CHAPTER 45
FURNACES

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45.1 SCOPE AND INTENT

This chapter has been prepared for the use of engineers with access to an electronic calculator and to standard engineering reference books, but not necessarily to a computer terminal. The intent is to provide information needed for the solution of furnace engineering problems in areas of design, performance analysis, construction and operating cost estimates, and improvement programs.

In selecting charts and formulas for problem solutions, some allowance has been made for probable error, where errors in calculations will be minor compared with errors in the assumptions on which calculations are based. Conscientious engineers are inclined to carry calculations to a far greater degree of accuracy than can be justified by probable errors in data assumed. Approximations have accordingly been allowed to save time and effort without adding to probable margins for error. The symbols and abbreviations used in this chapter are given in Table 45.1.

45.2 STANDARD CONDITIONS

Assuming that the user will be using English rather than metric units, calculations have been based on pounds, feet, Btu's, and degrees Fahrenheit, with conversion to metric units provided in the following text (see Table 45.2).

Assumed standard conditions include: ambient temperature for initial temperature of loads, for heat losses from furnace walls or open cooling of furnace loads—70°F.

Condition of air entering system for combustion or convection cooling: temperature, 70°F; absolute pressure, 14.7 psia; relative humidity, 60% at 70°F, for a water vapor content of about 1.4% by volume.

45.2.1 Probable Errors

Conscientious furnace engineers are inclined to carry calculations to a far greater degree of accuracy than can be justified by uncertainties in basic assumptions such as thermal properties of materials, system temperatures and pressures, radiation view factors and convection coefficients. Calculation procedures recommended in this chapter will, accordingly, include some approximations, identified in the text, that will result in probable errors much smaller than those introduced by basic assumptions, where such approximations will expedite problem solutions.

45.3 FURNACE TYPES

Furnaces may be grouped into two general types:

1. As a source of energy to be used elsewhere, as in firing steam boilers to supply process steam, or steam for electric power generation, or for space heating of buildings or open space
2. As a source of energy for industrial processes, other than for electric power

The primary concern of this chapter will be the design, operation, and economics of industrial furnaces, which may be classified in several ways:

By function:

Heating for forming in solid state (rolling, forging)
Melting metals or glass
Heat treatment to improve physical properties
Preheating for high-temperature coating processes, galvanizing, vitreous enameling, other coatings
Smelting for reduction of metallic ores
Firing of ceramic materials
Incineration

By method of load handling:

Batch furnaces for cyclic heating, including forge furnaces arranged to heat one end of a bar or billet inserted through a wall opening, side door, stationary-hearth-type car bottom designs
Continuous furnaces with loads pushed through or carried by a conveyor
Tilting-type furnace

To avoid the problem of door warpage or leakage in large batch-type furnaces, the furnace can be a refractory-lined box with an associated firing system, mounted above a stationary hearth, and arranged to be tilted around one edge of the hearth for loading and unloading by manual handling, forklift trucks, or overhead crane manipulators.

Table 45.1 Symbols and Abbreviations

A	area in ft^2
a	absorptivity for radiation, as fraction of black body factor for receiver temperature:
	a_g combustion gases
	a_w furnace walls
	a_s load surface
	a_m combined emissivity–absorptivity factor for source and receiver
C	specific heat in Btu/lb · °F or cal/g · °C
cfm	cubic feet per minute
D	diameter in ft or thermal diffusivity (k/dC)
d	density in lb/ft^3
e	emissivity for radiation as fraction of black-body factor for source temperature, with subscripts as for a above
F	factor in equations as defined in text
fpm	velocity in ft/min
G	mass velocity in $lb/ft^2 \cdot hr$
g	acceleration by gravity (32.16 ft/sec^2)
H	heat-transfer coefficient ($Btu/hr \cdot ft^2 \cdot °F$)
	H_r for radiation
	H_c for convection
	H_t for combined $H_r + H_c$
HHV	higher heating value of fuel
h	pressure head in units as defined
k	thermal conductivity (Btu/hr · ft · °F)
L	length in ft, as in effective beam length for radiation, decimal rather than feet and inches
LHV	lower heating value of fuel
ln	logarithm to base e
MTD	log mean temperature difference
N	a constant as defined in text
psi	pressure in lb/in^2
	psig, pressure above atmospheric
	psia, absolute pressure
Pr	Prandtl number ($\mu C/k$)
Q	heat flux in Btu/hr
R	thermal resistance (r/k) or ratio of external to internal thermal resistance (k/rH)
Re	Reynolds number (DG/μ)
r	radius or depth of heat penetration in ft
T	temperature in °F, except for radiation calculations where °S = (°F + 460)/100
	T_g, combustion gas temperature
	T_w, furnace wall temperature
	T_s, heated load surface
	T_c, core or unheated surface of load
t	time in hr
μ	viscosity in lb/hr · ft
wc	inches of water column as a measure of pressure
V	volume in ft^3
v	velocity in ft/sec
W	weight in lb
X	time factor for nonsteady heat transfer (tD/r^2)
x	horizontal coordinate
y	vertical coordinate
z	coordinate perpendicular to plane xy

For handling heavy loads by overhead crane, without door problems, the furnace can be a portable cover unit with integral firing and temperature control. Consider a cover-type furnace for annealing steel strip coils in a controlled atmosphere. The load is a stack of coils with a common vertical axis, surrounded by a protective inner cover and an external heating cover. To improve heat transfer parallel to coil laminations, they are loaded with open coil separators between them, with heat transferred from the inner cover to coil ends by a recirculating fan. To start the cooling cycle, the heating cover

Table 45.2 Conversion of Metric to English Units

Length	1 m = 3.281 ft
	1 cm = 0.394 in
Area	1 m^2 = 10.765 ft^2
Volume	1 m^3 = 35.32 ft^3
Weight	1 kg = 2.205 lb
Density	1 g/cm^3 = 62.43 lb/ft^2
Pressure	1 g/cm^2 = 2.048 lb/ft^2 = 0.0142 psi
Heat	1 kcal = 3.968 Btu
	1 kwh = 3413 Btu
Heat content	1 cal/g = 1.8 Btu/lb
	1 kcal/m^2 = 0.1123 Btu/ft^3
Heat flux	1 W/cm^2 = 3170 Btu/hr · ft^2
Thermal conductivity	$\frac{1 \text{ cal}}{\text{sec cm °C}} = \frac{242 \text{ Btu}}{\text{hr ft °F}}$
Heat transfer	$\frac{1 \text{ cal}}{\text{sec cm}^2 \text{ °C}} = \frac{7373 \text{ Btu}}{\text{hr ft}^2 \text{ °F}}$
Thermal diffusivity	$\frac{1 \text{ cal/sec} \cdot \text{cm} \cdot \text{°C}}{\text{C} \cdot \text{g/cm}^3} = \frac{3.874 \text{ Btu/hr} \cdot \text{ft} \cdot \text{°F}}{\text{C} \cdot \text{lb/ft}^3}$

is removed by an overhead crane, while atmosphere circulation by the base fan continues. Cooling may be enhanced by air-blast cooling of the inner cover surface.

For heating heavy loads of other types, such as weldments, castings, or forgings, car bottom furnaces may be used with some associated door maintenance problems. The furnace hearth is a movable car, to allow load handling by an overhead traveling crane. In one type of furnace, the door is suspended from a lifting mechanism. To avoid interference with an overhead crane, and to achieve some economy in construction, the door may be mounted on one end of the car and opened as the car is withdrawn. This arrangement may impose some handicaps in access for loading and unloading.

Loads such as steel ingots can be heated in pit-type furnaces, preferably with units of load separated to allow radiating heating from all sides except the bottom. Such a furnace would have a cover displaced by a mechanical carriage and would have a compound metal and refractory recuperator arrangement. Loads are handled by overhead crane equipped with suitable gripping tongs.

Continuous-Type Furnaces

The simplest type of continuous furnace is the hearth-type pusher furnace. Pieces of rectangular cross section are loaded side by side on a charge table and pushed through the furnace by an external mechanism. In the design shown, the furnace is fired from one end, counterflow to load travel, and is discharged through a side door by an auxiliary pusher lined up by the operator.

Furnace length is limited by thickness of the load and alignment of abutting edges, to avoid buckling up from the hearth.

A more complex design would provide multiple zone firing above and below the hearth, with recuperative air preheating.

Long loads can be conveyed in the direction of their length in a roller-hearth-type furnace. Loads can be bars, tubes, or plates of limited width, heated by direct firing, by radiant tubes, or by electric-resistor-controlled atmosphere, and conveyed at uniform speed or at alternating high and low speeds for quenching in line.

Sequential heat treatment can be accomplished with a series of chain or belt conveyors. Small parts can be loaded through an atmosphere seal, heated in a controlled atmosphere on a chain belt conveyor, discharged into an oil quench, and conveyed through a washer and tempering furnace by a series of mesh belts without intermediate handling.

Except for pusher-type furnaces, continuous furnaces can be self-emptying. To secure the same advantage in heating slabs or billets for rolling and to avoid scale loss during interrupted operation, loads can be conveyed by a walking-beam mechanism. Such a walking-beam-type slab heating furnace would have loads supported on water-cooled rails for over- and underfiring, and would have an overhead recuperator.

Thin strip materials, joined in continuous strand form, can be conveyed horizontally or the strands can be conveyed in a series of vertical passes by driven support rolls. Furnaces of this type can be incorporated in continuous galvanizing lines.

Unit loads can be individually suspended from an overhead conveyor, through a slot in the furnace roof, and can be quenched in line by lowering a section of the conveyor.

Small parts or bulk materials can be conveyed by a moving hearth, as in the rotary-hearth-type or tunnel kiln furnace. For roasting or incineration of bulk materials, the shaft-type furnace provides a simple and efficient system. Loads are charged through the open top of the shaft and descend by gravity to a discharge feeder at the bottom. Combustion air can be introduced at the bottom of the furnace and preheated by contact with the descending load before entering the combustion zone, where fuel is introduced through sidewalls. Combustion gases are then cooled by contact with the descending load, above the combustion zone, to preheat the charge and reduce flue gas temperature.

With loads that tend to agglomerate under heat and pressure, as in some ore-roasting operations, the rotary kiln may be preferable to the shaft-type furnace. The load is advanced by rolling inside an inclined cylinder. Rotary kilns are in general use for sintering ceramic materials.

Classification by Source of Heat

The classification of furnaces by source of heat is as follows:

Direct-firing with gas or oil fuels

Combustion of material in process, as by incineration with or without supplemental fuel

Internal heating by electrical resistance or induction in conductors, or dielectric heating of nonconductors

Radiation from electric resistors or radiant tubes, in controlled atmospheres or under vacuum

45.4 FURNACE CONSTRUCTION

The modern industrial furnace design has evolved from a rectangular or cylindrical enclosure, built up of refractory shapes and held together by a structural steel binding. Combustion air was drawn in through wall openings by furnace draft, and fuel was introduced through the same openings without control of fuel/air ratios except by the judgment of the furnace operator. Flue gases were exhausted through an adjacent stack to provide the required furnace draft.

To reduce air infiltration or outward leakage of combustion gases, steel plate casings have been added. Fuel economy has been improved by burner designs providing some control of fuel/air ratios, and automatic controls have been added for furnace temperature and furnace pressure. Completely sealed furnace enclosures may be required for controlled atmosphere operation, or where outward leakage of carbon monoxide could be an operating hazard.

With the steadily increasing costs of heat energy, wall structures are being improved to reduce heat losses or heat demands for cyclic heating. The selection of furnace designs and materials should be aimed at a minimum overall cost of construction, maintenance, and fuel or power over a projected service life. Heat losses in existing furnaces can be reduced by adding external insulation or rebuilding walls with materials of lower thermal conductivity. To reduce losses from intermittent operation, the existing wall structure can be lined with a material of low heat storage and low conductivity, to substantially reduce mean wall temperatures for steady operation and cooling rates after interrupted firing.

Thermal expansion of furnace structures must be considered in design. Furnace walls have been traditionally built up of prefired refractory shapes with bonded mortar joints. Except for small furnaces, expansion joints will be required to accommodate thermal expansion. In sprung arches, lateral expansion can be accommodated by vertical displacement, with longitudinal expansion taken care of by lateral slots at intervals in the length of the furnace. Where expansion slots in furnace floors could be filled by scale, slag, or other debris, they can be packed with a ceramic fiber that will remain resilient after repeated heating.

Differential expansion of hotter and colder wall surfaces can cause an inward-bulging effect. For stability in self-supporting walls, thickness must not be less than a critical fraction of height.

Because of these and economic factors, cast or rammed refractories are replacing prefired shapes for lining many types of large, high-temperature furnaces. Walls can be retained by spaced refractory shapes anchored to the furnace casing, permitting reduced thickness as compared to brick construction. Furnace roofs can be suspended by hanger tile at closer spacing, allowing unlimited widths.

Cast or rammed refractories, fired in place, will develop discontinuities during initial shrinkage that can provide for expansion from subsequent heating, to eliminate the need for expansion joints.

As an alternate to cast or rammed construction, insulating refractory linings can be gunned in place by jets of compressed air and retained by spaced metal anchors, a construction increasingly popular for stacks and flues.

Thermal expansion of steel furnace casings and bindings must also be considered. Where the furnace casing is constructed in sections, with overlapping expansion joints, individual sections can be separately anchored to building floors or foundations. For gas-tight casings, as required for controlled atmosphere heating, the steel structure can be anchored at one point and left free to expand elsewhere. In a continuous galvanizing line, for example, the atmosphere furnace and cooling zone can be anchored to the foundation near the casting pot, and allowed to expand toward the charge end.

45.5 FUELS AND COMBUSTION

Heat is supplied to industrial furnaces by combustion of fuels or by electrical power. Fuels now used are principally fuel oil and fuel gas. Because possible savings through improved design and operation are much greater for these fuels than for electric heating or solid fuel firing, they will be given primary consideration in this section.

