiency. This is done by calculating the flow of product to be heated (in pounds per hour) multiplied by the enthalpy difference of the heated product divided by the specified furnace efficiency. Then select an excess air percentage and determine the flue gas rate from Figure 1-6.

Normal heaters will handle about 70% of the total duty in the radiant section. On this basis, estimate the radiant heat absorption q_R (Btu per hour). From enthalpy or specific heat data for the process fluid, estimate the temperature at the crossover between the convection and radiant sections and the average fluid temperature in the radiant section. Add 100°F to the average fluid temperature to get the average tube wall temperature.

Using the specified allowable radiant transfer rate, calculate the total radiant tube surface.

The convection tube surface is usually about equal to the surface in the radiant section. On this basis, select a tube size and pass arrangement that will give the required total surface and meet the specified pressure drop limitation.

Radiant Section Rating

Choose a center-to-center spacing for the radiant tubes which is compatible with the selected tube size. Wide tube-spacing permits high radiant absorption rates with relatively low firebox temperatures and gives good circumferential heat distribution. Closer spacing permits more tubes to be installed in a given size firebox. The usual spacing is about twice the nominal pipe size corresponding to the outside tube diameter, for example, 8-inch spacing for 4.5-inch OD tubes.

Using the approximate radiant tube surface determined previously, choose firebox dimensions to accommodate the required total tube length. The exact proportions depend on judgment and past experience. Long furnaces minimize the number of return bends required, thus decreasing total cost. Shorter and wider fireboxes, on the other hand, usually give more uniform heat distribution and lessen the probability of flame impingement on the tube surface.

The procedures described here will be covered step-by-step in Example 1.

For the selected firebox dimensions, calculate the equivalent cold plane surface, the effective refractory surface and the mean beam length, L. Read the partial pressure of carbon dioxide plus water from Figure 1-7 and calculate the PL product (P is the partial pressure of the radiating components as a function of excess air for the usual hydrocarbon fuels).

If preheated air or fuel is used, calculate the heat content of each stream above 60°F and its ratio to the total net heat release. The heat content of any of these streams below 100°F can usually be neglected. Estimate the heat loss through the setting.

Assume a value for the average temperature of the flue gas in the radiant section, t_g. At this temperature, read the gas emissivity from Figure 1-8 and then the exchange factor from Figure 1-9. Calculate the ratio $q_n/aA_{cp}F$ where q_n is the heat of combustion in Btu per hour, a is the factor of comparison between a tube bank and a plane, A_{cp} is the cold plane area in square feet, and F is the radiant exchange factor.

If the furnace design is one for which experience has shown a significant difference between average effective gas temperature and exit temperature, apply the appropriate correction to determine the exit temperature t_{g2}. Read the corresponding value of q_{g2}/q_n from Figure 1-10. Then, calculate $q_R/aA_{cp}F$ and plot it against average furnace temperature on Figure 1-11.

If the assumed value of t_g was correct, the computed point will be on the absorption curve for the average tube wall temperature. Usually, of course, there will be a discrepancy. In that case, select another value of t_g on the other side of the absorption curve and repeat the above procedure.

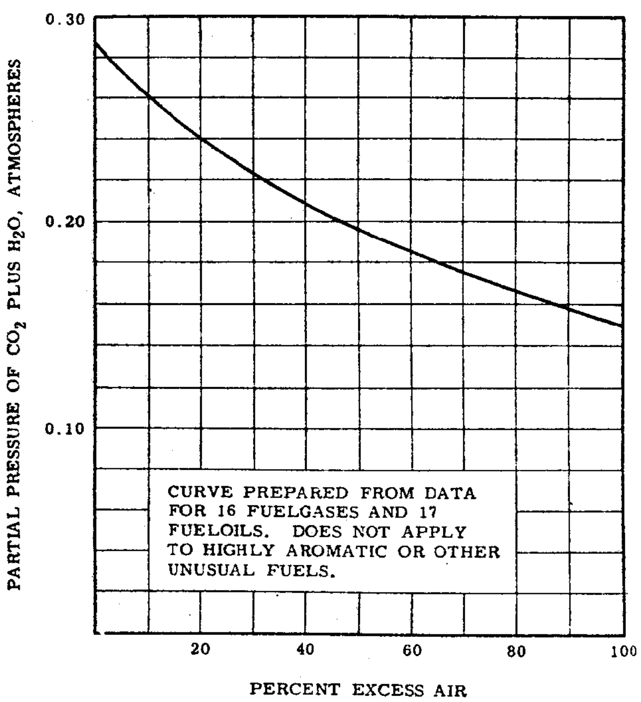

Figure 1-7. Partial pressure of CO_2 plus H_2O in flue gas.