Heat supply and demand may be expressed in units of *Btu* or *kcal* or as gallons or barrels of fuel oil, tons of coal or *kwh* of electric power. For the large quantities considered for national or world energy loads, a preferred unit is the "quad," one quadrillion or 10^{15} Btu. Conversion factors are:

$$
\begin{aligned}
1 \text{ quad} &= 10^{15} \text{ Btu} \\
&= 172 \times 10^6 \text{ barrels of fuel oil} \\
&= 44.34 \times 10^6 \text{ tons of coal} \\
&= 10^{12} \text{ cubic feet of natural gas} \\
&= 2.93 \times 10^{11} \text{ kwh electric power}
\end{aligned}
$$

At 30% generating efficiency, the fuel required to produce 1 quad of electrical energy is 3.33 quads. One quad fuel is accordingly equivalent to 0.879×10^{11} kwh net power.

Fuel demand, in the United States during recent years, has been about 75 quads per year from the following sources:

Coal	15 quads
Fuel oil	
Domestic	18 quads
Imported	16 quads
Natural gas	23 quads
Other, including nuclear	3 quads

Hydroelectric power contributes about 1 quad net additional. Combustion of waste products has not been included, but will be an increasing fraction of the total in the future.

Distribution of fuel demand by use is estimated at:

Power generation	20 quads
Space heating	11 quads
Transportation	16 quads
Industrial, other than power	25 quads
Other	4 quads

Net demand for industrial furnace heating has been about 6%, or 4.56 quads, primarily from gas and oil fuels.

The rate at which we are consuming our fossil fuel assets may be calculated as (annual demand)/(estimated reserves). This rate is presently highest for natural gas, because, besides being available at wellhead for immediate use, it can be transported readily by pipeline and burned with the simplest type of combustion system and without air pollution problems. It has also been delivered at bargain prices, under federal rate controls.

As reserves of natural gas and fuel oil decrease, with a corresponding increase in market prices, there will be an increasing demand for alternative fuels such as synthetic fuel gas and fuel oil, waste materials, lignite, and coal.

Synthetic fuel gas and fuel oil are now available from operating pilot plants, but at costs not yet competitive.

As an industrial fuel, coal is primarily used for electric power generation. In the form of metallurgical coke, it is the source of heat and the reductant in the blast furnace process for iron ore reduction, and as fuel for cupola furnaces used to melt foundry iron. Powdered coal is also being used as fuel and reductant in some new processes for solid-state reduction of iron ore pellets to make synthetic scrap for steel production.

Since the estimated life of coal reserves, particularly in North America, is so much greater than for other fossil fuels, processes for conversion of coal to fuel gas and fuel oil have been developed almost to the commercial cost level, and will be available whenever they become economical. Processes for coal gasification, now being tried in pilot plants, include:

1. *Producer Gas.* Bituminous coal has been commercially converted to fuel gas of low heating value, around 110 Btu/scf LHV, by reacting with insufficient air for combustion and steam as a source of hydrogen. Old producers delivered a gas containing sulfur, tar volatiles, and suspended ash, and have been replaced by cheap natural gas. By reacting coal with a mixture of oxygen and steam, and removing excess carbon dioxide, sulfur gases, and tar, a clean fuel gas of about 300 Btu/scf LHV can be supplied. Burned with air preheated to 1000°F and with a flue gas temperature of 2000°F, the available heat is about 0.69 HHV, about the same as for natural gas.
2. *Synthetic Natural Gas.* As a supplement to dwindling natural gas supplies, a synthetic fuel gas of similar burning characteristics can be manufactured by adding a fraction of hydrogen to the product of the steam–oxygen gas producer and reacting with carbon monoxide at high temperature and pressure to produce methane. Several processes are operating successfully on a pilot plant scale, but with a product costing much more than market prices for natural gas. The process may yet be practical for extending available natural gas supplies by a fraction, to maintain present market demands. For gas mixtures or synthetic gas supplies to be interchangeable with present gas fuels, without readjustment of fuel/air ratio controls, they must fit the Wobbe Index:

$$\frac{\text{HHV Btu/scf}}{(\text{specific gravity})^{0.5}}$$

The fuel gas industry was originally developed to supply fuel gas for municipal and commercial lighting systems. Steam was passed through incandescent coal or coke, and fuel oil vapors were added to provide a luminous flame. The product had a heating value of around 500 HHV, and a high carbon monoxide content, and was replaced as natural gas or coke oven gas became available. Coke oven gas is a by-product of the manufacture of metallurgical coke that can be treated to remove sulfur compounds and volatile tar compounds to provide a fuel suitable for pipeline distribution. Blast furnace gas can be used as an industrial or steam-generating fuel, usually after enrichment with coke oven gas. Gas will be made from replaceable sources such as agricultural and municipal wastes, cereal grains, and wood, as market economics for such products improve.

Heating values for fuels containing hydrogen can be calculated in two ways:

1. Higher heating value (HHV) is the total heat developed by burning with standard air in a ratio to supply 110% of net combustion air, cooling products to ambient temperature, and condensing all water vapor from the combustion of hydrogen.
2. Lower heating value (LHV) is equal to HHV less heat from the condensation of water vapor. It provides a more realistic comparison between different fuels, since flue gases leave most industrial processes well above condensation temperatures.

HHV factors are in more general use in the United States, while LHV values are more popular in most foreign countries.

For example, the HHV value for hydrogen as fuel is 319.4 Btu/scf, compared to a LHV of 270.2.

The combustion characteristics for common fuels are tabulated in Table 45.3, for combustion with 110% standard air. Weights in pounds per 10^6 Btu HHV are shown, rather than corresponding volumes, to expedite calculations based on mass flow. Corrections for flue gas and air temperatures other than ambient are given in charts to follow.

The heat released in a combustion reaction is:

$$\text{total heats of formation of combustion products} - \text{total heats of formation of reactants}$$

Heats of formation can be conveniently expressed in terms of Btu per pound mol, with the pound mol for any substance equal to a weight in pounds equal to its molecular weight. The heat of formation for elemental materials is zero. For compounds involved in common combustion reactions, values are shown in Table 45.4.

Data in Table 45.4 can be used to calculate the higher and lower heating values of fuels. For methane:

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

Table 45.3 Combustion Characteristics of Common Fuels

Fuel	Btu/scf	Weight in lb/10^6 Btu Fuel	Air	Flue Gas
Natural gas (SW U.S.)	1073	42	795	837
Coke oven gas	539	57	740	707
Blast furnace gas	92	821	625	1446
Mixed blast furnace and coke oven gas:				
Ratio CO/BF 1/1	316	439	683	1122
1/3	204	630	654	1284
1/10	133	752	635	1387
Hydrogen	319	16	626	642
	Btu/lb			
No. 2 fuel oil	19,500	51	810	861
No. 6 fuel oil	18,300	55	814	
With air atomization				869
With steam atomization at 3 lb/gal				889
Carbon	14,107	71	910	981

HHV

$$169{,}290 + (2 \times 122{,}976) - 32{,}200 = 383{,}042 \text{ Btu/lb} \cdot \text{mol}$$
$$383{,}042/385 = 995 \text{ Btu/scf}$$

LHV

$$169{,}290 + (2 \times 104{,}040) - 32{,}200 = 345{,}170 \text{ Btu/lb} \cdot \text{mol}$$
$$345{,}170/385 = 897 \text{ Btu/scf}$$

Available heats from combustion of fuels, as a function of flue gas and preheated air temperatures, can be calculated as a fraction of the HHV. The net ratio is one plus the fraction added by preheated air less the fraction lost as sensible heat and latent heat of water vapor, from combustion of hydrogen, in flue gas leaving the system.

Available heats can be shown in chart form, as in the following figures for common fuels. On each chart, the curve on the right is the fraction of HHV available for combustion with 110% cold air, while the curve on the left is the fraction added by preheated air, as functions of air or flue gas temperatures. For example, the available heat fraction for methane burned with 110% air preheated to 1000°F, and with flue gas out at 2000°F, is shown in Fig. 45.1: 0.41 + 0.18 − 0.59 HHV.

Values for other fuels are shown in charts that follow:

Fig. 45.2, fuel oils with air or steam atomization
Fig. 45.3, by-product coke oven gas
Fig. 45.4, blast furnace gas
Fig. 45.5, methane

Table 45.4 Heats of Formation

Material	Formula	Molecular Weight	Heats of Formation (Btu/lb · mol[a])
Methane	CH_4	16	32,200
Ethane	C_2H_6	30	36,425
Propane	C_3H_8	44	44,676
Butane	C_4H_{10}	58	53,662
Carbon monoxide	CO	28	47,556
Carbon dioxide	CO_2	44	169,290
Water vapor	H_2O	18	104,040
Liquid water			122,976

[a] The volume of 1 lb mol, for any gas, is 385 scf.

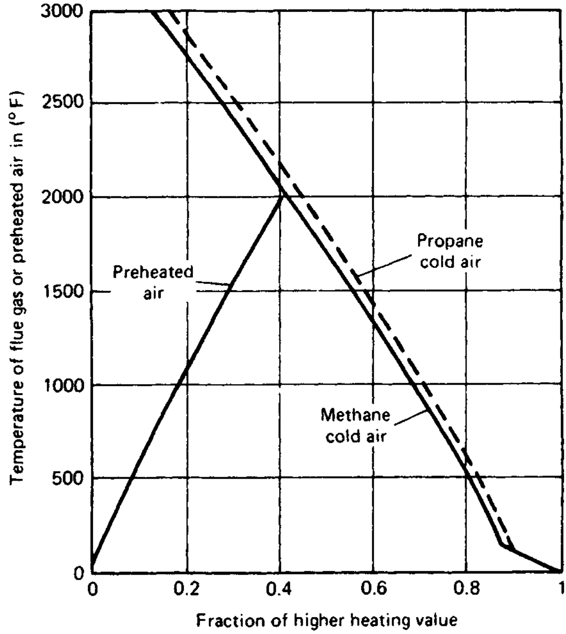


Fig. 45.1 Available heat for methane and propane combustion. Approximate high and low limits for commercial natural gas.[1]

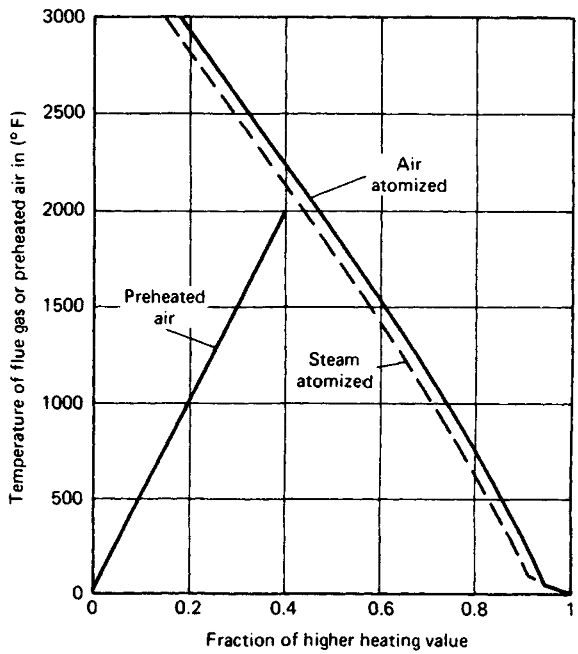


Fig. 45.2 Available heat ratios for fuel oils with air or steam atomization.[1]

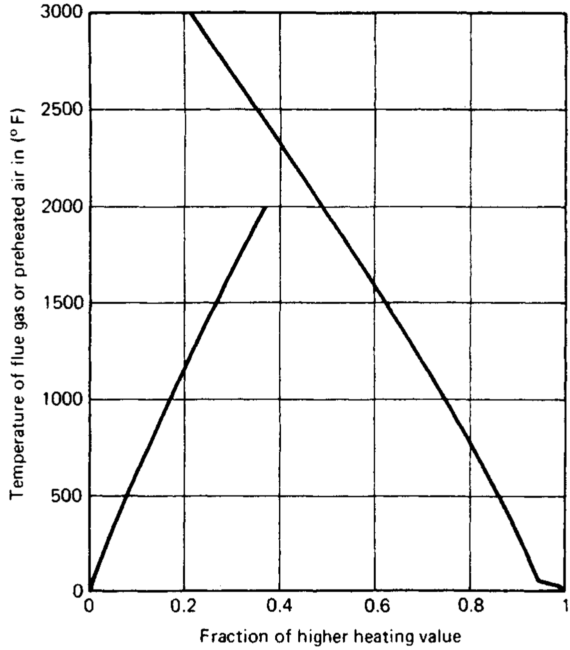


Fig. 45.3 Available heat ratios for by-product coke oven gas.[1]

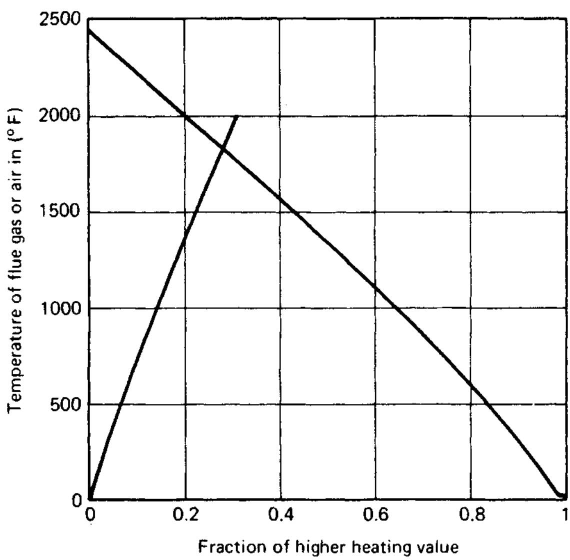


Fig. 45.4 Available heat ratios for blast furnace gas.[1]