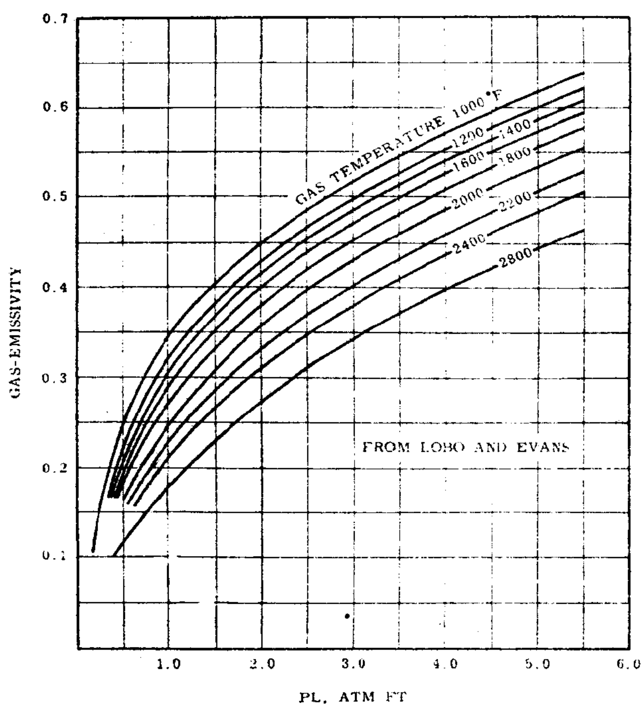

Figure 1-8. Gas emissivity.

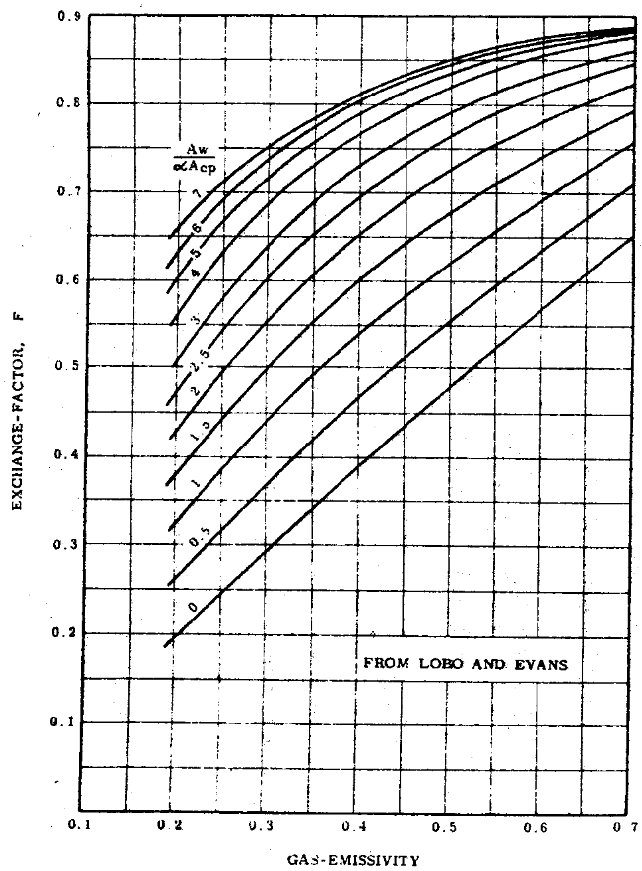

Figure 1-9. Overall radiant exchange factor.

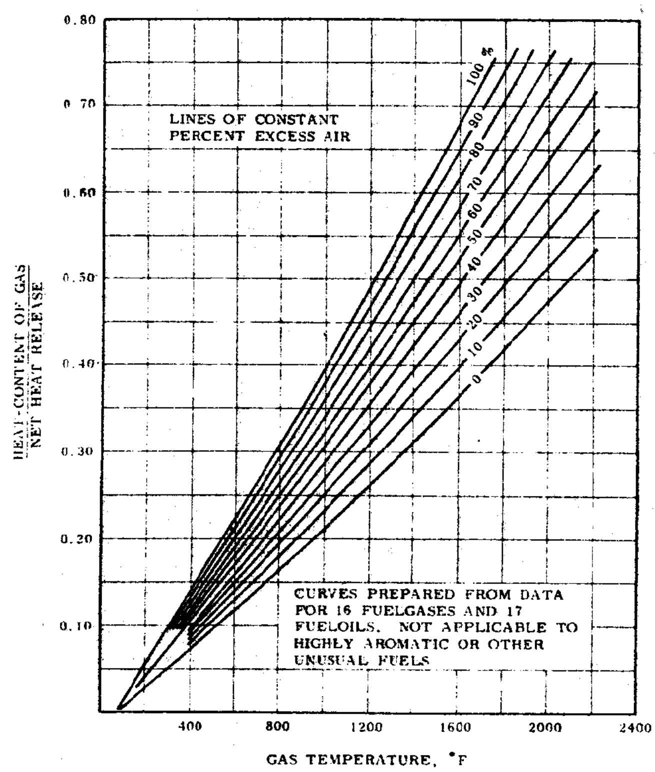

Figure 1-10. Heat content of flue gas.

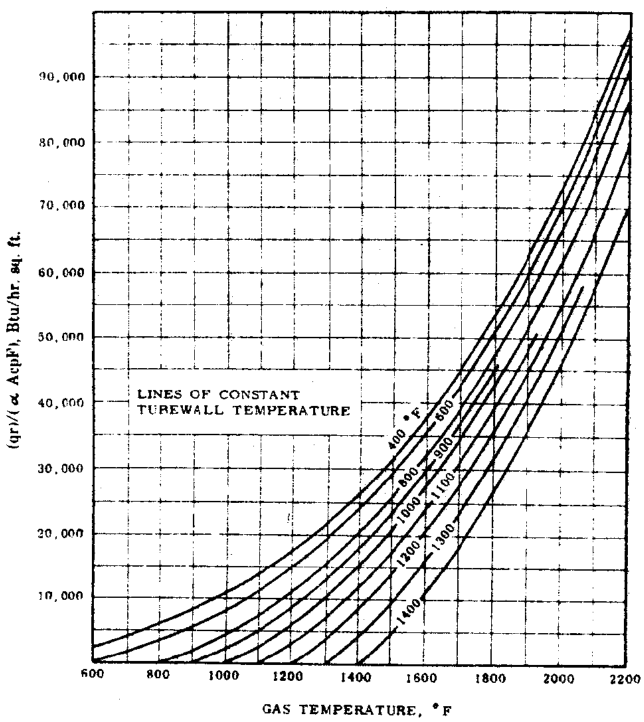

Figure 1-11. Total heat absorption in the radiant section.

When two points on opposite sides of the curve are obtained, join them by a straight line. The point of intersection of this line with the absorption curve is the correct firebox temperature. For this temperature calculate t_{g2}, read q_{g2}/q_n from Figure 1-10 and calculate q_R. This is the heat actually absorbed in the radiant section.

It is now necessary to check the computed radiant absorption to be sure it meets the design limitations. First, divide the heat absorption by the total exposed tube area to get the average heat flux. If it is higher than the allowable maximum, a new furnace with more tube surface must be selected and the rating repeated. If the actual heat flux is considerably below the allowable value, a smaller furnace should be considered.

Then, from enthalpy data on the process fluid, calculate the crossover temperature from convection to radiant section. If the computed temperature is significantly different from that assumed in starting the rating, it may be necessary to go back and repeat the calculation with a new value of average tube wall temperature. The tube wall