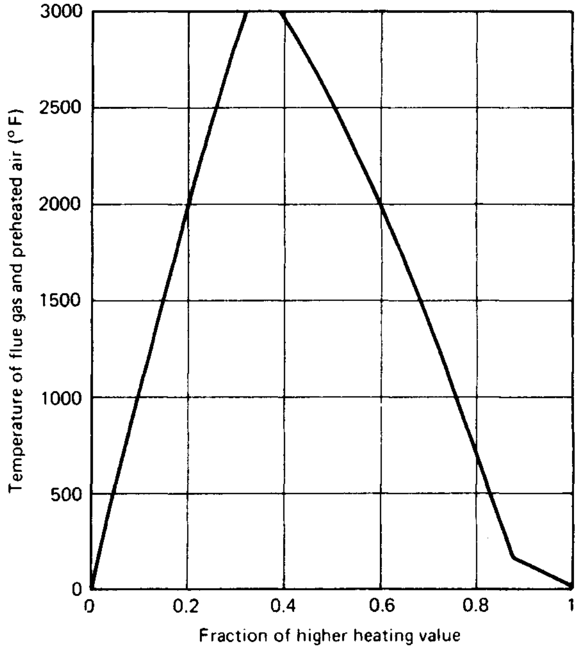


Fig. 45.5 Available heat ratios for combustion of methane with 110% air containing 35% O_2.[1]

For combustion with other than 110% of net air demand, the corrected available heat can be calculated as follows. For methane with preheated air at 1000°F and flue gas out at 2000°F and 150% net air supply:

Available heat from Fig. 58.1	0.59
Add excess air + 0.18 (1.5 − 1.1) =	0.072
− 0.41 (1.5 − 1.1) =	−0.164
Net total at 150%	0.498

Available heats for fuel gas mixtures can be calculated by adding the fractions for either fuel and dividing by the combined volume. For example, a mixture of one-quarter coke oven gas and three-quarters blast furnace gas is burned with 110% combustion air preheated to 1000°F, and with flue gas out at 2000°F. Using data from Table 45.3 and Figs. 45.3 and 45.4:

$$\begin{aligned} &\text{CO } (539 \times 0.25 = 134.75)\ (0.49 + 0.17) = 88.93 \\ &\text{BF } (92 \times 0.75 = \underline{69.00})\ (0.21 + 0.144) = \underline{24.43} \\ &\text{HHV } 203.75 \qquad \text{Available} = 113.36 \\ &\text{Net: } 113.36/203.75 = 0.556 \text{ combined HHV} \end{aligned}$$

45.6 OXYGEN ENRICHMENT OF COMBUSTION AIR

The available heats of furnace fuels can be improved by adding oxygen to combustion air. Some studies have been based on a total oxygen content of 35%, which can be obtained by adding 21.5 scf pure oxygen or 25.45 scf of 90% oxygen per 100 scf of dry air. The available heat ratios are shown in the chart in Fig. 45.5.

At present market prices, the power needed to concentrate pure oxygen for enrichment to 35% will cost more than the fuel saved, even with metallurgical oxygen from an in-plant source. As plants are developed for economical concentration of oxygen to around 90%, the cost balance may become favorable for very-high-temperature furnaces.

In addition to fuel savings by improvement of available heat ratios, there will be additional savings in recuperative furnaces by increasing preheated air temperature at the same net heat demand, de-

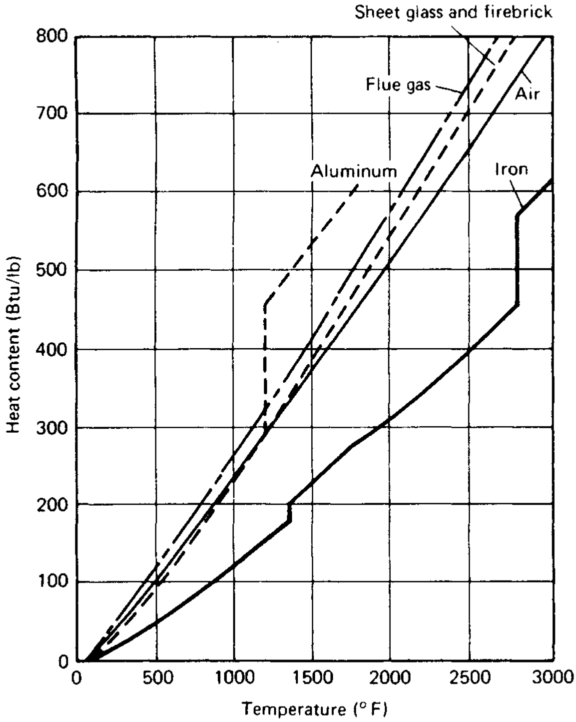


Fig. 45.6 Heat content of materials at temperature.[1]

pending on the ratio of heat transfer by convection to that by gas radiation in the furnace and recuperator.

45.7 THERMAL PROPERTIES OF MATERIALS

The heat content of some materials heated in furnaces or used in furnace construction is shown in the chart in Fig. 45.6, in units of Btu/lb. Vertical lines in curves represent latent heats of melting or other phase transformations. The latent heat of evaporation for water in flue gas has been omitted from the chart. The specific heat of liquid water is, of course, about 1.

Thermal conductivities in English units are given in reference publications as: $(\mathrm{Btu}/(\mathrm{ft}^2 \cdot \mathrm{hr}))/(°\mathrm{F}/\mathrm{in.})$ or as $(\mathrm{Btu}/(\mathrm{ft}^2 \cdot \mathrm{hr}))/(°\mathrm{F}/\mathrm{ft})$. To keep dimensions consistent, the latter term, abbreviated to $k = \mathrm{Btu}/\mathrm{ft} \cdot \mathrm{hr} \cdot °\mathrm{F}$ will be used here. Values will be $\frac{1}{12}$th of those in terms of °F/in.

Thermal conductivities vary with temperature, usually inversely for iron, steel, and some alloys, and conversely for common refractories. At usual temperatures of use, average values of k in $\mathrm{Btu}/(\mathrm{ft} \cdot \mathrm{hr} \cdot °\mathrm{F})$ are in Table 45.5.

Table 45.5 Average Values of k (Btu/ft · hr · °F)

	Mean Temperature (°F)				
	100	1000	1500	2000	2500
Steel, SAE 1010	33	23	17	17	
Type HH HRA	8	11	14	16	
Aluminum	127	133			
Copper	220	207	200		
Brass, 70/30	61	70			
Firebrick	0.81	0.82	0.85	0.89	0.93
Silicon carbide	11	10	9	8	6
Insulating firebrick	0.12	0.17	0.20	0.24	

To expedite calculations for nonsteady conduction of heat, it is convenient to use the factor for "thermal diffusivity," defined as

$$D = \frac{k}{dC} = \frac{\text{thermal conductivity}}{\text{density} \times \text{specific heat}}$$

in consistent units. Values for common furnace loads over the usual range of temperatures for heating are:

Carbon steels, 70–1650°F	0.32
70–2300°F	0.25
Low-alloy steels, 70–2000°F	0.23
Stainless steels, 70–2000°F	
300 type	0.15
400 type	0.20
Aluminum, 70–1000°F	3.00
Brass, 70/30, 70–1500°F	1.20

In calculating heat losses through furnace walls with multiple layers of materials with different thermal conductivities, it is convenient to add thermal resistance $R = r/k$, where r is thickness in ft. For example,

	r	k	r/k
9-in. firebrick	0.75	0.9	0.833
4½-in. insulating firebrick	0.375	0.20	1.875
2¼-in. block insulation	0.208	0.15	1.387
Total R for wall materials			4.095

Overall thermal resistance will include the factor for combined radiation and convection from the outside of the furnace wall to ambient temperature. Wall losses as a function of wall surface temperature, for vertical surfaces in still air, are shown in Fig. 45.7, and are included in the overall heat loss data for furnace walls shown in the chart in Fig. 45.8.

The chart in Fig. 45.9 shows the thermodynamic properties of air and flue gas, over the usual range of temperatures, for use in heat-transfer and fluid flow problems. Data for other gases, in formula form, are available in standard references.

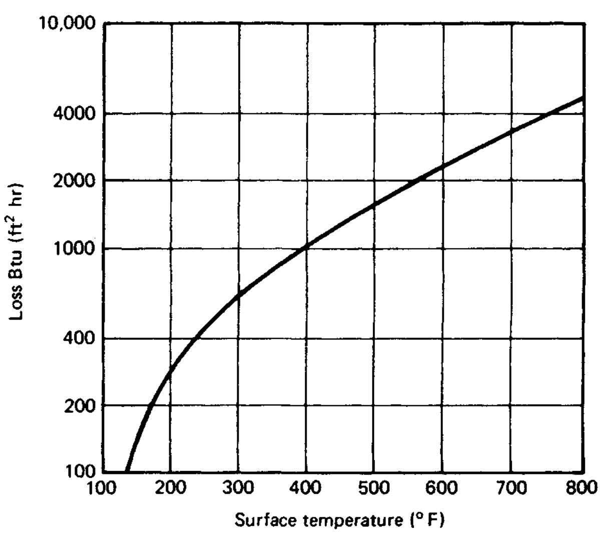


Fig. 45.7 Furnace wall losses as a function of surface temperature.[1]

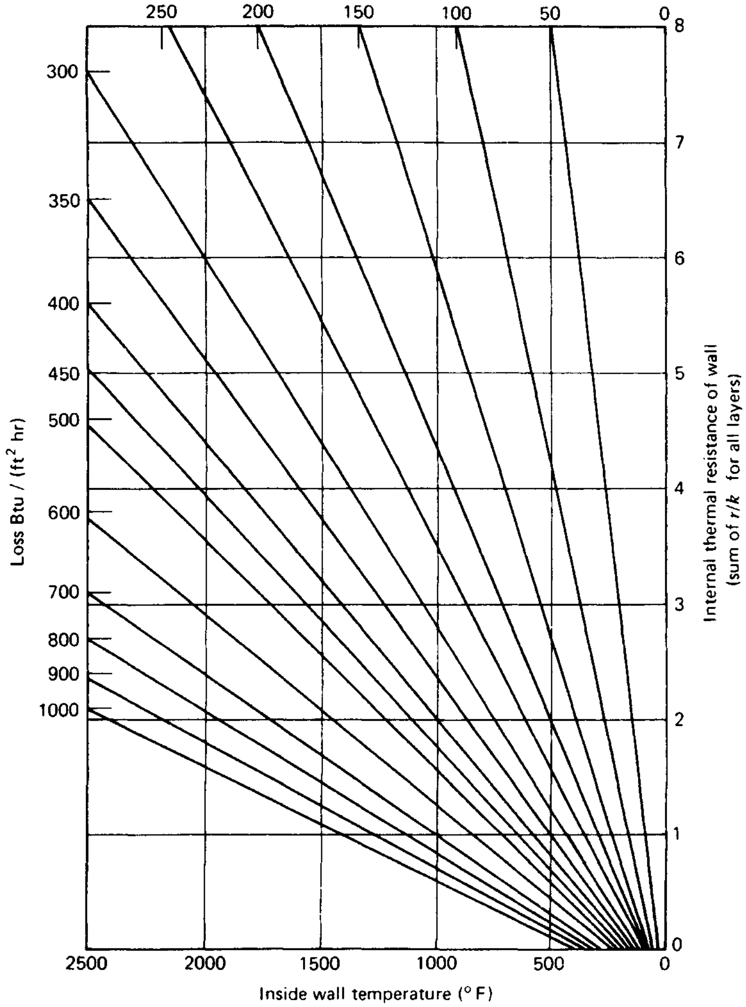


Fig. 45.8 Furnace wall losses as a function of composite thermal resistance.[1]

Linear coefficients of thermal expansion are the fractional changes in length per °F change in temperature. Coefficients in terms of $10^6 \times$ net values are listed below for materials used in furnace construction and for the usual range of temperatures:

Carbon steel	9
Cast HRA	10.5
Aluminum	15.6
Brass	11.5
Firebrick, silicon carbide	3.4
Silica brick	3.4

Coefficients for cubical expansion of solids are about $3 \times$ linear coefficients. The cubical coefficient for liquid water is about 185×10^{-6}.

45.8 HEAT TRANSFER

Heat may be transmitted in industrial furnaces by radiation—gas radiation from combustion gases to furnace walls or direct to load, and solid-state radiation from walls, radiant tubes, or electric heating

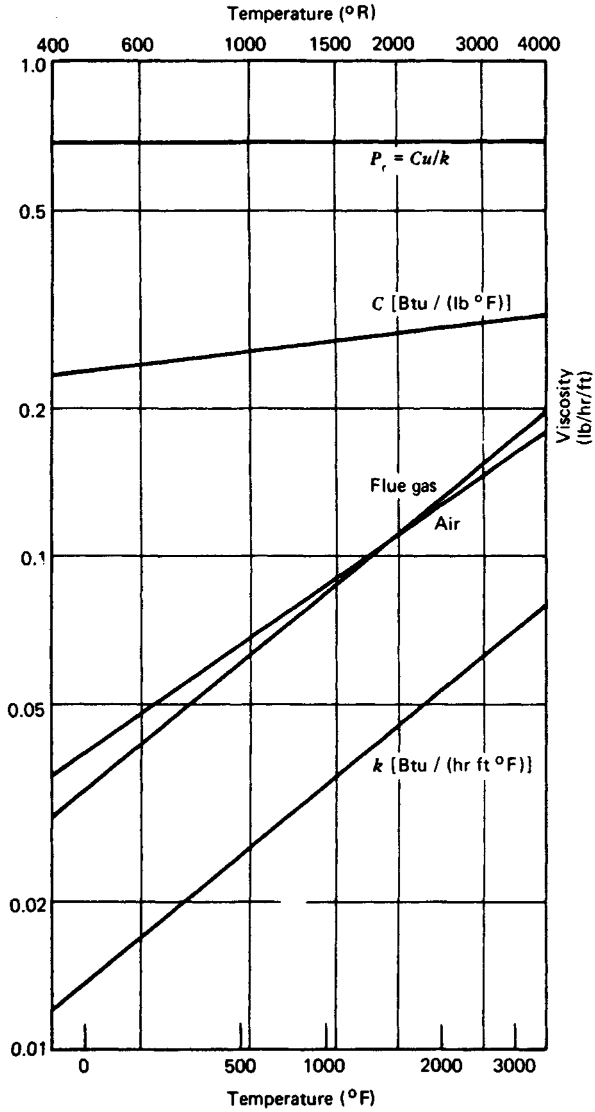


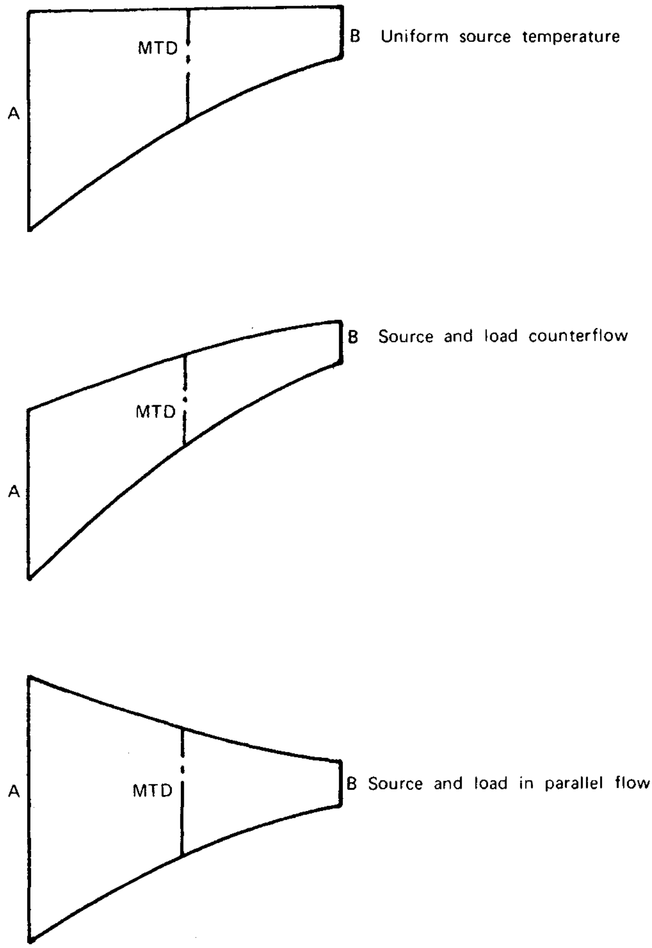
Fig. 45.9 Thermodynamic properties of air and flue gas.[1]

elements to load—or by convection—from combustion gases to walls or load. Heat may be generated inside the load by electrical resistance to an externally applied voltage or by induction, with the load serving as the secondary circuit in an alternating current transformer. Nonconducting materials may be heated by dielectric heating from a high-frequency source.

Heat transfer in the furnace structure or in solid furnace loads will be by conduction. If the temperature profile is constant with time, the process is defined as "steady-state conduction." If temperatures change during a heating cycle, it is termed "non-steady-state conduction."

Heat flow is a function of temperature differentials, usually expressed as the "log mean temperature difference" with the symbol MTD. MTD is a function of maximum and minimum temperature differences that can vary with position or time. Three cases encountered in furnace design are illustrated in Fig. 45.10. If the maximum differential, in any system of units, is designated as A and the minimum is designated by B:

$$\text{MTD} = \frac{A - B}{\ln (A/B)}$$



General formula: $\text{MTD} = \dfrac{A - B}{\ln (A/B)}$

where A = maximum temperature difference
B = minimum temperature difference

$\ln A/B = 2.3 \log_{10} (A/B)$

Fig. 45.10 Diagrams of log mean temperature difference (MTD).[1]

45.8.1 Solid-State Radiation

"Black-body" surfaces are those that absorb all radiation received, with zero reflection, and exist only as limits approached by actual sources or receivers of solid radiation. Radiation between black bodies is expressed by the Stefan–Boltzmann equation:

$$Q/A = N(T^4 - T_0^4) \quad \text{Btu/hr} \cdot \text{ft}^2$$

where N is the Stefan–Boltzmann constant, now set at about 0.1713×10^{-8} for T and T_0, source and receiver temperatures, in °R. Because the fourth powers of numbers representing temperatures in °R are large and unwieldy, it is more convenient to express temperatures in °S, equivalent to (°F + 460)/100. The constant N is then reduced to 0.1713.

With source and receiver temperatures identified as T_s and T_r in °S, and with allowance for emissivity and view factors, the complete equation becomes:

$$Q/A = 0.1713 \times em \times Fr(T_s^4 - T_r^4) \quad \text{Btu/hr} \cdot \text{ft}^2$$

at the receiving surface,

where em = combined emissivity and absorptivity factors for source and receiving surfaces
Fr = net radiation view factor for receiving surface
T_s and T_r = source and receiving temperature in °S

The factor em will be somewhat less than e for the source or a for the receiving surface, and can be calculated:

$$em = 1\Big/\frac{1}{a} + \frac{A_r}{A_s}\left(\frac{1}{e} - 1\right)$$

where a = receiver absorptivity at T_r
A_r/A_s = area ratio, receiver/source
e = source emissivity at T_s

45.8.2 Emissivity—Absorptivity

While emissivity and absorptivity values for solid materials vary with temperatures, values for materials commonly used as furnace walls or loads, in the usual range of temperatures, are:

Refractory walls	0.80–0.90
Heavily oxidized steel	0.85–0.95
Bright steel strip	0.25–0.35
Brass cake	0.55–0.60
Bright aluminum strip	0.05–0.10
Hot-rolled aluminum plate	0.10–0.20
Cast heat-resisting alloy	0.75–0.85

For materials such as sheet glass, transparent in the visible light range, radiation is reflected at both surfaces at about 4% of incident value, with the balance absorbed or transmitted. Absorptivity decreases with temperature, as shown in Fig. 45.11.

The absorptivity of liquid water is about 0.96.

45.8.3 Radiation Charts

For convenience in preliminary calculations, black-body radiation, as a function of temperature in °F, is given in chart form in Fig. 45.12. The value for the receiver surface is subtracted from that of the source to find net interchange for black-body conditions, and the result is corrected for emissivity and view factors. Where heat is transmitted by a combination of solid-state radiation and convection, a black-body coefficient, in Btu/hr · °F, is shown in the chart in Fig. 45.13. This can be added to the convection coefficient for the same temperature interval, after correcting for emissivity and view factor, to provide an overall coefficient (H) for use in the formula

$$Q/A = H(T - T_r)$$

45.8.4 View Factors for Solid-State Radiation

For a receiving surface completely enclosed by the source of radiation, or for a flat surface under a hemispherical radiating surface, the view factor is unity. Factors for a wide range of geometrical configurations are given in available references. For cases commonly involved in furnace heat-transfer calculations, factors are shown by the following charts.

For two parallel planes, with edges in alignment as shown in Fig. 45.14a, view factors are given in Fig. 45.15 in terms of ratios of x, y, and z. For two surfaces intersecting at angle of 90° at a common edge, the view factor is shown in Fig. 45.16. If surfaces do not extend to a common intersection, the view factor for the missing areas can be calculated and deducted from that with surfaces extended as in the figure, to find the net value for the remaining areas.

For spaced cylinders parallel to a furnace wall, as shown in Fig. 45.17, the view factor is shown in terms of diameter and spacing, including wall reradiation. For tubes exposed on both sides to source or receiver radiation, as in some vertical strip furnaces, the following factors apply if sidewall reradiation is neglected:

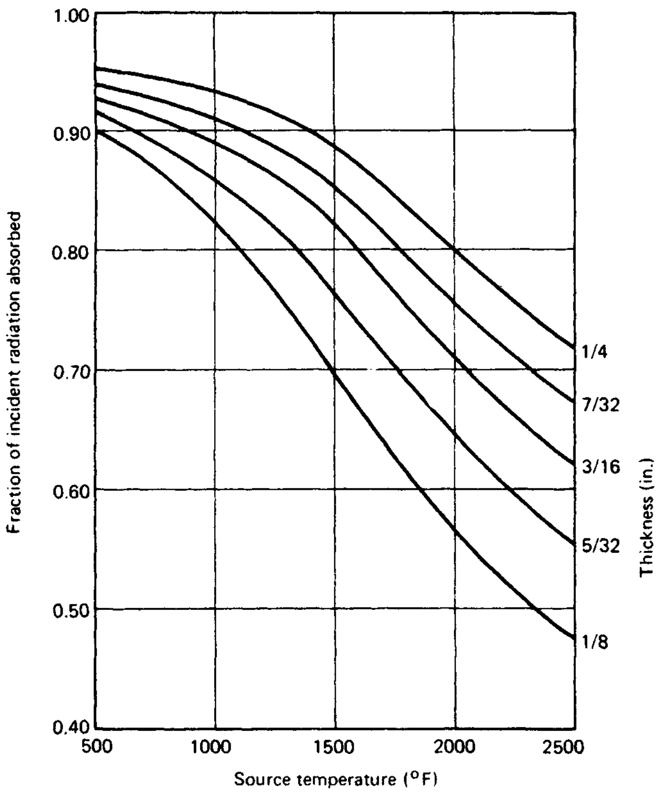


Fig. 45.11 Radiation absorptivity of sheet glass with surface reflection deducted.[1]

Ratio C/D	1.0	1.5	2.0	2.5	3.0
Factor	0.67	0.793	0.839	0.872	0.894

For ribbon-type electric heating elements, mounted on a back-up wall as shown in Fig. 45.18, exposure factors for projected wall area and for total element surface area are shown as a function of the (element spacing)/(element width) ratio. Wall reradiation is included, but heat loss through the back-up wall is not considered. The emission rate from resistor surface will be W/in.^{2} = $Q/491A$, where

$$\frac{Q}{A} = \frac{\text{Btu/hr}}{\text{ft}^2}$$

For parallel planes of equal area, as shown in Fig. 45.14, connected by reradiating walls on four sides, the exposure factor is increased as shown in Fig. 45.19. Only two curves, for $z/x = 1$ and $z/x = 10$ have been plotted for comparison with Fig. 45.13.

45.8.5 Gas Radiation

Radiation from combustion gases to walls and load can be from luminous flames or from nonluminous products of combustion. Flame luminosity results from suspended solids in combustion gases, either incandescent carbon particles or ash residues, and the resulting radiation is in a continuous spectrum corresponding to that from solid-state radiation at the same source temperature. Radiation from nonluminous gases is in characteristic bands of wavelengths, with intensity depending on depth and density of the radiating gas layer, its chemical composition, and its temperature.

For combustion of hydrocarbon gases, flame luminosity is from carbon particles formed by cracking of unburned fuel during partial combustion, and is increased by delayed mixing of fuel and air in the combustion chamber. With fuel and air thoroughly premixed before ignition, products of combustion will be nonluminous in the range of visible light, but can radiate strongly in other

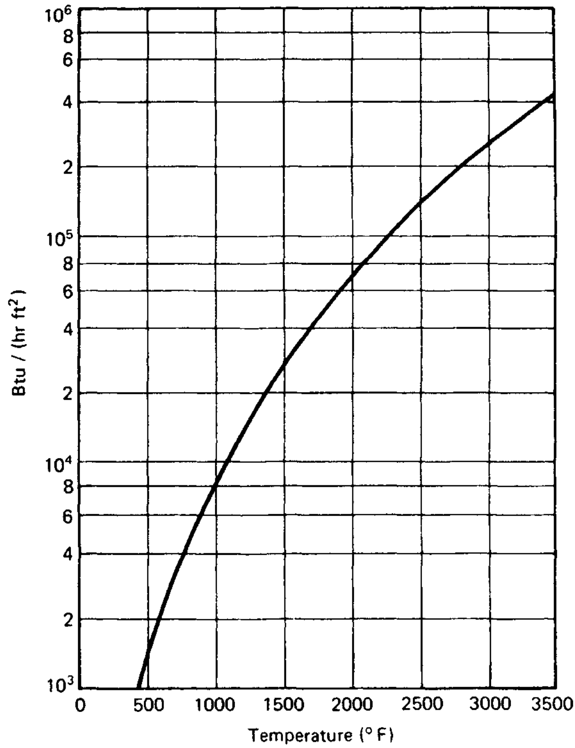


Fig. 45.12 Black-body radiation as function of load surface temperature.

wavelength bands for some products of combustion including carbon dioxide and water vapor. Published data on emissivities of these gases show intensity of radiation as a function of temperature, partial pressure, and beam length. The combined emissivity for mixtures of carbon dioxide and water vapor requires a correction factor for mutual absorption. To expedite calculations, a chart has been prepared for the overall emissivity of some typical flue gases, including these correction factors. The chart in Fig. 45.20 has been calculated for products of combustion of methane with 110% of net air demand, and is approximately correct for other hydrocarbon fuels of high heating value, including

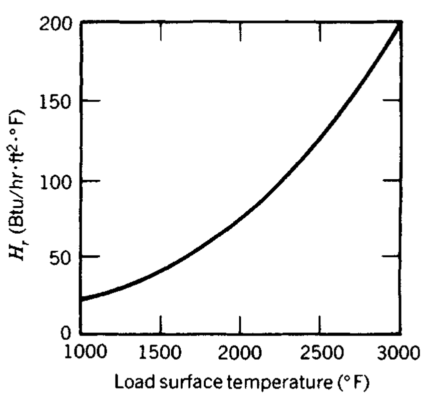


Fig. 45.13 Black-body radiation coefficient for source temperature uniform at 50–105° above final load surface temperature.