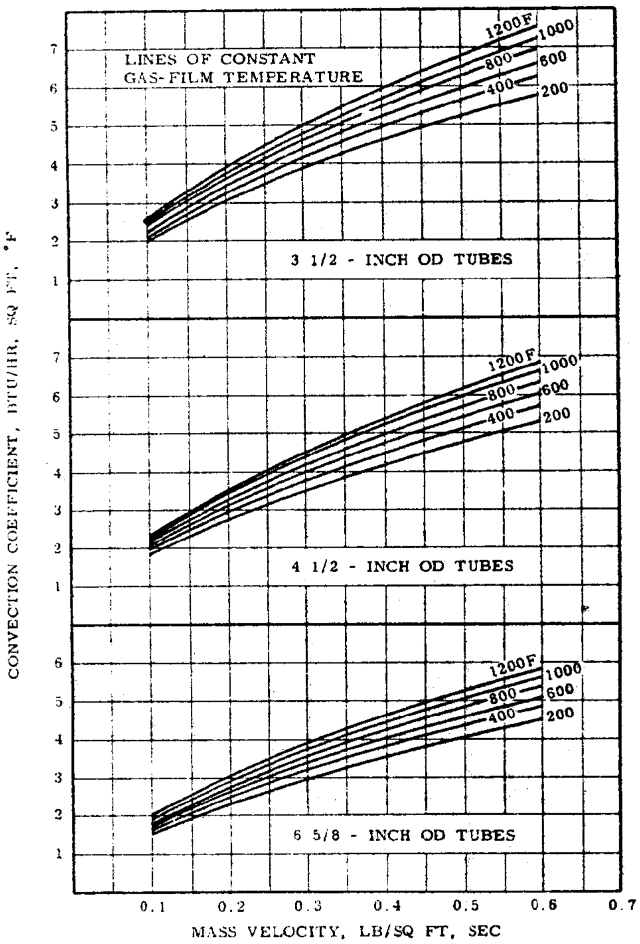

Figure 1-12. Flue gas convection coefficients for flow across staggered banks of bare tubes.

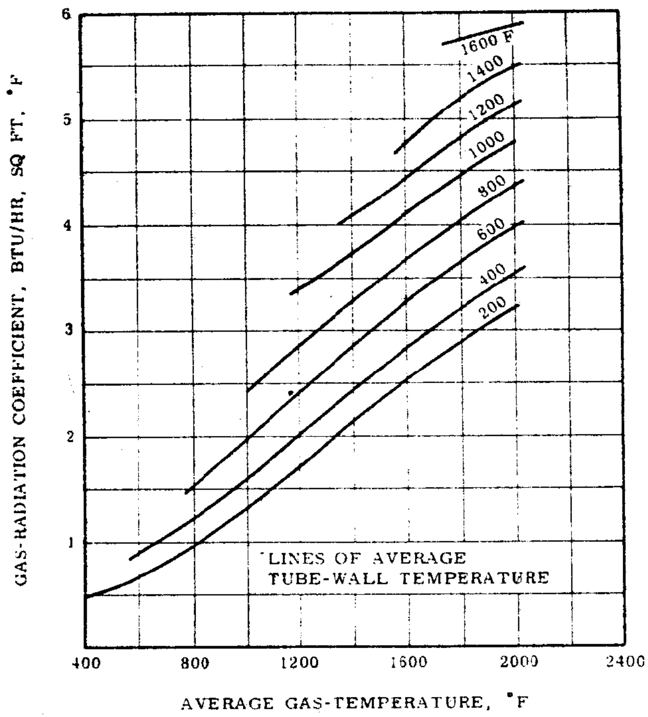

Figure 1-13. Gas radiation coefficients.

temperature, however, has such a small effect on the radiant absorption rate that such refinement is usually unnecessary.

There is one point to watch in regard to the crossover temperature. The fluid temperature determined as previously described corresponds to the transition from true convection tubes to shield tubes. In many heaters the shield tubes are part of the convection section, and the crossover piping is actually between the shield and radiant tubes. The temperature in the crossover piping is then higher than that calculated in the rating by an amount corresponding to heat absorption in the shield tubes.

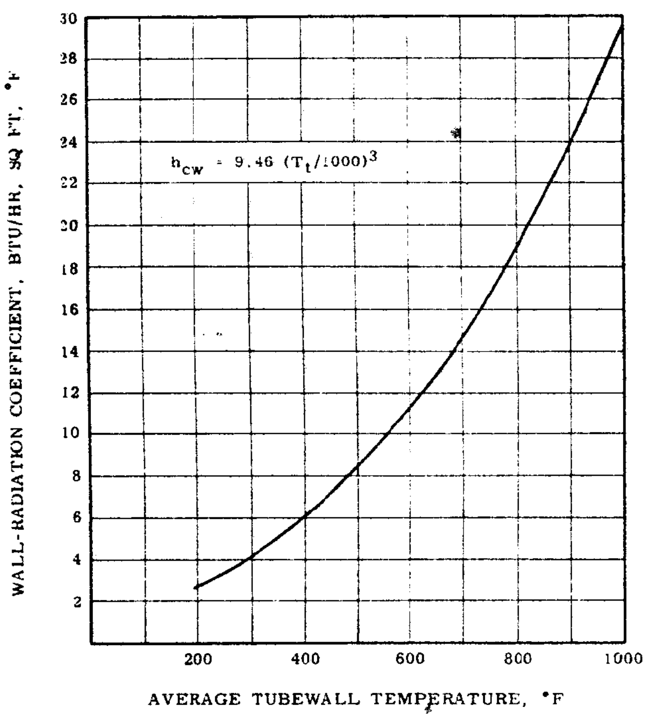

Figure 1-14. Coefficient for radiation from convection section walls.

Convection Section Rating

The convection section must pick up the difference between the heat absorbed in the radiant section and the total heater duty. The temperatures of the flue gas entering the convection section and of the process fluid leaving it have already been calculated as part of the radiant section rating. The incoming fluid temperature is set by the original design conditions. The flue gas temperature going to the stack is determined by an overall heat balance and Figure 1-10.

$$\frac{q_s}{q_n} = 1 + \left(\frac{q_a}{q_n}\right) + \left(\frac{q_f}{q_n}\right) - \left(\frac{q_R + q_C}{q_n}\right)$$

where:

q = Heat rate, Btu/hr.
s = Heat content of the gas
n = Net heat of combustion
a = Combustion air
f = Fuel heat rate
R = Radiant section heat rate
C = Convection section heat rate

Using these temperatures, calculate the average fluid temperature, the log mean temperature difference, the average gas temperature and the average gas film temperature.

Select a convection tube arrangement that will give a maximum flue gas mass velocity of about 0.3 to 0.4 pounds per square foot per second. Calculate the mass velocity, G. For precise ratings, calculate the gas film coefficient using Figures 1-12, 1-13 and 1-14 and equations for f, the factor for wall radiation in the convection section and h_c, the convective heat transfer coefficient. Calculate the intube coefficient using any applicable method. Then calculate the overall transfer coefficient from the equation for U_c, the convective heat transfer velocity.

$$f = \frac{h_{cw}}{h_{cc} + h_{cr} + h_{cw}} \left(\frac{A_{cw}}{A_{ct}}\right)$$

where:

f = Convection section wall radiation factor
h_{cc} = See Figure 1-12
h_{cr} = See Figure 1-13
h_{cw} = See Figure 1-14
A_{cw} = (Row-to-row tube spacing, ft.)(exposed tube area, ft.)
= wall area per row, sq. ft.
A_{ct} = (Number of rows)(surface area per tube, sq.ft.)
= tube area per row, sq. ft.

$$h_c = (1 + f)(h_{cc} + h_{cr})$$

$$U_c = \frac{(\text{Intube film coefficient})(h_c)}{(\text{Intube film coefficient}) + (h_c)}$$

For less critical ratings, read the overall design coefficient from Figure 1-15. In either case, finally determine the convection area from the usual heat transfer equation:

$$A_C = q_C / U_C(LMTD)$$

where:

A_C = Convection area, sq. ft.
q_C = Convective heat transfer rate in Btu/hr.
U_C = Overall convective heat transfer rate in Btu/hr. sq. ft. °F
$LMTD$ = Log mean temperature difference, °F

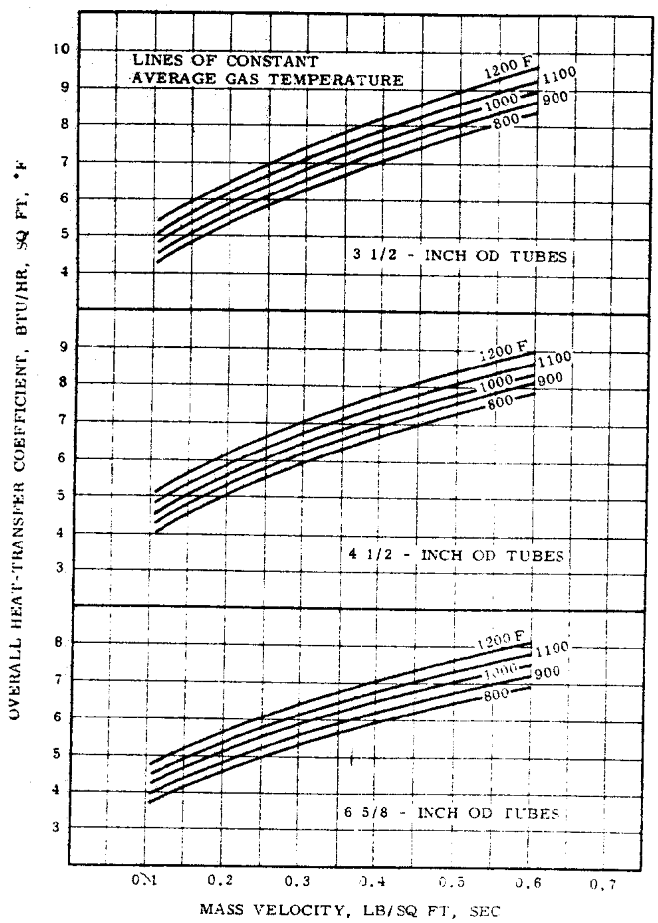

Figure 1-15. Overall heat transfer coefficients in the convection section.