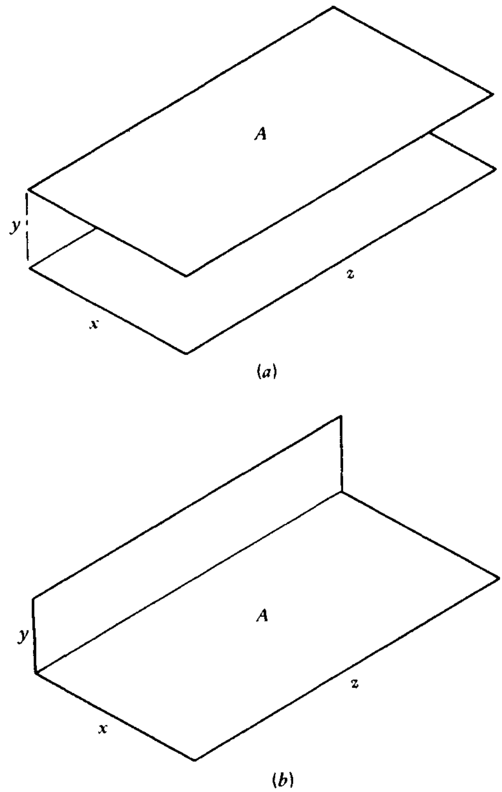


Fig. 45.14 Diagram of radiation view factors for parallel and perpendicular planes.[1]

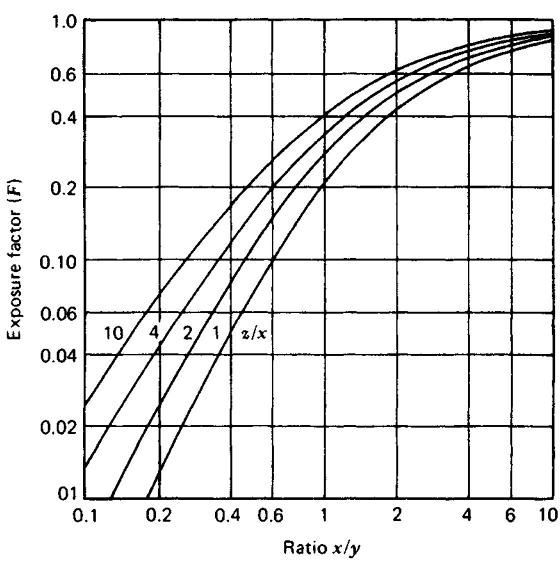


Fig. 45.15 Radiation view factors for parallel planes.[1]

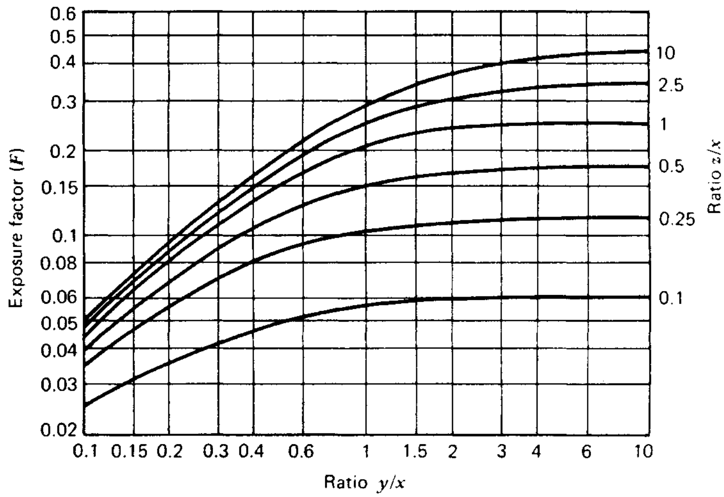


Fig. 45.16 Radiation view factors for perpendicular planes.[1]

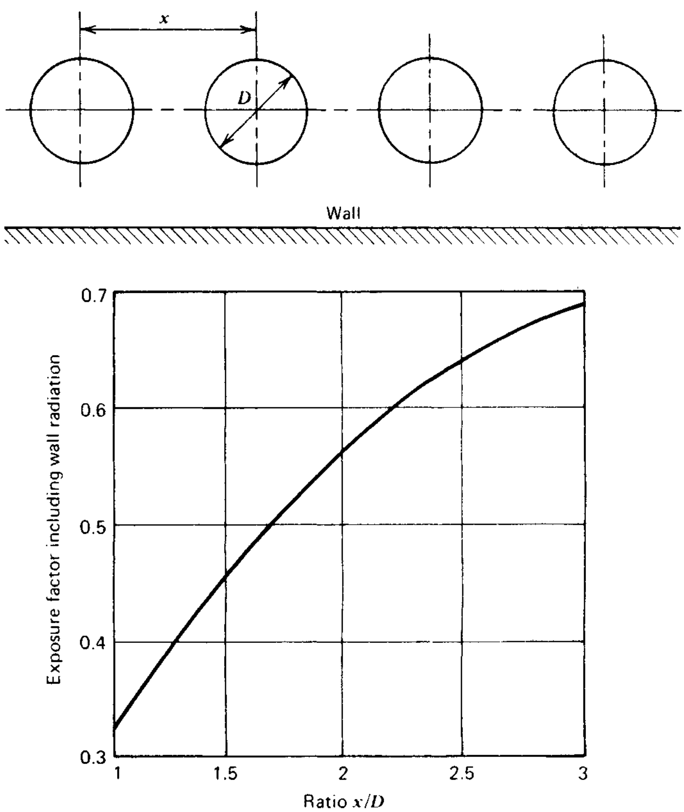


Fig. 45.17 View factors for spaced cylinders with back-up wall.[1]

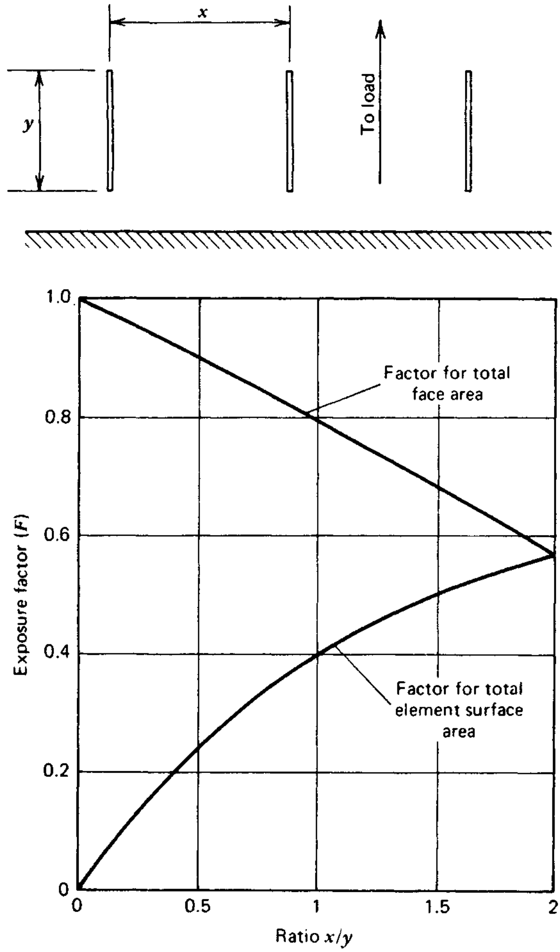


Fig. 45.18 View factors for ribbon-type electric heating elements mounted on back-up wall.[1]

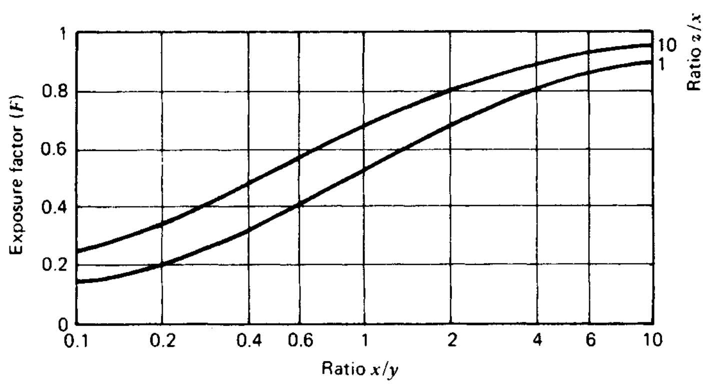


Fig. 45.19 View factors for parallel planes connected by reradiating sidewalls.[1]

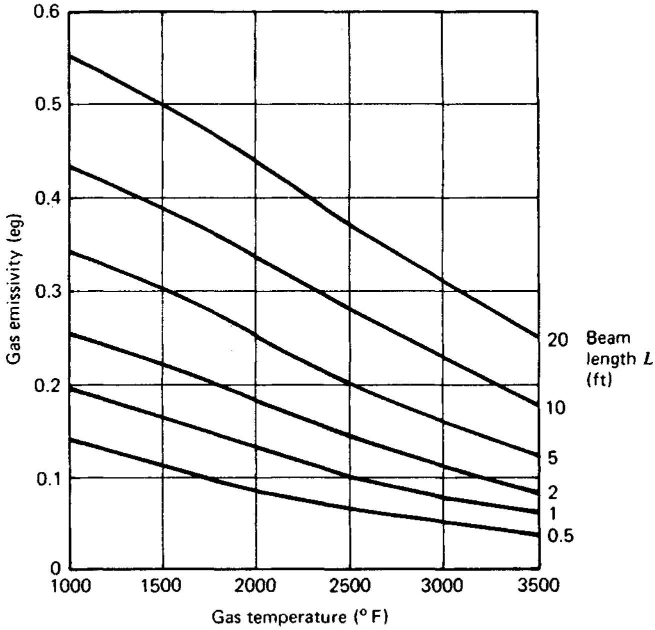


Fig. 45.20 Gas-emissivity for products of combustion of methane burned with 110% air. Approximate for fuel oils and coke oven gas.[1]

coke oven gas and fuel oils. Emissivities for producer gas and blast furnace gas will be lower, because of dilution of radiating gases by nitrogen.

The emissivity of a layer of combustion gases does not increase directly with thickness or density, because of partial absorption during transmission through the depth of the layer. The chart provides several curves for a range of values of L, the effective beam length in feet, at a total pressure of 1 atm. For other pressures, the effective beam length will vary directly with gas density.

Beam lengths for average gas densities will be somewhat less than for very low density because of partial absorption. For some geometrical configurations, average beam lengths are:

Between two large parallel planes, $1.8 \times$ spacing

Inside long cylinder, about $0.85 \times$ diameter in feet

For rectangular combustion chambers, $3.4V/A$ where V is volume in cubic feet and A is total wall area in square feet

Transverse radiation to tube banks, with tubes of D outside diameter spaced at x centers: L/D ranges from 1.48 for staggered tubes at $x/D = 1.5$ to 10.46 for tubes in line and $x/D = 3$ in both directions

45.8.6 Evaluation of Mean Emissivity–Absorptivity

For a gas with emissivity e_g radiating to a solid surface at a temperature of T_s °F, the absorptivity a_g will be less than e_g at T_s because the density of the gas is still determined by T_g. The effective PL becomes $T_s/T_g \times PL$ at T_s. Accurate calculation of the combined absorptivity for carbon dioxide and water vapor requires a determination of a_g for either gas and a correction factor for the total. For the range of temperatures and PL factors encountered in industrial heat transfer, the net heat transfer can be approximated by using a factor e_{gm} somewhat less than e_g at T_g in the formula:

$$Q/A = 0.1713 e_{gm} F(T_g^4 - T_s^4)$$

where T_g is an average of gas temperatures in various parts of the combustion chamber; the effective emissivity will be about $e_{gm} = 0.9e_g$ at T_g and can be used with the chart in Fig. 45.20 to approximate net values.

45.8.7 Combined Radiation Factors

For a complete calculation of heat transfer from combustion gases to furnace loads, the following factors will need to be evaluated in terms of the equivalent fraction of black-body radiation per unit area of the exposed receiving surface:

F_{gs} = Coefficient for gas direct to load, plus radiation reflected from walls to load.

F_{gw} = Coefficient for gas radiation absorbed by walls.

F_{ws} = Coefficient for solid-state radiation from walls to load.

Convection heat transfer from gases to walls and load is also involved, but can be eliminated from calculations by assuming that gas to wall convection is balanced by wall losses, and that gas to load convection is equivalent to a slight increase in load surface absorptivity. Mean effective gas temperature is usually difficult to measure, but can be calculated if other factors are known. For example, carbon steel slabs are being heated to rolling temperature in a fuel-fired continuous furnace. At any point in the furnace, neglecting convection,

$$F_{gw}\,(T_g^4 - T_w^4) = F_{ws}\,(T_w^4 - T_s^4)$$

where T_g, T_w, and T_s are gas, wall, and load surface temperatures in °S.

For a ratio of 2.5 for exposed wall and load surfaces, and a value of 0.17 for gas-to-wall emissivity, $F_{gw} = 2.5 \times 0.17 = 0.425$. With wall to load emissitivity equal to $F_{ws} = 0.89$, wall temperature constant at 2350°F (28.1°S), and load temperature increasing from 70 to 2300°F at the heated surface (T_s = 5.3–27.6°S), the mean value of gas temperature (T_g) can be determined:

$$\text{MTD, walls to load} = \frac{2280 - 50}{\ln(2280/50)} = 584°\text{F}$$

Mean load surface temperature $T_{sm} = 2350 - 584 = 1766°\text{F}$ (22.26°S)

Q/A per unit of load surface, for reradiation:

$$0.425 \times 0.1713(T_g^4 - 28.1^2) = 0.89 \times 0.1713(28.1^4 - 22.26^4) = 57{,}622 \text{ Btu/hr} \cdot \text{ft}^2$$

$$T_g = 34.49°\text{S } (2989°\text{F})$$

With a net wall emissivity of 0.85, 15% of gas radiation will be reflected to the load, with the balance being absorbed and reradiated. Direct radiation from gas to load is then

$$1.15 \times 0.17 \times 0.1713(34.49^4 - 22.26^4) = 47{,}389 \text{ Btu/hr} \cdot \text{ft}^2$$

$$\text{Total radiation: } 57{,}622 + 47{,}389 = 105{,}011 \text{ Btu/hr} \cdot \text{ft}^2$$

For comparison, black-body radiation from walls to load, without gas radiation, would be 64,743 Btu/hr · ft^2 or 62% of the combined total.

With practical furnace temperature profiles, in a counterflow, direct-fired continuous furnace, gas and wall temperatures will be depressed at the load entry end to reduce flue gas temperature and stack loss. The resulting net heating rates will be considered in Section 45.8.12.

Overall heat-transfer coefficients have been calculated for constant wall temperature, in the upper chart in Fig. 45.21, or for constant gas temperature in the lower chart. Coefficients vary with mean gas emissivity and with A_w/A_s, the ratio of exposed surface for walls and load, and are always less than one for overall radiation from gas to load, or greater than one for wall to load radiation. Curves can be used to find gas, wall, or mean load temperatures when the other two are known.

45.8.8 Steady-State Conduction

Heat transfer through opaque solids and motionless layers of liquids or gases is by conduction. For constant temperature conditions, heat flow is by "steady-state" conduction and does not vary with time. For objects being heated or cooled, with a continuous change in internal temperature gradients, conduction is termed "non-steady-state."

Thermal conduction in some solid materials is a combination of heat flow through the material, radiation across internal space resulting from porosity, and convection within individual pores or through the thickness of porous layers.

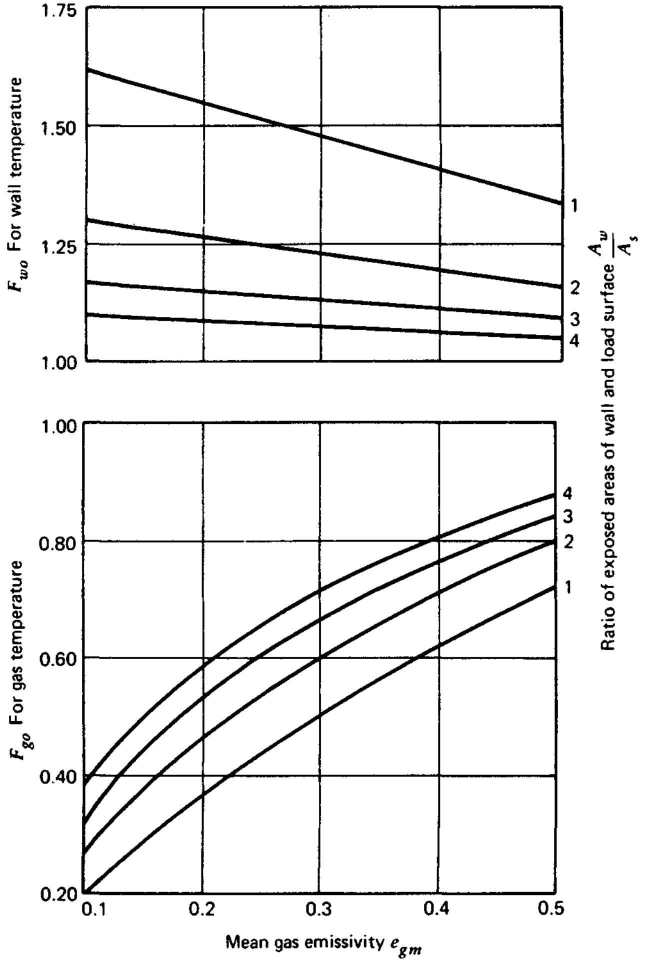


Fig. 45.21 Overall heat-transfer coefficients for gas and solid radiation, as function of gas emissivity and wall-to-load area ratio, for uniform gas or wall temperature, compared to black-body radiation.[1]

Conductivities of refractory and insulating materials tend to increase with temperature, because of porosity effects. Values for most metals decrease with temperature, partly because of reduced density. Conductivity coefficients for some materials used in furnace construction or heated in furnaces are listed in Table 45.5.

A familiar problem in steady-state conduction is the calculation of heat losses through furnace walls made up of multiple layers of materials of different thermal conductivities. A convenient method of finding overall conductance is to find the thermal resistance (r/k = thickness/conductivity in consistent units) and add the total for all layers. Because conductivities vary with temperature, mean temperatures for each layer can be estimated from a preliminary temperature profile for the composite wall. Overall resistance will include the effects of radiation and conduction between the outer wall surface and its surroundings.

A chart showing heat loss from walls to ambient surrounding at 70°F, combining radiation and convection for vertical walls, is shown in Fig. 45.7. The corresponding thermal resistance is included in the overall heat-transfer coefficient shown in Fig. 45.8 as a function of net thermal resistance of the wall structure and inside face temperature.

As an example of application, assume a furnace wall constructed as follows:

Material	r	k	r/k
9 in. firebrick	0.75	0.83	0.90
4½ in. 2000°F insulation	0.375	0.13	2.88
2½ in. ceramic fiber block	0.208	0.067	3.10
Total R for solid wall			6.88

With an inside surface temperature of 2000°F, the heat loss from Fig. 45.7 is about 265 Btu/ft · hr². The corresponding surface temperature from Fig. 45.8 is about 200°F, assuming an ambient temperature of 70°F.

Although not a factor affecting wall heat transfer, the possibility of vapor condensation in the wall structure must be considered by the furnace designer, particularly if the furnace is fired with a sulfur-bearing fuel. As the sulfur dioxide content of fuel gases is increased, condensation temperatures increase to what may exceed the temperature of the steel furnace casing in normal operation. Resulting condensation at the outer wall can result in rapid corrosion of the steel structure.

Condensation problems can be avoided by providing a continuous membrane of aluminum or stainless steel between layers of the wall structure, at a point where operating temperatures will always exceed condensation temperatures.

45.8.9 Non-Steady-State Conduction

Heat transfer in furnace loads during heating or cooling is by transient or non-steady-state conduction, with temperature profiles within loads varying with time. With loads of low internal thermal resistance, heating time can be calculated for the desired load surface temperature and a selected time–temperature profile for furnace temperature. With loads of appreciable thermal resistance from surface to center, or from hot to colder sides, heating time will usually be determined by a specified final load temperature differential, and a selected furnace temperature profile for the heating cycle.

For the case of a slab-type load being heated on a furnace hearth, with only one side exposed, and with the load entering the furnace at ambient temperature, the initial gradient from the heated to the unheated surface will be zero. The heated surface will heat more rapidly until the opposite surface starts to heat, after which the temperature differential between surfaces will taper off with time until the desired final differential is achieved.

In Fig. 45.22 the temperatures of heated and unheated surface or core temperature are shown as a function of time. In the lower chart temperatures are plotted directly as a function of time. In the upper chart the logarithm of the temperature ratio (Y = load temperature/source temperature) is plotted as a function of time for a constant source temperature. After a short initial heating time, during which the unheated surface or core temperature reaches its maximum rate of increase, the two curves in the upper diagram become parallel straight lines.

Factors considered in non-steady-state conduction and their identifying symbols are listed in Table 45.6.

Charts have been prepared by Gurney-Lurie, Heisler, Hottel, and others showing values for Y_s and Y_c for various R factors as a function of X. Separate charts are provided for Y_s and Y_c, with a series of curves representing a series of values of R. These curves are straight lines for most of their length, curving to intersect at $Y = 1$ and $X = 0$. If straight lines are extended to $Y = 1$, the curves for Y_c at all values of R converge at a point near $X = 0.1$ on the line for $Y_c = 1$. It is accordingly possible to prepare a single line chart for $-\ln Y_c/(X - 0.1)$ to fit selected geometrical shapes. This has been done in Fig. 45.23 for slabs, long cylinders, and spheres. Values of Y_c determined with this chart correspond closely with those from conventional charts for $X - 0.1$ greater than 0.2.

Because the ratio Y_s/Y_c remains constant as a function of R after initial heating, it can be shown in chart form, as in Fig. 45.24, to allow Y_s to be determined after Y_c has been found.

By way of illustration, a carbon steel slab 8 in. thick is being heated from cold to T_s = 2350°F in a furnace with a constant wall temperature of 2400°F, with a view factor of 1 and a mean emissivity–absorptivity factor of 0.80. The desired final temperature of the unheated surface is 2300°F, making the Y_c factor

$$Y_c = \frac{2400 - 2300}{2400 - 70} = 0.0429$$

From Fig. 45.23 $H_r = 114 \times 0.80 = 91$; $r = 8/12 = 0.67$; R is assumed at 17. The required heating time is determined from Fig. 45.24:

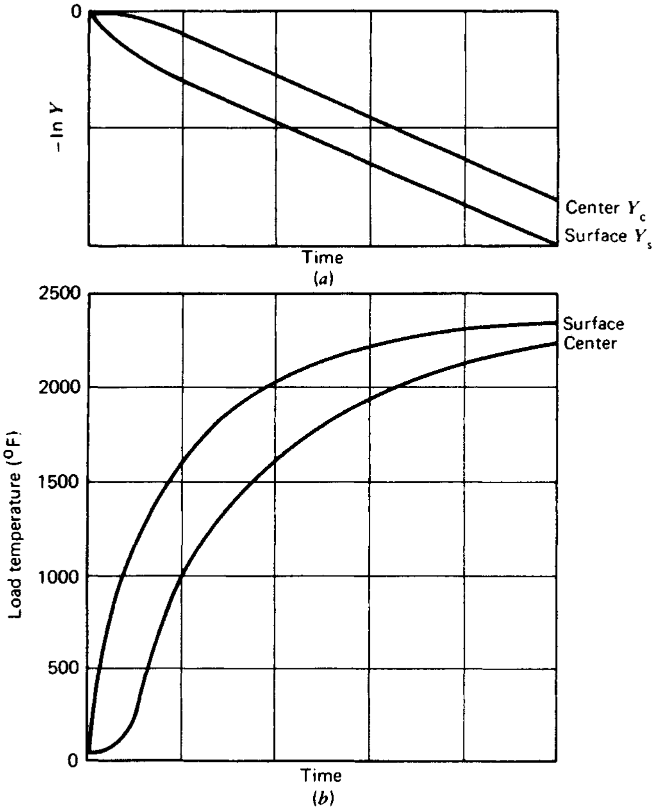


Fig. 45.22 Maximum and minimum load temperatures, and $-\ln Y_s$ or $-\ln Y_c$ as a function of heating time with constant source temperature.[1]

Table 45.6 Non-Steady-State Conduction Factors and Symbols

T_f = Furnace temperature, gas or wall as defined
T_s = Load surface temperature
T_c = Temperature at core or unheated side of load
T_0 = Initial load temperature with all temperatures in units of (°F − 460)/100 or °S

$$Y_s = \frac{T_f - T_s}{T_f - T_0}$$

$$Y_c = \frac{T_f - T_c}{T_f - T_0}$$

R = External/internal thermal resistance ratio = k/rH
X = Time factor = tD/r^2
D = Diffusivity as defined in Section 45.7
r = Depth of heat penetration in feet
k = Thermal conductivity of load (Btu/ft · hr · °F)
H = External heat transfer coefficient (Btu/ft^2 · hr · °F)

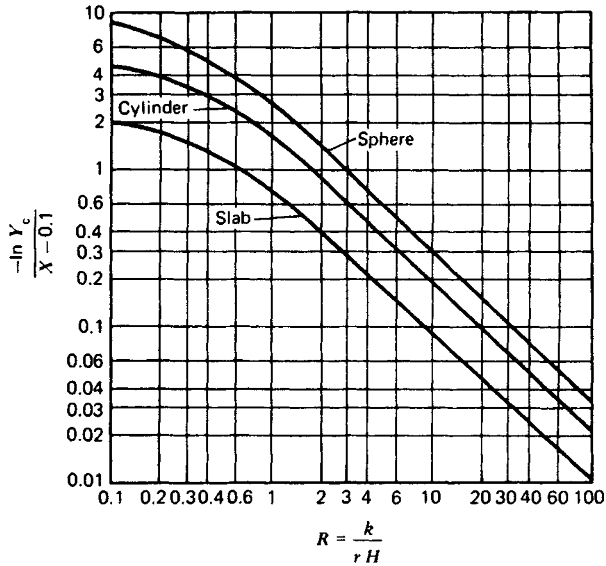


Fig. 45.23 A plot of $-\ln Y_c/(X - 0.1)$ as a function of R.[1]

$$R = \frac{17}{0.67 \times 91} = 0.279$$

$$\frac{-\ln Y_c}{X - 0.1} = 1.7$$

and

$$X = \frac{-\ln 0.0429}{1.7} + 0.1 = 1.95 = tD/r^2$$

With $D = 0.25$, from Section 45.7,

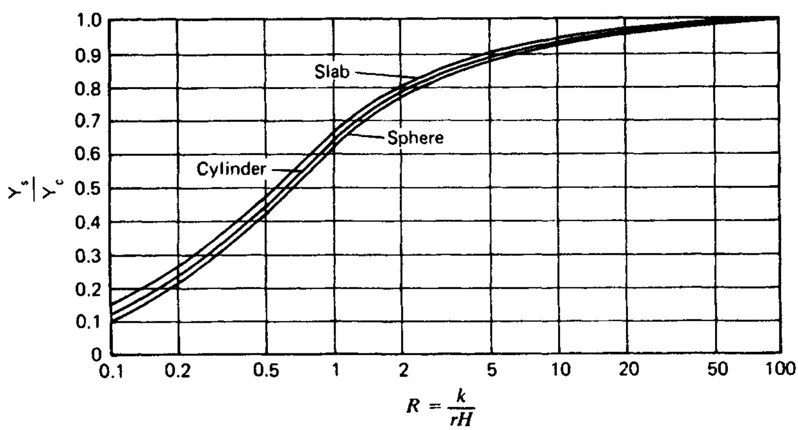


Fig. 45.24 The ratio Y_s/Y_c plotted as a function of R.[1]

$$t = \frac{Xr^2}{D} = \frac{1.95 \times 0.67^2}{0.25} = 3.50 \text{ hr}$$

Slabs or plates heated from two sides are usually supported in the furnace in a horizontal position on spaced conveyor rolls or rails. Support members may be uncooled, in which case radiation to the bottom surface will be reduced by the net view factor. If supports are water cooled, the additional heat input needed to balance heat loss from load to supports can be balanced by a higher furnace temperature on the bottom side. In either case, heating times will be greater than for a uniform input from both sides.

Furnace temperatures are normally limited to a fraction above final load temperatures, to avoid local overheating during operating delays. Without losses to water cooling, top and bottom furnace temperature will accordingly be about equal.

45.8.10 Heat Transfer with Negligible Load Thermal Resistance

When heating thin plates or small-diameter rods, with internal thermal resistance low enough to allow heating rates unlimited by specified final temperature differential, the non-steady-state-conduction limits on heating rates can be neglected. Heating time then becomes

$$t = \frac{W \times C \times (T_s - T_0)}{A \times H \times \text{MTD}}$$

The heat-transfer coefficient for radiation heating can be approximated from the chart in Fig. 45.13 or calculated as follows:

$$H_r = \frac{0.1713 e_m F_s [T_f^4 - (T_f - \text{MTD})^4]}{\text{MTD} \times A_s}$$

As an illustration, find the time required to heat a steel plate to 2350°F in a furnace at a uniform temperature of 2400°F. The plate is 0.25 in. thick with a unit weight of 10.2 lb/ft^2 and is to be heated from one side. Overall emissivity–absorptivity is $e_m = 0.80$. Specific heat is 0.165. The view factor is $F_s = 1$. MTD is

$$\frac{(2400 - 70) - (2400 - 2350)}{\ln(2400 - 70)/(2400 - 2350)} = 588°\text{F}$$

$$H_r = \frac{0.1713 \times 0.80 \times 1[28.6^4 - (28.6 - 5.88)^4]}{588} = 93.8$$

$$t = \frac{10.2 \times 0.165(2350 - 70)}{1 \times 93.8 \times 588} = 0.069 \text{ hr}$$

45.8.11 Newman Method

For loads heated from two or more perpendicular sides, final maximum temperatures will be at exposed corners, with minimum temperatures at the center of mass for heating from all sides, or at the center of the face in contact with the hearth for hearth-supported loads heated equally from the remaining sides. For surfaces not fully exposed to radiation, the corrected H factor must be used.

The Newman method can be used to determine final load temperatures with a given heating time t. To find time required to reach specified maximum and minimum final load temperatures, trial calculations with several values of t will be needed.

For a selected heating time t, the factors Y_s and Y_c can be found from charts in Figs. 45.23 and 45.24 for the appropriate values of the other variables—T_s, T_c, H, k, and r—for each of the heat flow paths involved—r_x, r_y, and r_z. If one of these paths is much longer than the others, it can be omitted from calculations:

$$Y_c = Y_{cx} \times Y_{cy} \times Y_{cz}$$

$$Y_s = Y_{sx} \times Y_{sy} \times Y_{sz}$$

For two opposite sides with equal exposure only one is considered. With T_c known, T_s and T_f (furnace temperature, T_g or T_w) can be calculated.

As an example, consider a carbon steel ingot, with dimensions 2 ft × 4 ft × 6 ft, being heated in a direct-fired furnace. The load is supported with one 2 ft × 4 ft face in contact with the refractory hearth and other faces fully exposed to gas and wall radiation. Maximum final temperature will be at an upper corner, with minimum temperature at the center of the 2 ft × 4 ft bottom surface.

Assuming that the load is a somewhat brittle steel alloy, the initial heating rate should be suppressed and heating with a constant gas temperature will be assumed. Heat-transfer factors are then

Flow paths r_s = 1 ft and r_y = 2 ft, the contribution of vertical heat flow, on axis r_z, will be small enough to be neglected.

Desired final temperatures: T_c = 2250°F and T_s (to be found) about 2300°F, with trial factor t = 9 hr.

H from gas to load = 50

k mean value for load = 20 and D = 0.25

Radial heat flow path	rx	ry
r	1	2
$X = tD/r^2$	2.25	0.5625
$R = k/H_r$	0.4	0.2
$-\ln Y_c/(X - 0.1)$ from Fig. 45.23	1.3	1.7
Y_s/Y_c from Fig. 45.24	0.41	0.26
Y_c	0.0611	0.455
Y_s	0.025	0.119

Combined factors:

$$Y_c = 0.0611 \times 0.455 = 0.0278 = \frac{T_g - T_c}{T_g - 70}$$

$$Y_s = 0.025 \times 0.119 = 0.003 = \frac{T_g - T_s}{T_g - 70}$$

For T_c = 2250°F, T_g = 2316°F
T_s = 2309°F

This is close enough to the desired T_s = 2300°F.

The time required to heat steel slabs to rolling temperature, as a function of the thickness heated from one side and the final load temperature differential, is shown in Fig. 45.25. Relative heating times for various hearth loading arrangements, for square billets, are shown in Fig. 45.26. These have

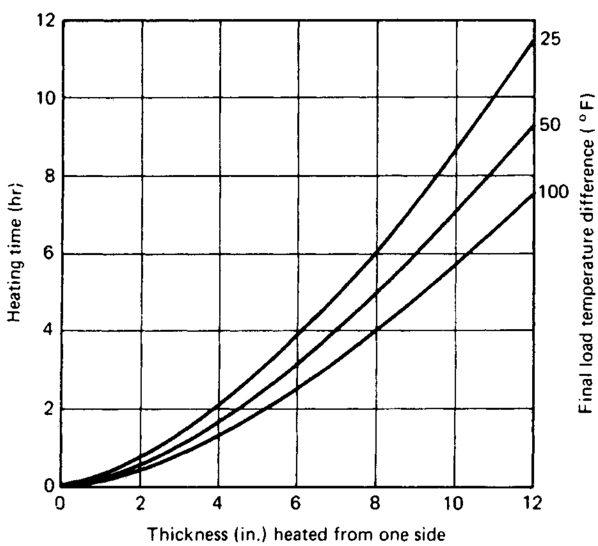


Fig. 45.25 Relative heating time for square billets as a function of loading pattern.[1]

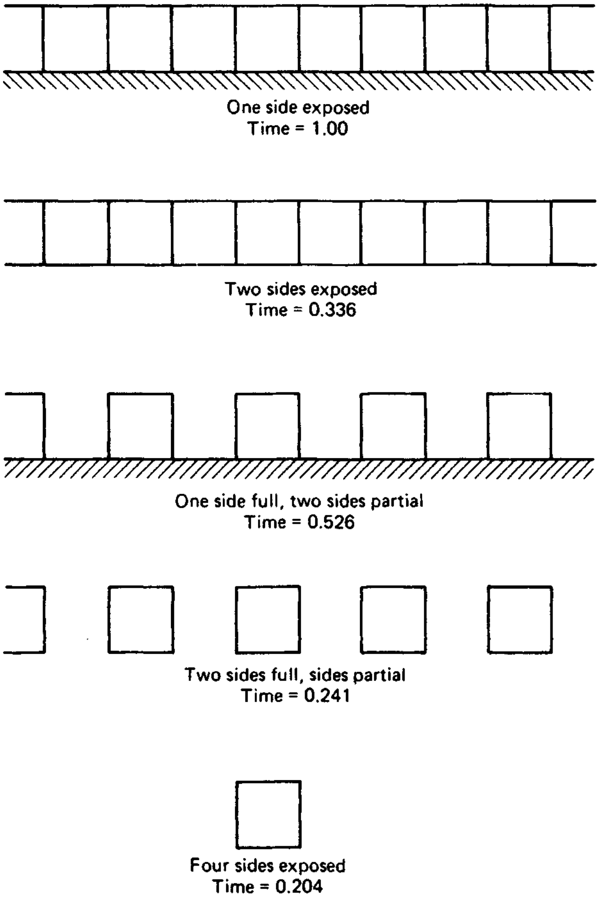


Fig. 45.26 Heating time for carbon steel slabs to final surface temperature of 2300°F, as a function of thickness and final load temperature differential.[1]

been calculated by the Newman method, which can also be used to evaluate other loading patterns and cross sections.

45.8.12 Furnace Temperature Profiles

To predict heating rates and final load temperatures in either batch or continuous furnaces, it is convenient to assume that source temperatures, gas (T_g) or furnace wall (T_w), will be constant in time. Neither condition is achieved with contemporary furnace and control system designs. With constant gas temperature, effective heating rates are unnecessarily limited, and the furnace temperature control system is dependent on measurement and control of gas temperatures, a difficult requirement. With uniform wall temperatures, the discharge temperature of flue gases at the beginning of the heating cycle will be higher than desirable. Three types of furnace temperature profiles, constant T_g, constant T_w, and an arbitrary pattern with both variables, are shown in Fig. 45.27.

Contemporary designs of continuous furnaces provide for furnace temperature profiles of the third type illustrated, to secure improved capacity without sacrificing fuel efficiency. The firing system comprises three zones of length: a preheat zone that can be operated to maintain minimum flue gas temperatures in a counterflow firing arrangement, a firing zone with a maximum temperature and firing rate consistent with furnace maintenance requirements and limits imposed by the need to avoid overheating of the load during operating delays, and a final or soak zone to balance furnace temperature with maximum and minimum load temperature specifications. In some designs, the preheat zone is unheated except by flue gases from the firing zone, with the resulting loss of furnace capacity offset by operating the firing zone at the maximum practical limit.

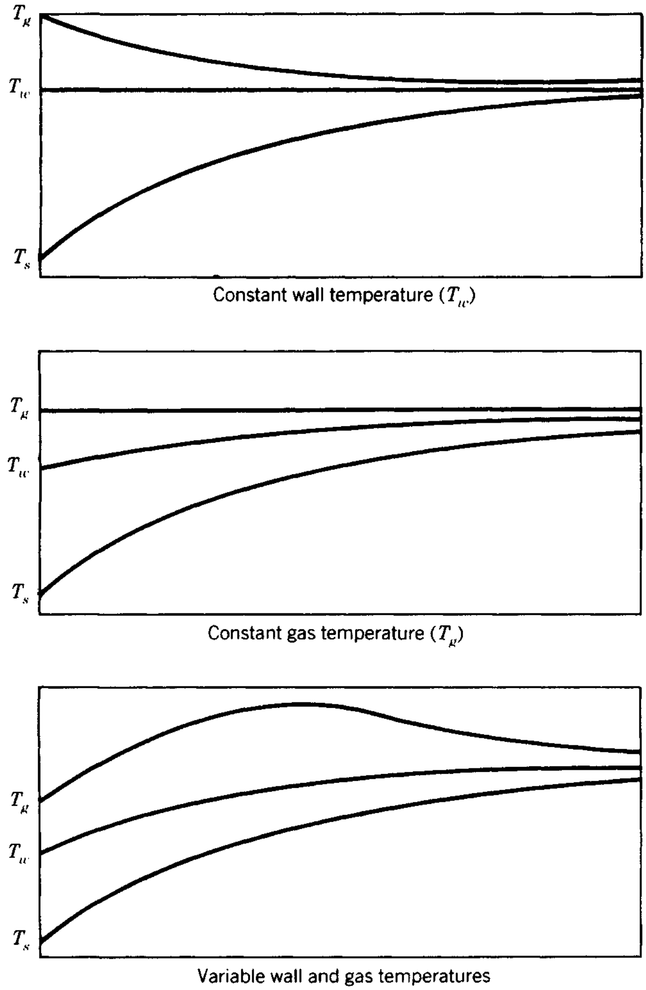


Fig. 45.27 Furnace temperature profiles.

45.8.13 Equivalent Furnace Temperature Profiles

Furnace heating capacities are readily calculated on the assumption that furnace temperature, either combustion gases or radiating walls, is constant as a function of position or time. Neither condition is realized in practice; and to secure improved capacity with reduced fuel demand in a continuous furnace, contemporary designs are based on operation with a variable temperature profile from end to end, with furnace wall temperature reduced at the load charge and flue gas discharge end, to improve available heat of fuel, and at the load discharge end, to balance the desired maximum and minimum load temperatures. Any loss in capacity can be recovered by operating the intermediate firing zones at a somewhat elevated temperature.

Consider a furnace designed to heat carbon steel slabs, 6 in. thick, from the top only to final temperatures of 2300°F at top and 2250°F at the bottom. To hold exit flue gas temperature to about 2000°F, wall temperature at the charge end will be about 1400°F. The furnace will be fired in four zones of length, each 25 ft long for an effective total length of 100 ft. The preheat zone will be unfired, with a wall temperature tapering up to 2400°F at the load discharge end. That temperature will be held through the next two firing zones and dropped to 2333°F to balance final load temperatures in the fourth or soak zone. With overall heating capacity equal to the integral of units of length times their absolute temperatures, effective heat input will be about 87% of that for a uniform temperature of 2400°F for the entire length.

Heat transfer from combustion gases to load will be by direct radiation from gas to load, including reflection of incident radiation from walls, and by radiation from gas to walls, absorbed and reradiated from walls to load. Assuming that wall losses will be balanced by convection heat transfer from gases, gas radiation to walls will equal solid-state radiation from walls to load:

$$A_w/A_s \times 0.1713 \times e_{gm}(T_g^4 - T_w^4) = e_{ws} \times 0.1713(T_w^4 - T_s^4)$$

where A_w/A_s = exposed area ratio for walls and load
e_{gm} = emissivity–absorptivity, gas to walls
e_{ws} = emissivity–absorptivity, walls to load

At the midpoint in the heating cycle, MTD = 708°F and mean load surface temperature = T_{sm} = 1698°F.

With a_s = 0.85 for refractory walls, 15% of gas radiation will be reflected to load, and total gas to load radiation will be:

$$1.15 \times e_{gm} \times 0.1713(T_g^4 - T_s^4)$$

For A_w/A_s = 2.5, e_{gm} = 0.17, and e_{ws} = 0.89 from walls to load, the mean gas temperature = T_g = 3108°F, net radiation, gas to load = 47,042 Btu/hr · ft² and gas to walls = walls to load = 69,305 Btu/hr · ft² for a total of 116,347 Btu/hr · ft². This illustrates the relation shown in Fig. 45.21, since black-body radiation from walls to load, without gas radiation, would be 77,871 Btu/hr · ft². Assuming black-body radiation with a uniform wall temperature from end to end, compared to combined radiation with the assumed wall temperature, overall heat transfer ratio will be

$$(0.87 \times 116{,}347)/77{,}871 = 1.30$$

As shown in Fig. 45.26, this ratio will vary with gas emissivity and wall to load areas exposed. For the range of possible values for these factors, and for preliminary estimates of heating times, the chart in Fig. 45.26 can be used to indicate a conservative heating time as a function of final load temperature differential and depth of heat penetration, for a furnace temperature profile depressed at either end.

Radiation factors will determine the mean coefficient of wall to load radiation, and the corresponding non-steady-state conduction values. For black-body radiation alone, H_r is about 77,871/708 = 110. For combined gas and solid-state radiation, in the above example, it becomes 0.87 × 116,347/708 = 143. Values of R for use with Figs. 45.23 and 45.24, will vary correspondingly ($R = k/4H$).

45.8.14 Convection Heat Transfer

Heat transferred between a moving layer of gas and a solid surface is identified by "convection." Natural convection occurs when movement of the gas layer results from differentials in gas density of the boundary layer resulting from temperature differences and will vary with the position of the boundary surface: horizontal upward, horizontal downward, or vertical. A commonly used formula is:

$$H_c = 0.27(T_g - T_s)^{0.25}$$

where H_c = Btu/hr · ft² · °F
$T_g - T_s$ = temperature difference between gas and surface, in °F

Natural convection is a significant factor in estimating heat loss from the outer surface of furnace walls or from uninsulated pipe surfaces.

"Forced convection" is heat transfer between gas and a solid surface, with gas velocity resulting from energy input from some external source, such as a recirculating fan.

Natural convection can be increased by ambient conditions such as building drafts and gas density. Forced convection coefficients will depend on surface geometry, thermal properties of the gas, and Reynolds number for gas flow. For flow inside tubes, the following formula is useful:

$$H_c = 0.023 \frac{k}{D} \mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \mathrm{Btu/hr \cdot ft^2 \cdot {}^\circ F}$$

where k = thermal conductivity of gas
D = inside diameter of tube in ft
Re = Reynolds number
Pr = Prandtl number

Forced convection coefficients are given in chart form in Fig. 45.28 for a Prandtl number assumed at 0.70.

For forced convection over plane surfaces, it can be assumed that the preceding formula will apply for a rectangular duct of infinitely large cross section, but only for a length sufficient to establish uniform velocity over the cross section and a velocity high enough to reach the Re value needed to promote turbulent flow.

In most industrial applications, the rate of heat transfer by forced convection as a function of power demand will be better for perpendicular jet impingement from spaced nozzles than for parallel flow. For a range of dimensions common in furnace design, the heat-transfer coefficient for jet impingement of air or flue gas is shown in Fig. 45.29, calculated for impingement from slots 0.375 in. wide spaced at 18–24 in. centers and with a gap of 8 in. from nozzle to load.

Forced convection factors for gas flow through banks of circular tubes are shown in the chart in Fig. 45.30 and for tubes spaced as follows:

A: staggered tubes with lateral spacing equal to diagonal spacing.
B: tubes in line, with equal spacing across and parallel to direction of flow.
C: tubes in line with lateral spacing less than half longitudinal spacing.
D: tubes in line with lateral spacing over twice longitudinal spacing.

With F the configuration factor from Fig. 45.30, heat-transfer coefficients are

$$H_c = Fk\mathrm{Re}^{0.6}/D$$

Convection coefficients from this formula are approximately valid for 10 rows of tubes or more, but are progressively reduced to a factor of 0.65 for a single row.

For gas to gas convection in a cross-flow tubular heat exchanger, overall resistance will be the sum of factors for gas to the outer diameter of tubes, tube wall conduction, and inside diameter of

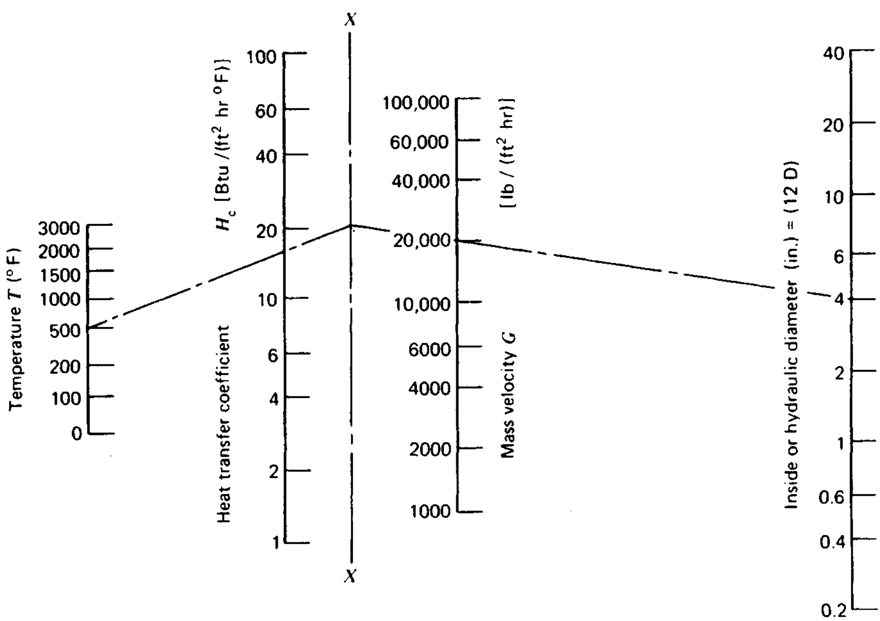


Fig. 45.28 Convection coefficient (H_c) for forced convection inside tubes to air or flue gas.[1]

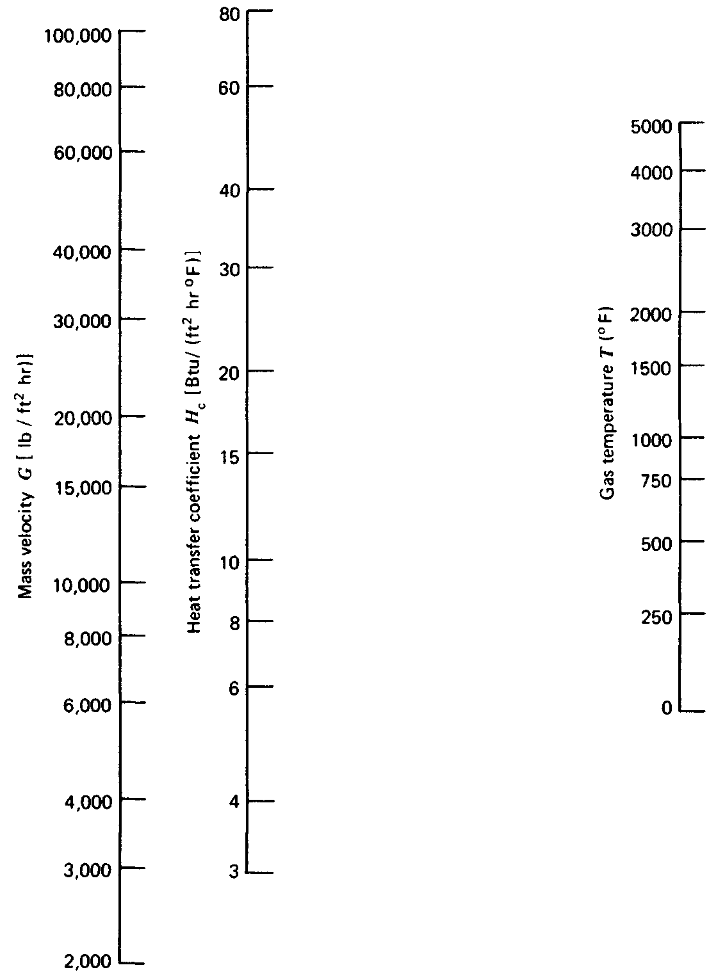


Fig. 45.29 Convection coefficient (H_c) for jet impingement of air or flue gas on plane surfaces, for spaced slots, 0.375 in. wide at 18–24 in. centers, 8 in. from load.[1]

tubes to gas. Factors for the outer diameter of tubes may include gas radiation as calculated in Section 45.7.5.

45.8.15 Fluidized-Bed Heat Transfer

For gas flowing upward through a particular bed, there is a critical velocity when pressure drop equals the weight of bed material per unit area. Above that velocity, bed material will be suspended in the gas stream in a turbulent flow condition. With the total surface area of suspended particles on the order of a million times the inside surface area of the container, convection heat transfer from gas to bed material is correspondingly large. Heat transfer from suspended particles to load is by conduction during repeated impact. The combination can provide overall coefficients upward of 10 times those available with open convection, permitting the heating of thick and thin load sections to nearly uniform temperatures by allowing a low gas to load thermal head.

45.8.16 Combined Heat-Transfer Coefficients

Many furnace heat-transfer problems will combine two or more methods of heat transfer, with thermal resistances in series or in parallel. In a combustion chamber, the resistance to radiation from gas to load will be parallel to the resistance from gas to walls to load, which is two resistances in series.

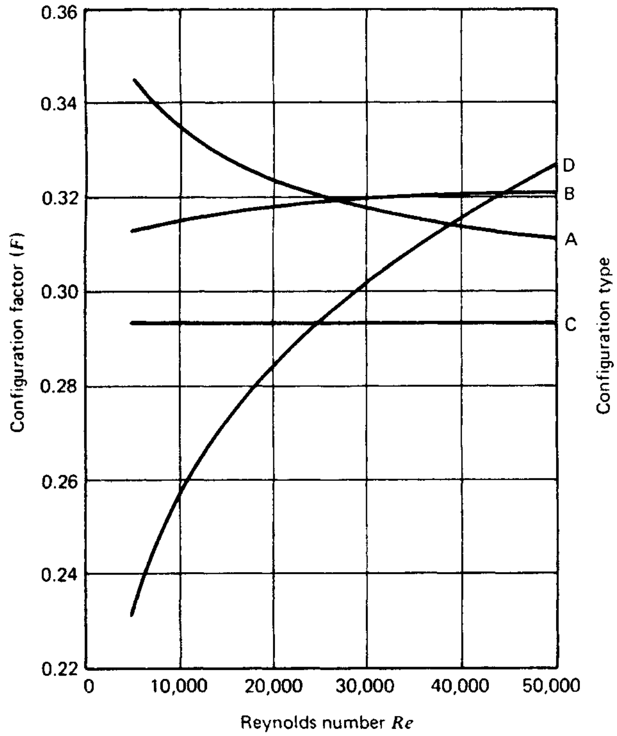


Fig. 45.30 Configuration factors for convection heat transfer, air or flue gas through tube banks.[1]

Heat flow through furnace walls combines a series of resistances in series, combustion gases to inside wall surface, consecutive layers of the wall structure, and outside wall surface to surroundings, the last a combination of radiation and convection in parallel.

As an example, consider an insulated, water-cooled tube inside a furnace enclosure. With a tube outside diameter of 0.5 ft and a cylindrical insulation enclosure with an outside diameter of 0.75 ft, the net thickness will be 0.125 ft. The mean area at midthickness is $\pi(0.5 + 0.75)/2$, or 1.964 ft^2 per ft of length. Outer surface area of insulation is 0.75π, or 2.36 ft^2 per linear foot. Conductivity of insulation is $k = 0.20$. The effective radiation factor from gas to surface is assumed at 0.5 including reradiation from walls. For the two resistances in series,

$$0.1713 \times 0.5 \times 2.36(29.6^4 - T_s^4) = 1.964(T_s - 150) \times \frac{0.20}{0.125}$$

By trial, the receiver surface temperature is found to be about 2465°F. Heat transfer is about 7250 Btu/hr · linear ft or 9063 Btu/hr · ft^2 water-cooled tube surface.

If the insulated tube in the preceding example is heated primarily by convection, a similar treatment can be used to find receiver surface temperature and overall heat transfer.

For radiation through furnace wall openings, heat transfer in Btu/hr · ft^2 · °F is reduced by wall thickness, and the result can be calculated similarly to the problem of two parallel planes of equal size connected by reradiating walls, as shown in Fig. 45.19.

Heat transfer in internally fired combustion tubes ("radiant tubes") is a combination of convection and gas radiation from combustion gases to tube wall. External heat transfer from tubes to load will be direct radiation and reradiation from furnace walls, as illustrated in Fig. 45.19. The overall factor for internal heat transfer can be estimated from Fig. 45.31, calculated for 6 in. and 8 in. inside diameter tubes. The convection coefficient increases with firing rate and to some extent with tem-

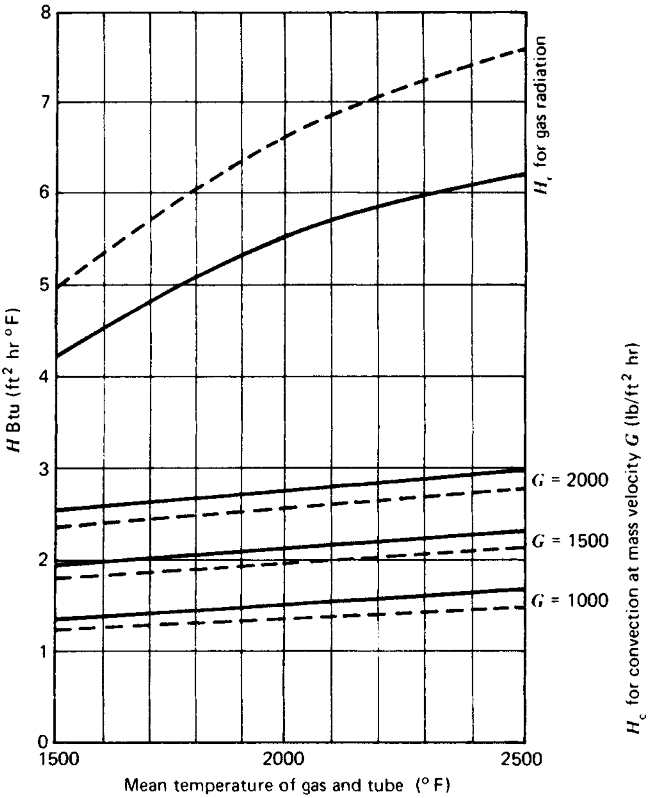


Fig. 45.31 Gas radiation (H_r) and convection (H_c) coefficients for flue gas inside radiant tubes.[1]

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: