
2 Heat Transfer

Charles E. Baukal, Jr., Ph.D., P.E.

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2.1 INTRODUCTION

Heat transfer plays a critical role in industrial combustion processes where the primary objective is to transfer heat from the hot combustion products to some type of load. In industrial processes, radiation is often the dominant mechanism and forced convection also plays an important function. The more efficient the heat transfer processes, the higher the overall system efficiency. This is important for many reasons. The operating costs in the form of fuel are inversely proportional to the efficiency — the higher the efficiency, the lower the fuel consumption. Pollution emissions are indirectly proportional to fuel efficiency. The higher the efficiency, the lower the fuel consumption and therefore the lower the emissions per unit of product, because less fuel is combusted and therefore less emissions are produced. More efficient processes usually mean that either the combustors can be smaller for a given production rate or, alternatively, more can be produced in a given size combustor. This means that capital costs may be reduced per unit of product. That also translates indirectly into more emissions reductions because less energy is needed to make the combustor and therefore less pollutants are generated. A smaller combustor per unit of product means that less space is required in the plant. Higher thermal efficiencies usually also mean that less energy is lost through the walls of the combustor, which benefits the workers in the vicinity who will be less likely to be overheated and burned. Improved thermal efficiencies are therefore dependent on maximizing the heat transfer from the combustion products to the load.

There are some nondimensional numbers that are commonly used in heat transfer analysis. The Reynolds number is the ratio of the inertial forces to the viscous forces in a flow:

$$\text{Re} = \frac{\rho v l}{\mu} = \frac{v l}{\nu} \quad (2.1)$$

where ρ is the fluid density, v is the fluid velocity, l is the characteristic length scale (e.g., diameter for a pipe for flow through a pipe), μ is the absolute viscosity, and ν is the kinematic viscosity. The Reynolds number is low for laminar flows and high for turbulent flows, with transition flow at values in between. The actual range for each type of flow depends on the flow geometry. The Prandtl number is the ratio of momentum diffusivity to thermal diffusivity and is defined as:

$$\text{Pr} = \frac{c_p \mu}{k} \quad (2.2)$$

where c_p is the fluid constant pressure specific heat and k is the fluid thermal conductivity. For many gases, $\text{Pr} \approx 0.7$. The Nusselt number is the ratio of the convective and conductive heat transfer rates:

$$\text{Nu} = \frac{h l}{k} \quad (2.3)$$

where h is the convective heat transfer coefficient. In forced convection flows, Nu is commonly a function of the Pr and Re , and is used to determine the convection coefficient.

The three primary modes of heat transfer include convection, radiation, and conduction. These are discussed next as they pertain to the heat transfer in industrial combustion processes, particularly from burners. There are other modes of heat transfer such as boiling and condensation. While these are important in some specific processes, they are not included due to their limited applicability. Much of the information in this chapter has been taken from Baukal et al. (2000, 2001).^{1,2}

2.2 RADIATION

Thermal radiation is one of the most important heat transfer mechanisms in industrial furnaces.³ Radiation is a unique method of heat transfer because no medium is required for energy transport — it can be transmitted through a vacuum or through a medium. Radiation is simply the transmission of energy by electromagnetic waves, which are characterized by their wavelength or frequency and are related as follows:

$$\lambda = \frac{c}{\nu} \quad (2.4)$$

where λ is the wavelength, c is the speed of light, and ν is the frequency. In the fields of heat transfer and combustion, wavelength is more commonly used. The classification of the various types of radiation is shown in [Figure 2.1](#). In industrial combustion heating, the most important type of radiation is infrared. The human optic nerves are sensitive to radiation in the wavelengths from 0.38 to 0.76 μm , which means that we can see radiation in that band. In practical terms, humans cannot see thermal radiation from bodies at temperatures below about 900°F (500°C).⁴

There are four possible things that can happen to radiation incident on a medium (solid, liquid, or gas). The radiation can be absorbed, reflected, transmitted, or some combination of these three, which is most often the case, as shown in [Figure 2.2](#). In general,

$$\alpha + \rho + \tau = 1 \quad (2.5)$$

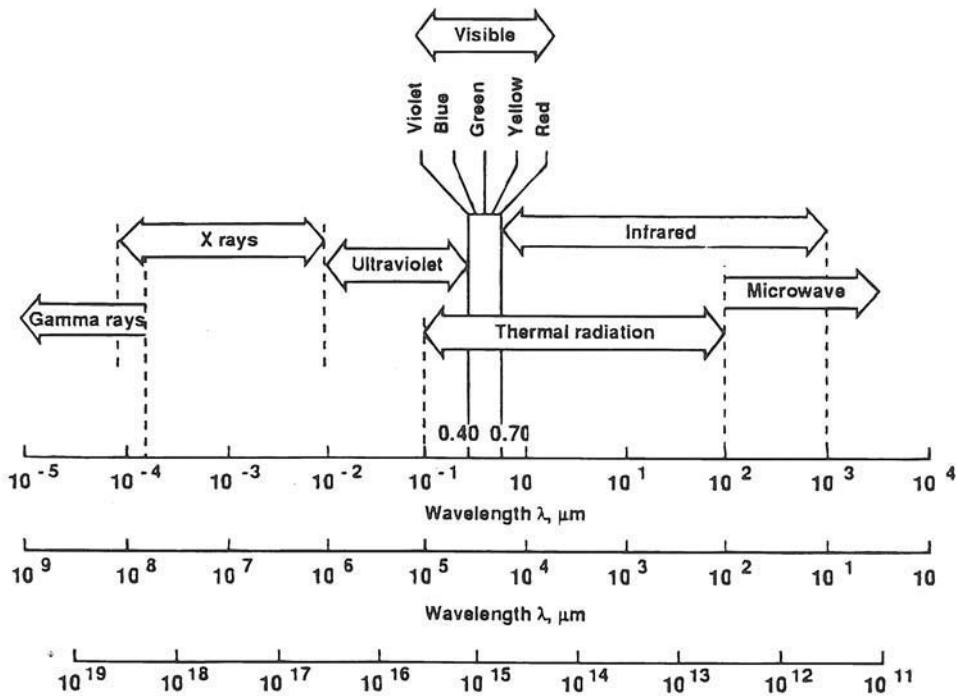


FIGURE 2.1 Electromagnetic spectrum.¹³⁵

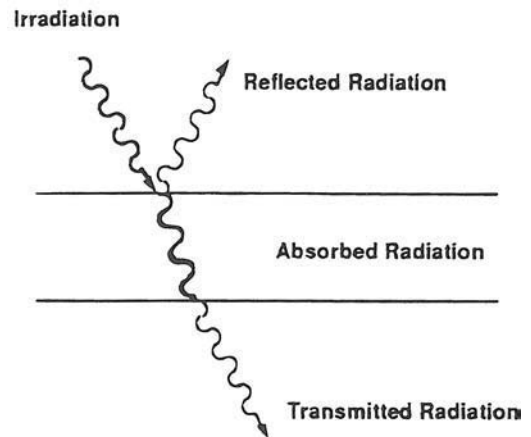


FIGURE 2.2 Radiant energy absorbed, reflected, and transmitted through a material.¹³⁵

where α is the absorptivity of the medium, ρ is the reflectivity, and τ is the transmissivity, which are defined as:

$$\alpha \equiv \frac{\text{absorbed part of incoming radiation}}{\text{total incoming radiation}}$$

$$\rho \equiv \frac{\text{reflected part of incoming radiation}}{\text{total incoming radiation}}$$

$$\tau \equiv \frac{\text{transmitted part of incoming radiation}}{\text{total incoming radiation}}$$

For most solid materials, the transmissivity is low except for things like glass and plastics. The reflectivity of most solids is low, unless they are highly polished (e.g., new stainless steel). For liquids, the transmissivity may be significant, especially for fluids with high water contents. For most gases, the transmissivity is generally very high with negligible absorptance and reflectance. These radiative properties are extremely important in determining how much radiation will be transferred to and from a medium. This is further complicated by the fact that these radiative properties may be functions of wavelength, angle of incidence, surface condition, and thickness. An example of wavelength dependence is solid glass, which transmits shorter wavelength UV radiation but absorbs longer wavelength IR radiation. At high angles of incidence, a surface may be more reflective, while at normal incident angles, a surface may be more absorptive. Highly oxidized metals have high absorptivities and low reflectivities, while highly polished metals have lower absorptivities and high reflectivities. A very thin layer of a solid material can have a significant transmissivity, while a thick layer of the same solid may have no transmissivity. The radiative properties can also change over time, such as when an initially reflective metal surface becomes less reflective as it oxidizes. Therefore, an important challenge in computing radiative heat transfer in combustion systems is determining radiative properties.

There are three common forms of radiation heat transfer in industrial heating applications: (1) radiation from a solid surface, (2) radiation from a gaseous medium (usually referred to as non-luminous radiation), and (3) radiation from particles in a gaseous medium (usually referred to as luminous radiation). The latter two types are of most interest for the heat transfer from burners. Each of these three types is considerably different and must be treated accordingly. Also, it is not uncommon for two or all three types to be important in industrial heating. In industrial furnaces, as much as 90% of the heat transfer to the load can be by radiation.⁵

The “Total Schmidt Method” has been used to determine the amount of radiation coming from the walls and that coming from the flame, which are received by the load.⁶ The flame radiation is measured by sighting the flame with a radiation pyrometer, where the background behind the flame is water-cooled. The radiation from the water-cooled wall is negligible compared to that from the flame and is ignored:

$$q_1 = \sigma A \epsilon_f T_f^4 \quad (2.6)$$

where σ is the Stefan-Boltzmann constant (see Equation 3.28), ϵ_f is the average flame emissivity, A is the area radiating, and T_f is the average absolute flame temperature. Then the radiation from the flame with a hot blackbody background (furnace wall) is measured:

$$q_2 = \sigma A \epsilon_f T_f^4 + \sigma A (1 - \alpha_f) T_w^4 \quad (2.7)$$

where α_f is the average flame absorptivity and T_w is the average wall temperature. Finally, the radiation from the hot blackbody background (furnace wall) is measured immediately after the flame is extinguished:

$$q_3 = \sigma A T_w^4 \quad (2.8)$$

The flame is assumed to be a graybody so that:

$$\alpha_f = \epsilon_f \quad (2.9)$$

Solving the above four equations for the flame emissivity ϵ_f :

$$\epsilon_f = 1 - \left(\frac{q_2 - q_1}{q_3} \right) \quad (2.10)$$

and then solving for flame temperature T_f :

$$T_f = \left(\frac{q_1}{\sigma A \epsilon_f} \right)^{1/4} \quad (2.11)$$

Lowes and Newall (1974) note that there are some errors that arise with this method:⁷

1. The absorptivity of the flame is not equal to the emissivity unless the blackbody source is at the flame temperature.
2. The emissivities obtained assume a graybody, which is normally not strictly true.
3. Soot particles in the flame can cause radiation scattering and may give high absorptivity values.

They showed that the error due to the blackbody not being at the flame temperature can be 42% if the blackbody temperature was 1000K and the flame temperature was 2000K. Leblanc et al. (1974) used a variation of this technique to distinguish between the radiation from the walls and from the flame, to the load.⁸ An ellipsoidal radiometer was used to measure the total radiation received by the load. Then the flame was turned off and the radiation was again measured, which was the contribution by the walls. The radiation from the flame was then the difference between that total and the wall:

$$q_{\text{rad, total}} = K q_{\text{rad, walls}} + q_{\text{rad, flame}} \quad (2.12)$$

where the flame radiation from the walls has been corrected by a nonabsorption factor K that is obtained experimentally. Lihou (1977) reviewed analytical techniques for calculating nonspectral radiation for use in furnace design.⁹ Many general books on radiation heat transfer are available.^{4,10-17}

2.2.1 SURFACE RADIATION

The spectral or monochromatic emissive power of a blackbody is dependent on the wavelength of the radiation and on the absolute temperature of the blackbody, and is calculated as follows:

$$E_{b\lambda} = \frac{C_1}{\lambda^5 (e^{C_2/\lambda T} - 1)} \quad (2.13)$$

where $E_{b\lambda}(T)$ is the monochromatic emissive power of a blackbody in Btu/hr-ft²-μm (W/m³), $C_1 = 1.1870 \times 10^8$ Btu/μm⁴-ft²-hr (3.7415×10^{-16} W-m²), $C_2 = 2.5896 \times 10^4$ μm-°R (1.4388×10^{-2} m-K), T is the absolute temperature of the body in °R (K), and λ is the wavelength in μm (m). Figure 2.3a and b show some graphs of this relation as a function of wavelength and temperature. As can be seen, the peak radiation shifts to shorter wavelengths as the body temperature increases. That wavelength can be computed from Wein's displacement law as follows:

$$\lambda_{\text{max}} = \frac{c}{T} \quad (2.14)$$

where $c = 5215.6$ μm-°R (2897.6 μm-K).

Example 2.1

Given: Blackbody temperature = 2000°F.

Find: Wavelength for peak radiation.

$$\text{Solution: } \lambda_{\text{max}} = \frac{c}{T} = \frac{5215.6 \mu\text{-}^\circ\text{R}}{(2000 + 460)^\circ\text{R}} = 2.12 \mu\text{m}$$

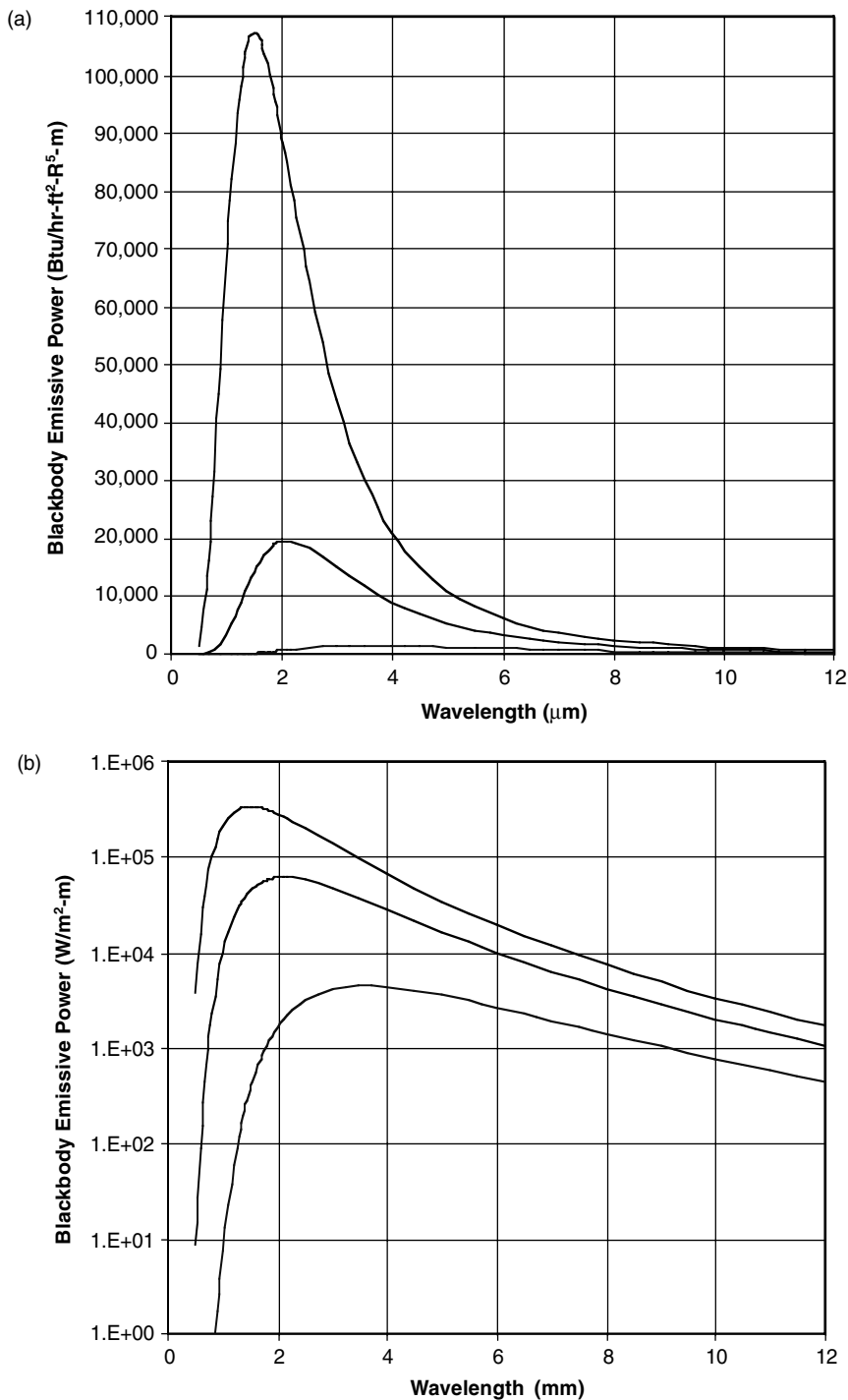


FIGURE 2.3 Blackbody emissive power: (a) English units, linear scale, and (b) metric units, logarithmic scale.¹

This wavelength may be important for certain types of heat loads that selectively absorb radiation at preferred wavelengths. In those cases, to maximize the radiant efficiency, the heat source temperature should be coupled to the preferred wavelength of the load. Note that this will not necessarily maximize the radiation to the load as higher temperatures produce more radiation and

more energy may be absorbed by the load, even if the spectral absorptivity is lower because of the higher radiant power density, as will be shown in a later example.

The integrated hemispherical emissive power of a blackbody, computed over all wavelengths, is a function only of its absolute temperature and is described by:

$$e_b = \sigma T^4 \quad (2.15a)$$

$$E_b = \sigma A T^4 \quad (2.15b)$$

where A is the area (ft^2 or m^2), T is the absolute temperature of the body ($^\circ\text{R}$ or K), and σ is the Stefan-Boltzmann constant, which has the value of:

$$\sigma = 0.1714 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-}^\circ\text{R}^4 \quad (5.669 \times 10^{-8} \text{ W/m}^2\text{-K}^4) \quad (2.16)$$

Example 2.2

Given: Surface temperature = 2000°F .

Find: Blackbody emissive power.

Solution: $e_b = \sigma T^4 = (0.1714 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-}^\circ\text{R}^4)(2000 + 460^\circ\text{R})^4 = 62,800 \text{ Btu/hr-ft}^2$

If all of the radiation received at a solid surface is absorbed by that surface, the body is referred to as a *blackbody* and has an absorptivity of unity ($\alpha = 1$). In addition, a blackbody not only absorbs all incident radiation, but it also emits the maximum amount of energy possible for the given temperature. The emissivity (ϵ) of a medium is the fraction of energy a body emits (E) for a given temperature, compared to the amount it could emit (E_b):

$$\epsilon = \frac{E}{E_b} \quad (2.17)$$

Then, for a blackbody $E = E_b$ and $\epsilon = 1$. Real bodies are typically not perfect blackbodies and do not absorb all the incident energy received on their surfaces or emit the maximum amount of energy possible. The radiant heat transfer absorbed by real surfaces is a function of the absorptivity of the surface, and the total radiant energy emitted by a body can be calculated from:

$$E = \epsilon \sigma A T^4 \quad (2.18)$$

Example 2.3

Given: Surface temperature = 2000°F , $\epsilon = 0.4$, $A = 10 \text{ ft}^2$.

Find: Total radiant energy emitted by the body.

Solution: $E_b = \epsilon \sigma A T^4 = (0.4)(0.1714 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-}^\circ\text{R}^4)(10 \text{ ft}^2)(2000 + 460^\circ\text{R})^4 = 251,000 \text{ Btu/hr}$

Thus, a surface with an emissivity of 0.4 will emit 40% of the energy that a blackbody would emit at the same temperature.

The above equation for E_b assumes that the emissivity is a constant value. The absorptivity and emissivity of some surfaces is a function of the temperature and the wavelength of radiation. Kirchoff's law states that:

$$\epsilon(\lambda, T) = \alpha(\lambda, T) \quad (2.19)$$

so that, at equilibrium conditions, a surface absorbs and emits the same amount of radiation. This assumes that the surface condition remains the same. If the absorptivity and emissivity are independent of wavelength, the surface is said to be a *graybody*. In engineering calculations, most surfaces can be treated as graybodies. The spectral emissive power can be written as:

$$e_{\lambda} = \epsilon_{\lambda} \sigma T^4 \quad (2.20)$$

The following example shows a comparison of the radiation absorbed as a function of wavelength for a load whose emissivity varies with wavelength.

Example 2.4

Given: Surface emissivities: $\epsilon_{2\mu\text{m}} = 0.4$, $\epsilon_{3\mu\text{m}} = 0.8$.

Find: Radiant energy absorbed for heat sources matched to these wavelengths.

Solution: Find temperature whose blackbody curve peaks at the given wavelengths:

$$T_1 = \frac{5215.6 \mu\text{m}^\circ\text{R}}{2 \mu\text{m}} = 2608^\circ\text{R} \quad \text{and} \quad T_2 = \frac{5215.6 \mu\text{m}^\circ\text{R}}{3 \mu\text{m}} = 1739^\circ\text{R}$$

Then calculate the radiation that would be absorbed at those temperatures at the given wavelength:

$$\begin{aligned} e_{2\mu\text{m}} &= \epsilon_{2\mu\text{m}} \sigma T_1^4 = (0.4)(0.1714 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-}^\circ\text{R}^4)(2608^\circ\text{R})^4 \\ &= 31,700 \text{ Btu/hr-ft}^2 \end{aligned}$$

$$\begin{aligned} e_{3\mu\text{m}} &= \epsilon_{3\mu\text{m}} \sigma T_2^4 = (0.8)(0.1714 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-}^\circ\text{R}^4)(1739^\circ\text{R})^4 \\ &= 12,500 \text{ Btu/hr-ft}^2 \end{aligned}$$

As can be seen in this example, the spectral radiant efficiency is higher at 3 μm with a source temperature of 1739 $^\circ\text{R}$ because 80% of the radiant energy is absorbed ($\epsilon_{3\mu\text{m}} = 0.8$). However, much more energy is absorbed at the higher source temperature, although it is less efficient. Figure 2.4

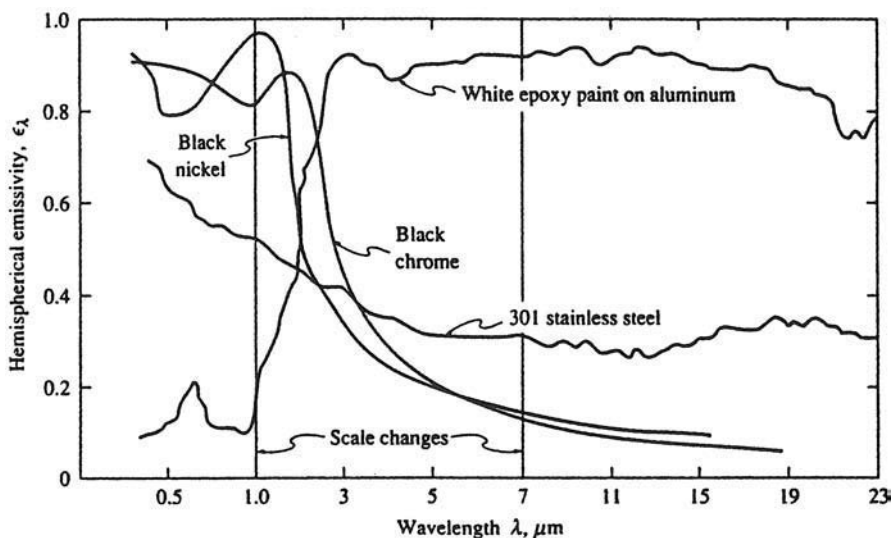


FIGURE 2.4 Spectral, hemispherical emissivities of several spectrally selective surfaces.¹³⁶

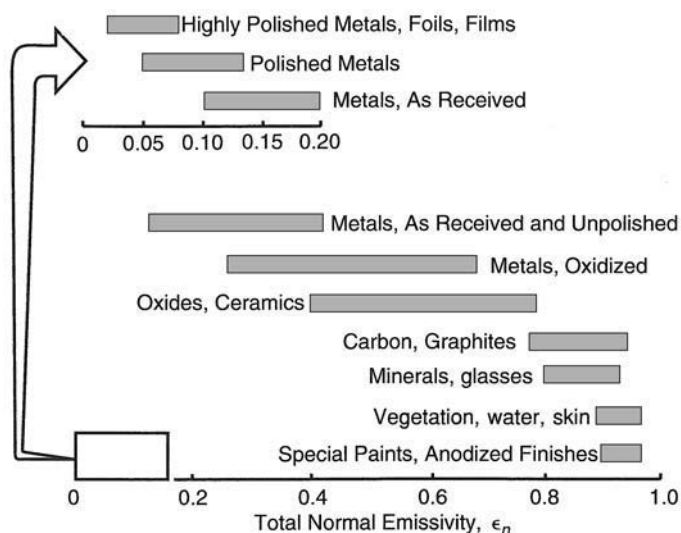


FIGURE 2.5 Total emissivity ranges for various materials.¹³⁷

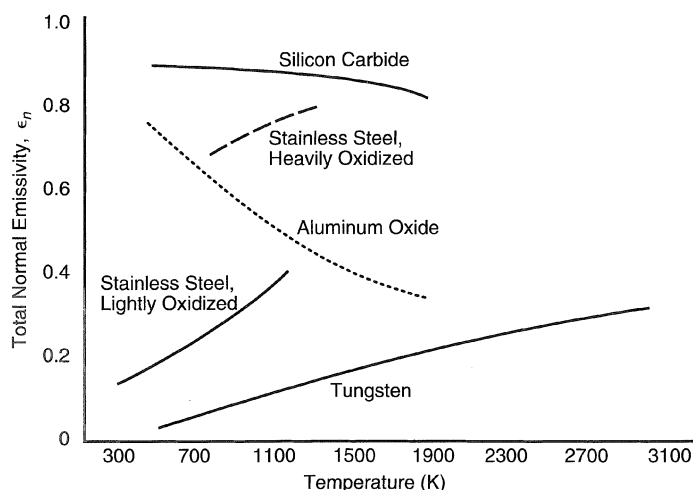


FIGURE 2.6 Total normal emissivity as a function of temperature for various materials.¹³⁷

shows an example of the spectral emissivity of some materials. Figure 2.5 shows that some materials may have a wide a range of total emissivities. Figure 2.6 shows how the total normal emissivity varies with temperature. Table 2.1 gives a list of normal total emissivities for various surfaces as a function of temperature. Table 2.2 shows total emissivities for refractory materials, commonly used in furnaces, as a function of temperature.¹⁸ Glinkov (1974) gave several examples of how the emissivity varies with temperature and wavelength for different solids, including refractories, copper, brass, steel and steel alloys, and open-hearth slags.¹⁹

The emissivity of the refractory used in a combustor is important in determining the surface radiation heat transfer between the walls, the load, and the flame. The heat transfer in many industrial combustion processes is dominated by radiation from the hot refractory walls. Docherty and Tucker (1986) numerically studied the influence of wall emissivity on furnace performance.²⁰ The predictions

TABLE 2.1
Normal Total Emissivities of Various Surfaces as a Function of Temperature¹³⁷

Surface	$t(^{\circ}\text{C})$	Emissivity
A. Metals and their Oxides		
Aluminum		
Polished	100	0.095
Commercial sheet	100	0.09
Heavily oxidized	95–505	0.20–0.31
Brass		
Highly polished		
73.2 Cu, 26.7 Zn	245–355	0.028–0.031
62.4 Cu, 36.8 Zn, 0.4 Pb, 0.3 Al	255–375	0.033–0.037
82.9 Cu, 17.0 Zn	275	0.030
Rolled plate, natural surface	20	0.06
Dull plate	50–350	0.22
Oxidized by heating at 600°C	200–600	0.61–0.59
Copper		
Carefully polished electrolytic copper	80	0.018
Commercial emiered, polished, but pits remaining	20	0.030
Commercial, scraped shiny, but not mirrolike	20	0.072
Plate, heated for a long time, covered with thick oxide layer	25	0.78
Molten copper	1075–1275	0.16–0.13
Gold		
Pure, highly polished	225–625	0.018–0.035
Iron and steel (not including stainless)		
Electrolytic iron, highly polished	175–225	0.052–0.064
Steel, polished	100	0.066
Iron, polished	425–1025	0.14–0.38
Cast iron, polished	200	0.21
Cast iron, newly turned	20	0.44
Cast iron, turned and heated	880–988	0.60–0.70
Mild steel, cleaned with organic solvents	25	0.12, 0.15
Oxidized surfaces		
Iron plate, pickled, then rusted red	20	0.61
Iron plate, completely rusted	20	0.69
Rolled sheet steel	20	0.66
Cast iron, oxidized at 600°C	200–600	0.64–0.78
Steel, oxidized at 600°C	200–600	0.79
Sheet steel		
With rough oxide layer	25	0.80
With shiny oxide layer	25	0.82
Steel plate, rough	40–370	0.94–0.97
Molten surfaces		
Cast iron	1300–1400	0.29
Mild steel	1600–1800	0.28
Lead		
Pure (99.96%) unoxidized	125–225	0.057–0.075
Oxidized at 150°C	200	0.63
Mercury	0–100	0.09–0.12
Nickel		
Electroplated, polished	25	0.045
Polished	100	0.072

(Continued)

TABLE 2.1
Normal Total Emissivities of Various Surfaces as a Function of Temperature (Continued)

Surface	$t(^{\circ}\text{C})$	Emissivity
Electroplated, not polished	20	0.11
Plate, oxidized by heating at 600°C	200–600	0.37–0.48
Nickel alloys		
Copper-nickel, polished	100	0.059
Nichrome wire, bright	50–1000	0.65–0.79
Platinum		
Pure polished plate	225–625	0.054–0.104
Silver		
Polished, pure	225–625	0.020–0.032
Stainless steels		
Type 304 (8 Cr; 18 Ni) Polished	100	0.074
Light silvery, rough, brown, after heating	215–490	0.44–0.36
After 42 h at 525°C	215–525	0.62–0.73
Thorium oxide	275–500	0.58–0.36
Thorium oxide	500–825	0.36–0.21
Tin		
Bright	50	0.06
Commercial tin-plated sheet iron	100	0.07, 0.08
Zinc		
Commercial 99.1% pure, polished	225–325	0.045–0.053
Oxidized by heating at 400°C	400	0.11
Galvanized sheet iron, fairly bright	30	0.23
Galvanized sheet iron, gray oxidized	25	0.28
B. Refractories, Building Materials, Paints, and Miscellaneous Materials		
Alumina (99.5–85 Al_2O_3 ; 0–12 SiO_2 ; 0–1 Fe_2O_3)		
Effect of mean grain size		
10 μm	1010–1565	0.30–0.18
50 μm		0.39–0.28
100 μm		0.50–0.40
Alumina-silica (showing effect of Fe)		
80–58 Al_2O_3 ; 16–38 SiO_2 ; 0.4 Fe_2O_3	1010–1570	0.61–0.43
36–26 Al_2O_3 ; 50–60 SiO_2 ; 1.7 Fe_2O_3		0.73–0.62
61 Al_2O_3 ; 35 SiO_2 ; 2.9 Fe_2O_3		0.78–0.68
Asbestos		
Board	25	0.96
Paper	40–370	0.93–0.94
Brick		
Red, rough but no gross irregularities	20	0.93
Building	1000	0.45
Fireclay	1000	0.75
Carbon		
Filament	1040–1405	0.526
Graphitized	100–320	0.76–0.75
Graphitized	320–500	0.75–0.71
Thin layer on iron plate	20	0.927
Thick coat	20	0.967
Glass		
Smooth	20	0.94
Magnesite refractory brick	1000	0.38

(Continued)

TABLE 2.1
Normal Total Emissivities of Various Surfaces as a Function of Temperature (Continued)

Surface	$t(^{\circ}\text{C})$	Emissivity
Paints, lacquers, varnishes		
Snow-white enamel varnish on rough iron plate	25	0.906
Black shiny lacquer, sprayed on iron	25	0.875
Radiator paint; white, cream, bleach	100	0.79, 0.77, 0.84
Plaster, rough line	10–90	0.91
Quartz		
Rough, fused	20	0.93
Glass, 1.98 mm thick	280–840	0.90–0.41
Glass, 6.88 mm thick	280–840	0.93–0.47
Silica (98 SiO ₂ ; Fe-free), effect of grain size		
10 μm	1010–1565	0.42–0.33
70–600 μm		0.62–0.46
Water	0–100	0.95–0.963

From data tabulated by Hottel and Sarofim (1967). With permission.

TABLE 2.2
Normal Total Emissivities of Refractories as a Function of Temperature¹⁸

Material	Temperature, $^{\circ}\text{F}$ ($^{\circ}\text{C}$)						
	200 (93)	400 (200)	800 (430)	1600 (870)	2000 (1090)	2400 (1320)	2800 (1540)
Fireclay brick	0.90	0.90	0.90	0.81	0.76	0.72	0.68
Silica brick	0.90	—	—	0.82–0.65	0.78–0.60	0.74–0.57	0.67–0.52
Chrome-magnesite brick	—	—	—	0.87	0.82	0.75	0.67
Chrome brick	0.90	—	—	0.97	0.98	—	—
High-alumina brick	0.90	0.85	0.79	0.50	0.44	—	—
Mullite brick	—	—	—	0.53	0.53	0.62	0.63
Silicon carbide brick	—	—	—	0.92	0.89	0.87	0.86

showed that fuel consumption decreases as furnace wall emissivity increases, although there is no effect when the furnace atmosphere is gray. Transient furnace operation or poor wall insulation reduces the beneficial effects of high wall emissivity. Elliston et al. (1987) analytically and numerically showed that wall surface emissivity has only a negligible impact on heat transfer to a load inside a furnace.²¹ In a study sponsored by the Gas Research Institute (Chicago, IL), it was determined that the total normal emittances of common, commercially available refractories, including dense insulating firebrick and porous ceramic fiber, ranged from 0.3 to 0.7 at 1800°F (1300K).²² The emissivity of so-called high-emittance coatings was also measured, and it was found that few of the coatings actually had high emissivities. Those coatings could extend the range of refractory emittances from 0.3 to 0.9 at 1800°F (1300K). Another part of that study investigated the effect of using high-emittance coatings in large industrial furnaces.²³ Testing was done on a 50-ton/hr (45 m-ton/hr) reheat furnace. Experimental results showed no significant change in furnace efficiency (at most, a 6% increase) using the highest available emissivity coatings on the refractory walls of the furnace. Another study showed that high-emissivity coatings applied to hundreds of furnaces in metallurgical, petrochemical, ceramic, mechanical, and other industries in the Peoples Republic

of China afforded energy savings of 5 to 10%, up to a maximum of 28%.²⁴ The reported emissivity of the coating ranged from 0.80 to 0.92, depending on the surface temperature and wavelength.

In some industrial combustion processes, the surface absorptivity can change over time, which affects the performance of the system. This is particularly true of coal-fired processes, where the ash may be deposited on tube surfaces. Wall et al. (1993) showed that the normal emittance of particulate ash is highly dependent on the particle size and surface temperature.²⁵ In a subsequent study, Wall et al. (1995) studied the effects of ash deposits on the heat transfer in a coal-fired furnace.²⁶ The heat transfer from the combustion products to the tubes is primarily by radiation, with a lesser amount by convection heating. The heat must conduct through the ash deposits and the tube wall, before heating the fluid inside the tubes primarily by convection. As the ash deposit melts to form a slag, the absorptance increases dramatically to values approaching 0.9. The thermal conductivity of the deposit is highly dependent on its physical state, especially its porosity. It is interesting to note that the heat flux through the deposits decreases during the initial phases of its growth and then actually increases before reaching a steady-state value when the deposits have reached maturity. The measured thermal conduction coefficient ranged from 0.3 to 0.5 kW/m²-K (53 to 88 Btu/ft²-hr-°F). The important conclusion of these studies is that the radiative and conductive properties of ash deposits depend on the physical and chemical character of the deposits.

An important analysis in combustion processes is often radiation heat transfer in an enclosure. The furnace walls are normally at a higher temperature than the heat load and radiate energy to that load. When the space inside an enclosure is either a vacuum or contains a gas such as air that is essentially transparent to radiation, then the medium in the combustion space is referred to as nonparticipating. This means that it does not absorb any of the radiation passing through it. If the combustion space contains a radiatively absorbing gas such as CO₂, H₂O, or CO, then the medium is referred to as *participating*, because it does absorb some of the radiation passing through it. In the combustion of fossil fuels, the products of combustion usually contain significant quantities of CO₂ and H₂O, so the combustion space contains participating media. An assumption that is often made for a first-order analysis is that the concentrations of participating gases are low enough due to dilution by N₂ that the combustion space can be treated as nonparticipating to simplify the analysis. The net radiant heat transfer from one surface to another can be calculated from:

$$q_{1\leftrightarrow 2} = \sigma A_1 F_{1\rightarrow 2} (T_1^4 - T_2^4) = \sigma A_2 F_{2\rightarrow 1} (T_1^4 - T_2^4) \quad (2.21)$$

where $q_{1\leftrightarrow 2}$ is the net energy transferred between surfaces 1 and 2; $F_{i\rightarrow j}$ is the view factor or radiation shape factor, which is the diffuse radiation leaving surface i and received at surface j ; and,

$$A_1 F_{1\rightarrow 2} = A_2 F_{2\rightarrow 1} \quad (2.22)$$

due to the reciprocity theory. Equations, charts, tables, and graphs of radiation view factors are available elsewhere.^{10,14} An example will be given to illustrate this type of analysis. The view factor between two identical, parallel, directly opposed rectangles (see [Figure 2.7](#)) is given by:

$$F_{1\rightarrow 2} = F_{2\rightarrow 1} = \frac{2}{\pi XY} \left\{ \ln \sqrt{\frac{(1+X^2)(1+Y^2)}{1+X^2+Y^2}} + X\sqrt{1+Y^2} \tan^{-1} \frac{X}{\sqrt{1+Y^2}} \right. \\ \left. + Y\sqrt{1+X^2} \tan^{-1} \frac{Y}{\sqrt{1+X^2}} - X \tan^{-1} X - Y \tan^{-1} Y \right\} \quad (2.23)$$

where $X = a/c$, $Y = b/c$, a = length of the rectangles, b = width of the rectangles, and c = spacing between the rectangles. As can be seen, this is a fairly complicated relationship for a relatively simple geometric configuration. For more complicated and for general surface orientations, computer analysis becomes necessary.

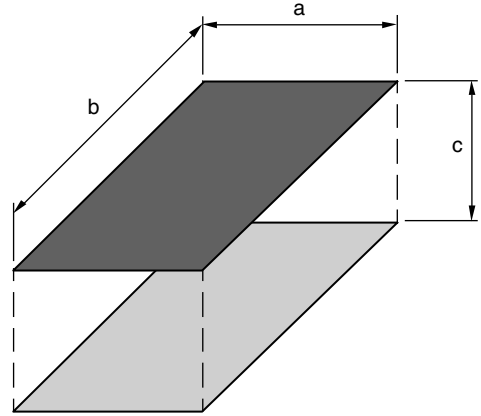


FIGURE 2.7 Geometry for radiation between parallel plates.¹

Example 2.5

Given: Furnace roof temperature of 2000°F, average load temperature = 300°F, combustion space length = 20 ft, width = 10 ft, height = 8 ft.

Find: Net radiant heat transfer from the roof to the floor assuming both are blackbodies.

Solution: Find view factor from floor to ceiling:

$$a = 20 \text{ ft}, b = 10 \text{ ft}, c = 8 \text{ ft}, X = a/c = 20/8 = 2.5, Y = b/c = 10/8 = 1.25$$

Using the formula above for the view factor between parallel rectangles,

$$F_{1 \rightarrow 2} = 0.356$$

$$T_1 = 2000^\circ\text{F} = 2460^\circ\text{R}, T_2 = 300^\circ\text{F} = 760^\circ\text{R}$$

$$\begin{aligned} q_{1 \rightarrow 2} &= (0.1714 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-R}^4)(0.356)(20' \times 10')[(2460^\circ\text{R})^4 - (760^\circ\text{R})^4] \\ &= 4.43 \times 10^6 \text{ Btu/hr} \end{aligned}$$

Beér studied flame impingement inside a furnace and experimentally determined that radiation was at least 10% of the total heat flux.⁹⁵ It was not specified how much of that radiation came from the hot walls and how much came from the flame. Vizioz¹¹¹ and Smith¹¹³ studied flame impingement on a flat plate in a hot furnace. It was determined that radiation and convection were of comparable magnitudes. Vizioz measured radiation to be 4 to 100% of the total heat flux. Smith calculated the surface radiant emission to the target using Hottel's zone method.¹⁰ It was 30 to 43% and 10 to 17% of the total heat transfer to water-cooled and air-cooled flat plates, respectively. Matsuo studied turbulent, preheated air/coke oven gas flames, impinging on a metal slab, inside a hot furnace.¹¹⁴ The top of the slab was exposed to the impinging flame. The remainder was exposed to the radiation from the furnace walls. Furnace radiation was the dominant mechanism: (1) for large L , (2) for high t_w , and (3) for large R . Ivernel calculated the radiation from hot furnace walls to a hemi-nosed cylinder.¹¹⁶ It was up to 42% of the total heat flux for impinging O_2 /natural gas flames. You measured the convective and total heat flux, using gages plated with gold and black foils, respectively.¹⁰⁷ By subtracting the convection from the total heat flux, radiation was calculated to be up to 35% of the convective flux. Van der Meer estimated that the radiation from the hot inner refractory wall of a tunnel burner was up to 15% of the total heat flux to the target.¹²²

2.2.2 NONLUMINOUS GASEOUS RADIATION

2.2.2.1 Theory

The equation of radiative transfer will not be considered here, as it is discussed in many radiation textbooks and is not commonly used as such to solve industrial combustion problems. The complete combustion of hydrocarbon fuels produces, among other things, CO_2 and H_2O . These gaseous

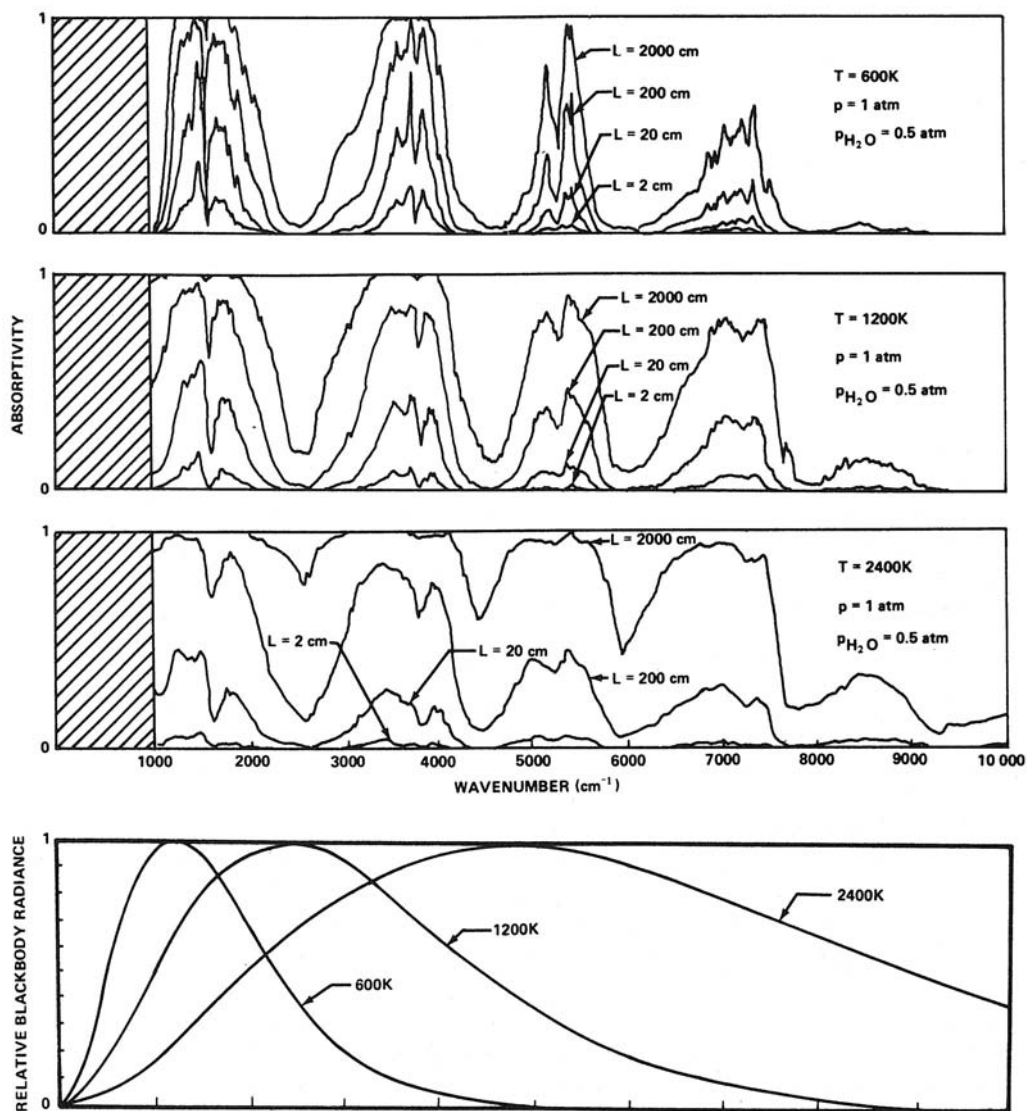


FIGURE 2.8 Calculated spectral emissivity of H_2O as a function of wavelength and path length. (From Ludwig et al., *Handbook of Infrared Radiation from Combustion Gases*, NASA, 1973.)

products generate nonluminous radiation, which has been extensively studied.¹¹ This heat transfer mode depends on the gas temperature level, the partial pressure and concentration of each species, and the molecular path length through the gas. Based on the study of CO flames, Garner (1965) concluded that most of the radiant energy from the flames was primarily chemiluminescence,²⁷ but this was later proven wrong.²⁸ Ludwig et al. (1973) gave a very detailed discussion of the theory of infrared radiation from combustion gases, including the terms and their definitions, calculation techniques for both homogeneous and inhomogeneous gases, models for specific molecules (both diatomic and polyatomic), actual computed radiation data, a discussion of the accuracy of the models, and some predictive techniques for calculating rocket exhausts.²⁹ Examples of the spectral emissivity of H_2O and O_2 as a function of wavelength and path length are shown in [Figure 2.8](#) and [Figure 2.9](#), respectively.

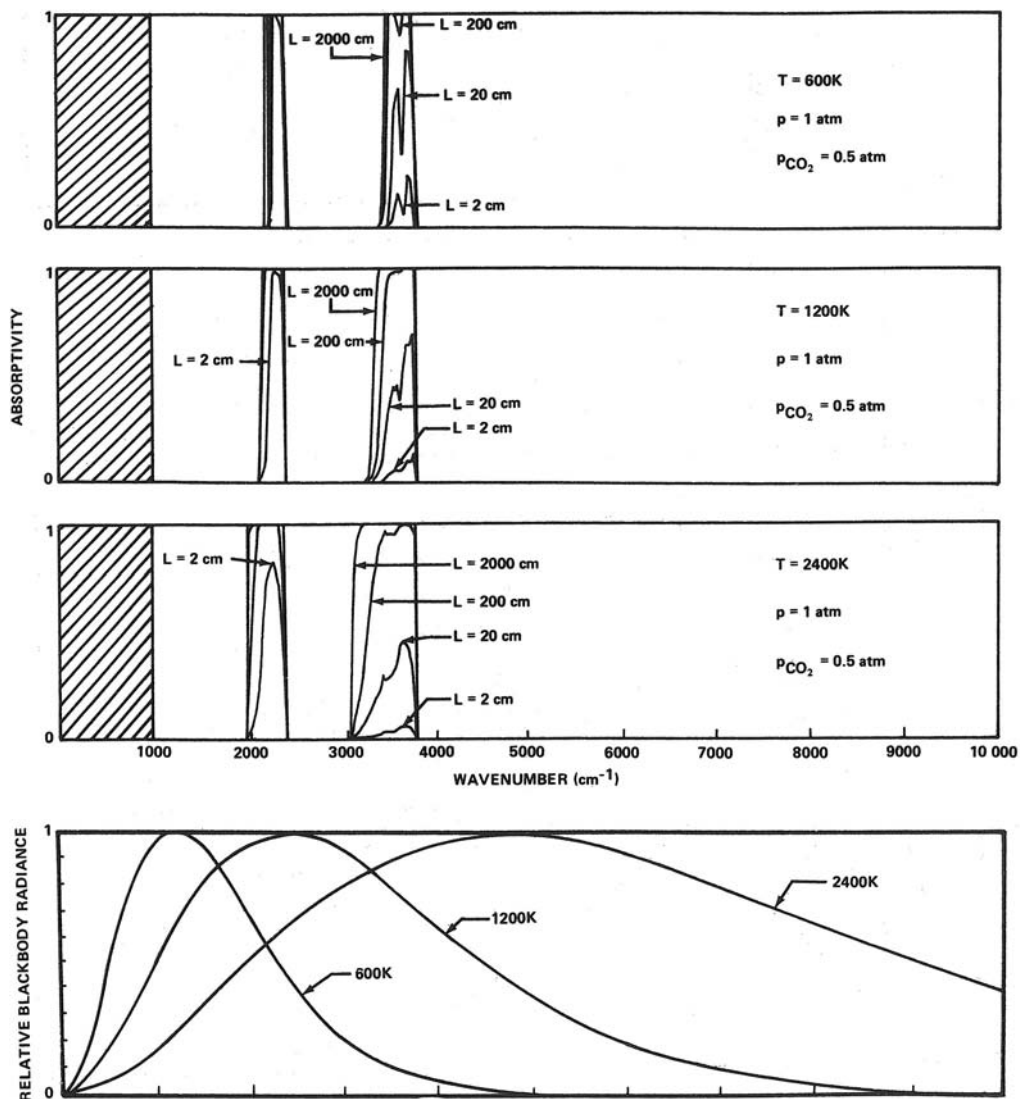


FIGURE 2.9 Calculated spectral emissivity of CO_2 as a function of wavelength and path length. (From Ludwig et al., *Handbook of Infrared Radiation from Combustion Gases*, NASA, 1973.)

The total emissivity can be calculated from Leckner (1972).³⁰ The individual emissivity of either CO_2 or H_2O is given by:

$$\epsilon_i(p_a L, p, T_g) = \epsilon_0(p_a L, T_g) \left(\frac{\epsilon}{\epsilon_0} \right) (p_a L, p, T_g) \quad (2.24)$$

where ϵ_i = emissivity of the individual gas, p_a = partial pressure of the gas, L = path length through the gas, T_g = absolute temperature of the gas, and ϵ_0 = emissivity of the individual gas at a reference

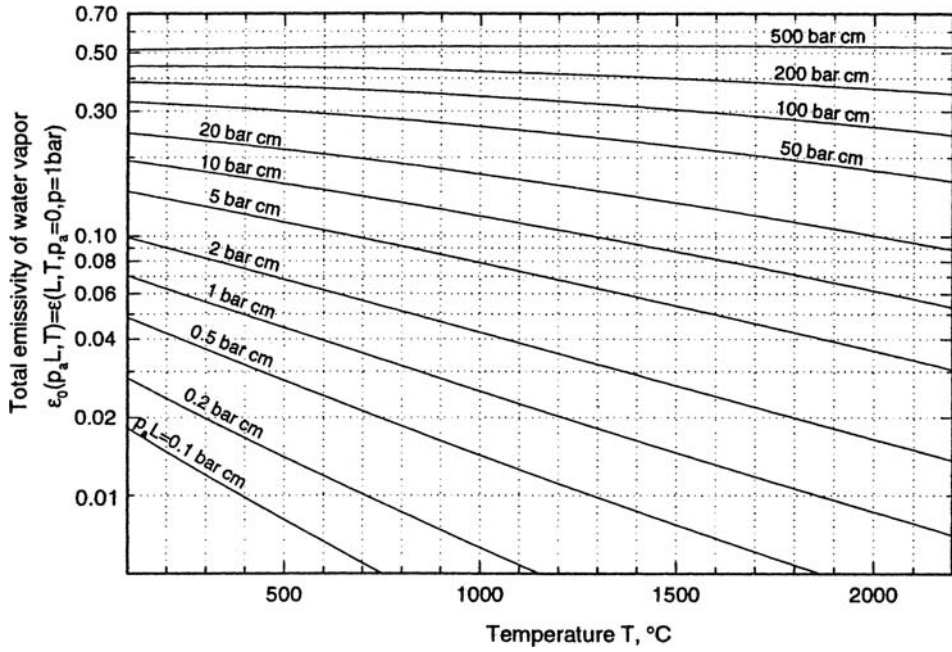


FIGURE 2.10 Total emissivity of water vapor at the reference state of a total gas pressure $p = 1$ bar and a partial pressure of H_2O $p_a \rightarrow 0$.¹³⁶

state (atmospheric pressure and $p_a \rightarrow 0$ but $p_a L > 0$). The first term in Equation 2.24 is calculated using:

$$\epsilon_0(p_a L, T_g) = \exp \left[\sum_{i=0}^M \sum_{j=0}^N c_{ij} \left(\frac{T_g}{T_0} \right)^j \left(\log_{10} \frac{p_a L}{(p_a L)_0} \right)^i \right] \quad (2.25)$$

where T_0 = absolute reference temperature of the gas (1000K) and c_{ij} are constants. The second term of the equation is calculated from:

$$\left(\frac{\epsilon}{\epsilon_0} \right) (p_a L, p, T_g) = \left\{ 1 - \frac{(a-1)(1-P_E)}{a+b-1+P_E} \exp \left[-c \left(\log_{10} \frac{(p_a L)_m}{p_a L} \right)^2 \right] \right\} \quad (2.26)$$

where a , b , c , P_E , and $(p_a L)_m/p_a L$ are given in Table 2.3. Graphical results for H_2O and CO_2 are shown in Figure 2.10 and Figure 2.11, respectively. The total emissivity is then calculated using:

$$\epsilon_{\text{CO}_2+\text{H}_2\text{O}} = \epsilon_{\text{CO}_2} + \epsilon_{\text{H}_2\text{O}} - \Delta\epsilon \quad (2.27)$$

where the $\Delta\epsilon$ accounts for the overlap between the H_2O and CO_2 bands and is calculated from:

$$\Delta\epsilon = \left(\frac{\xi}{10.7 + 101\xi} - 0.0089\xi^{10.4} \right) \left(\log_{10} \frac{(p_{\text{H}_2\text{O}} + p_{\text{CO}_2})L}{(p_a L)_0} \right)^{2.76} \quad (2.28)$$

TABLE 2.3
Correlation Constants for the Determination of the Total Emissivity for Water Vapor and Carbon Dioxide¹³⁶

Gas			Water Vapor				Carbon Dioxide		
M, N			2, 2				2, 3		
c_{00}	$\cdots$	c_{N1}	-2.2118	-1.1987	0.035596	-3.9893	2.7669	-2.1081	0.39163
$\vdots$	$\ddots$	$\vdots$	0.85667	0.93048	-0.14391	1.2710	-1.1090	1.0195	-0.21897
c_{0M}	$\cdots$	c_{NM}	-0.10838	-0.17156	0.045915	-0.23678	0.19731	-0.19544	0.04464
P_E			$(p + 2.5bp_a/\sqrt{t})/p_c$				$(p + 0.28p_a)/p_0$		
$(p_0L)_m/(p_aL)_0$			$13.2t^2$				$0.054/t^2, \quad t < 0.7$		
							$0.225t^2, \quad t > 0.7$		
a			$2.144, \quad t < 0.75$				$1 + 0.1/t^{1.45}$		
			$1.88-2.053 \log_{10} t, \quad t > 0.75$						
b			$1.10/t^{1.4}$				0.23		
c			0.5				1.47		

Note: $T_0 = 1000$ K, $p_0 = 1$ bar, $t = T/T_0$, $(p_a, L)_0 = 1$ bar cm.

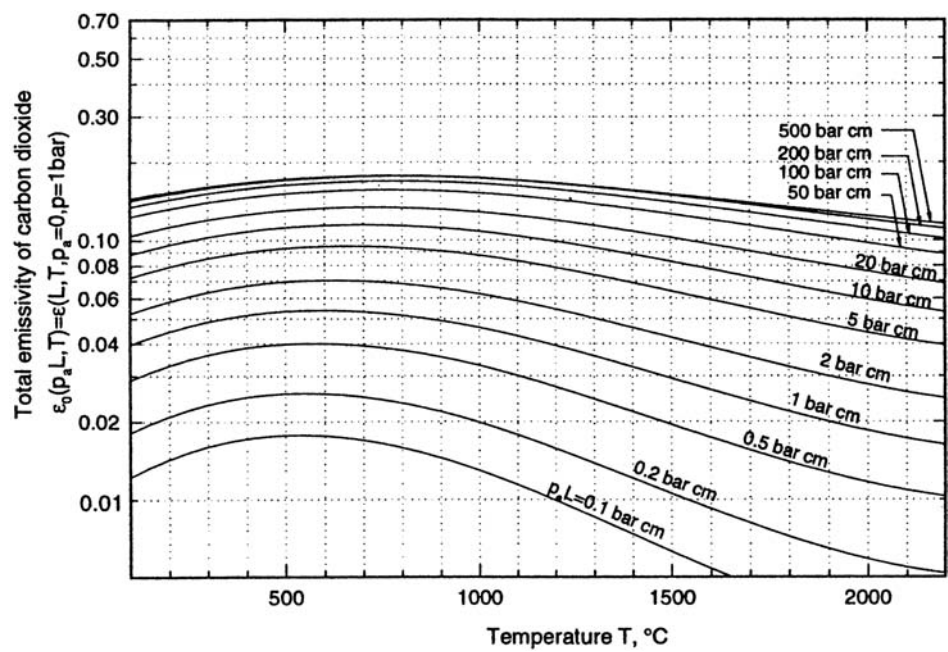


FIGURE 2.11 Total emissivity of carbon dioxide at the reference state of a total gas pressure $p = 1$ bar and a partial pressure of CO_2 $p_a \rightarrow 0$.¹³⁶

and

$$\xi = \frac{p_{\text{H}_2\text{O}}}{p_{\text{H}_2\text{O}} + p_{\text{CO}_2}} \tag{2.29}$$

Example 2.6

Given: Combustion products containing 9% CO₂, 18% H₂O, and the balance N₂, at a temperature of 1500°F, with a mean beam length of 10 ft, at atmospheric pressure.

Find: Gas emissivity.

Solution: Calculate $p_a L$ for CO₂ and H₂O to use graphs:

$$\text{CO}_2: p_a L = (0.09 \text{ bar})(305 \text{ cm}) = 27 \text{ bar-cm}$$

$$\text{H}_2\text{O}: p_a L = (0.18 \text{ bar})(305 \text{ cm}) = 55 \text{ bar-cm}$$

Look up on graphs the ϵ_0 for CO₂ and H₂O at a temperature of 820°C:

$$\text{CO}_2: \epsilon_0 \approx 0.12$$

$$\text{H}_2\text{O}: \epsilon_0 \approx 0.29$$

Calculate correction factors ϵ/ϵ_0 using Equation 2.26 and Table 2.3:

$$t = T/T_0 = (1090\text{K}/1000\text{K}) = 1.09$$

$$\text{CO}_2: P_E = (1.0 + 0.28(0.09))/1.0 = 1.03$$

$$\text{H}_2\text{O}: P_E = (1.0 + 2.56/\sqrt{1.09})/1.0 = 3.45$$

$$\text{CO}_2: \frac{(p_a L)_m}{p_a L} = 0.225t^2 = 0.225(1.09)^2 = 0.267$$

$$\text{H}_2\text{O}: \frac{(p_a L)_m}{p_a L} = 13.2t^2 = 13.2(1.09)^2 = 15.7$$

$$\text{CO}_2: a = 1 + 0.1/(1.09)^{1.45} = 1.09; b = 0.23; c = 1.47$$

$$\text{H}_2\text{O}: a = 1.88 - 2.053 \log_{10}(1.09) = 1.80; b = 1.10/(1.09)^{1.4} = 0.97; c = 0.5$$

$$\text{CO}_2: \left(\frac{\epsilon}{\epsilon_0} \right) = \left\{ 1 - \frac{(1.09 - 1)(1 - 1.03)}{1.09 + 0.23 - 1 + 1.03} \exp \left[-1.47 (\log_{10} 0.267)^2 \right] \right\} = 1.001$$

$$\text{H}_2\text{O}: \left(\frac{\epsilon}{\epsilon_0} \right) = \left\{ 1 - \frac{(1.80 - 1)(1 - 3.45)}{1.80 + 0.97 - 1 + 3.45} \exp \left[-0.5 (\log_{10} 15.7)^2 \right] \right\} = 1.184$$

$$\text{CO}_2: \epsilon = (\epsilon/\epsilon_0) \epsilon_0 = (1.001)(0.12) = 0.120$$

$$\text{H}_2\text{O}: \epsilon = (\epsilon/\epsilon_0) \epsilon_0 = (1.184)(0.29) = 0.343$$

$$\text{Calculate } \xi: \xi = \frac{0.18}{0.18 + 0.09} = 0.67$$

Calculate $\Delta\epsilon$:

$$\Delta\epsilon = \left(\frac{0.67}{10.7 + 101(0.67)} - 0.0089(0.67)^{10.4} \right) \left(\log_{10} \frac{(0.18 + 0.09)(305 \text{ cm})}{1 \text{ bar-cm}} \right)^{2.76}$$

$$\Delta\epsilon = 0.051$$

$$\epsilon = 0.120 + 0.343 - 0.051 = 0.412$$

The absorptivity of H₂O and CO₂ can be estimated using:

$$\alpha(p_a L, p, T_g, T_s) = \left(\frac{T_g}{T_s} \right)^{1/2} \epsilon \left(p_a L \frac{T_s}{T_g}, p, T_s \right) \quad (2.30)$$

where T_s is the surface temperature, such as a furnace wall. The correction for the band overlap between H₂O and CO₂ is calculated using:

$$\alpha_{\text{CO}_2+\text{H}_2\text{O}} = \alpha_{\text{CO}_2} + \alpha_{\text{H}_2\text{O}} - \Delta\epsilon \quad (2.31)$$

where $\Delta\epsilon$ is estimated with a pressure path length of $p_a L T_s / T_g$.

Example 2.7

Given: Using the data from the previous example and a wall temperature of 1000°F.

Find: Gas absorptivity.

Solution: Calculate $p_a L T_s / T_g$ for CO₂ and H₂O to use graphs:

$$\text{CO}_2: p_a L T_s / T_g = (0.09 \text{ bar})(305 \text{ cm})(1000 + 460)/(1500 + 460) = 20 \text{ bar-cm}$$

$$\text{H}_2\text{O}: p_a L T_s / T_g = (0.18 \text{ bar})(305 \text{ cm})(1000 + 460)/(1500 + 460) = 41 \text{ bar-cm}$$

Look up on graphs the ϵ_0 for CO₂ and H₂O at a temperature of 540°C (1000°F):

$$\text{CO}_2: \epsilon_0 \approx 0.12, \text{H}_2\text{O}: \epsilon_0 \approx 0.28$$

Calculate correction factors ϵ/ϵ_0 using Equation (2.26) and [Table 2.3](#):

$$t = (811\text{K}/1000 \text{ K}) = 0.811$$

$$\text{CO}_2: P_E = (1.0 + 0.28(0.09))/1.0 = 1.03$$

$$\text{H}_2\text{O}: P_E = (1.0 + 2.56/\sqrt{0.811})/1.0 = 3.84$$

$$\text{CO}_2: \frac{(p_a L)_m}{p_a L} = 0.225t^2 = 0.225(0.811)^2 = 0.148$$

$$\text{H}_2\text{O}: \frac{(p_a L)_m}{p_a L} = 13.2t^2 = 13.2(0.811)^2 = 8.68$$

$$\text{CO}_2: a = 1 + 0.1/(0.811)^{1.45} = 1.14; b = 0.23; c = 1.47$$

$$\text{H}_2\text{O}: a = 1.88 - 2.053 \log_{10}(0.811) = 2.07; b = 1.10/(0.811)1.4 = 1.47; c = 0.5$$

$$\text{CO}_2: \left(\frac{\epsilon}{\epsilon_0} \right) = \left\{ 1 - \frac{(1.14 - 1)(1 - 1.03)}{1.14 + 0.23 - 1 + 1.03} \exp \left[-1.47 (\log_{10} 0.148)^2 \right] \right\} = 1.001$$

$$\text{H}_2\text{O}: \left(\frac{\epsilon}{\epsilon_0} \right) = \left\{ 1 - \frac{(2.07 - 1)(1 - 3.84)}{2.07 + 1.47 - 1 + 3.84} \exp \left[-0.5 (\log_{10} 8.68)^2 \right] \right\} = 1.307$$

$$\text{CO}_2: \epsilon = (\epsilon/\epsilon_0) \epsilon_0 = (1.001)(0.12) = 0.120$$

$$\text{H}_2\text{O}: \epsilon = (\epsilon/\epsilon_0) \epsilon_0 = (1.307)(0.28) = 0.366$$

$$\text{CO}_2: \alpha = \sqrt{\frac{1500 + 460}{1000 + 460}} (0.120) = 0.139$$

$$\text{H}_2\text{O: } \alpha = \sqrt{\frac{1500 + 460}{1000 + 460}} \cdot 0.366 = 0.424$$

$$\alpha = 0.139 + 0.424 - 0.051 = 0.512$$

Relatively simple but accurate calculations for isothermal gases, which are absorbing–emitting but not scattering, in an isothermal black-walled enclosure can be computed using:

$$q = \sigma \left\{ [1 - \alpha(L_m)] T_w^4 - \epsilon(L_m) T_g^4 \right\} \quad (2.32)$$

where L_m is the average mean beam length and T_w is the absolute temperature of the isothermal wall. The mean beam length is a function of the geometry of the enclosure and can be calculated using the formulae in Table 2.4. For geometries not listed in that table, the following formula can be used:

$$L_m = 3.6 V/A \quad (2.33)$$

where V is the volume of the enclosure and A is the surface area inside the enclosure. Gulic (1974) gave a modified form to calculate the mean beam length:³¹

$$L_m = 3.6 \frac{V}{A} \left(\frac{\tau e^{-\tau}}{\tau - 1 + e^{-\tau} + \tau e^{-\tau}} \right) \quad (2.34)$$

where τ is the optical density of the gas.

Cess (1974) gave a relatively simple procedure for calculating infrared gaseous radiation that uses an analytical approximate band model.³² The results showed good agreement with a numerical solution. Greif (1974) presented some experimental and theoretical results for infrared radiation from a turbulent flow of air, CO₂, and steam.³³ Kovotny (1974) showed the importance of pressure in calculating gaseous radiation heat transfer.³⁴ In a review article, Edwards (1976) cautioned about the use of gray-gas models and analysis to represent the truly spectrally dependent phenomena of gaseous radiation.³⁵ Trout (1977) gave some simple methods for estimating both nonluminous and luminous radiation from industrial flames.³⁶ Tien and co-workers (1968, 1982) reviewed radiation from flames.^{37,38} They extensively reviewed the available models for both nonluminous and luminous gaseous radiation. Edwards and Balakrishnan (1973) presented some correlations for calculating the radiative properties of H₂O, CO₂, CO, NO, SO₂ and CH₄ for use in solving gas radiation problems.³⁹ Taylor and Foster (1974) presented empirical curve-fitted equations for the total emissivities of nonluminous and luminous flames, for CO₂–H₂O and CO₂–H₂O–soot mixtures arising in oil and gas combustion.⁴⁰ Coppalle and Vervisch (1983) calculated the total emissivities of high temperature (2000 to 3000K) gases (CO₂–H₂O mixtures).⁴¹ Edwards and Matavosian (1984) developed some scaling rules for calculating the total absorptivity and emissivity of gases and tabulated and graphed results for various combinations of gas composition, product of partial pressure times the path length, and gas temperature.⁴² Howell (1988) reviewed thermal radiation in participating media and gave numerous references for further information.⁴³ The P-N, two-flux, discrete ordinate, finite-element, zoning, and Monte Carlo methods of analysis were briefly discussed. Wieringa et al. (1990) modeled the spectral radiation produced by natural gas flames in a regenerative glass furnace.⁴⁴ The spectral radiation from H₂O and CO₂ were modeled using 15 spectral bands. The numerical results showed that the furnace wall emissivity had only a minimal impact on the furnace efficiency.

TABLE 2.4
Mean Beam Lengths for Radiation from a Gas Volume to a Surface on Its Boundary¹³⁶

Geometry of Gas Volume	Characterizing Dimension	Geometric Mean Beam Length	Average Mean Beam Length	
	L	L_0/L	L_m/L	L_m/L_0
Sphere radiating to its surface	Diameter, $L = D$	0.67	0.65	0.97
Infinite circular cylinder to bounding surface	Diameter, $L = D$	1.00	0.94	0.94
Semi-infinite circular cylinder to:	Diameter $L = D$			
Element at center of base		1.00	0.90	0.90
Entire base		0.81	0.65	0.80
Circular cylinder	Diameter, $L = D$			
(height/diameter = 1) to: Element at center of base		0.76	0.71	0.92
Entire surface		0.67	0.60	0.90
Circular cylinder	Diameter, $L = D$			
(height/diameter = 2) to:				
Plane base		0.73	0.60	0.82
Concave surface		0.82	0.76	0.93
Entire surface		0.80	0.73	0.91
Circular cylinder (height/diameter = 0.5) to:	Diameter, $L = D$			
Plane base		0.48	0.43	0.90
Concave surface		0.53	0.46	0.88
Entire surface		0.50	0.45	0.90
Infinite semicircular cylinder to center of plane rectangular face	Radius, $L = R$	—	1.26	—
Infinite slab to its surface	Slab thickness, L	2.00	1.76	0.88
Cube to a face	Edge L	0.67	0.6	0.90
Rectangular $1 \times 1 \times 4$ parallelepipeds:	Shortest edge, L			
To 1×4 face		0.90	0.82	0.91
To 1×1 face		0.86	0.71	0.83
To all faces		0.89	0.81	0.91

2.2.2.2 Combustion Studies

Two types of nonluminous radiation measurements have been made in previous combustion studies: total and spectral. The total radiation measurements were typically made with some type of detector that measured the overall radiation received, with no wavelength dependence. More recent measurements have been made which give the radiation as a function of the wavelength. These two methods of measurements applied to nonluminous radiation are discussed briefly here.

2.2.2.2.1 Total Radiation

In some previous combustion experimental studies, nonluminous gaseous radiation has been significant. Kilham (1949) tested nearly stoichiometric, laminar, air/CO flames, impinging normal to an uncooled refractory cylinder.⁹⁸ Flame radiation was 5 to 16% of the total heat flux. Jackson and Kilham (1956) also tested laminar flames, impinging normal to refractory cylinders.⁹⁷ A variety of fuels, oxidizers, and stoichiometries were tested. The measured flame radiation was up to 5% of the total heat flux. Dunham (1963) tested nearly stoichiometric, laminar, air/CO flames, impinging normal to the nose of a hemi-nosed cylinder.⁴⁵ The estimated nonluminous radiation was up to

13% of the total heat flux. Ivernel and Vernotte (1979) tested nearly stoichiometric, natural gas flames, impinging normal to the nose of a heminosed cylinder.¹¹⁶ The calculated nonluminous flame radiation was up to 34% of the total heat flux.

In other combustion studies, nonluminous radiation was not significant. Giedt et al. (1960) measured and calculated nonluminous radiation at less than 2% of the total heat flux.⁴⁶ Woodruff and Giedt (1966) used the same configuration as Giedt, except that the flames were turbulent.¹⁰⁰ The measured nonluminous radiation was negligible compared to the total heat flux. Shorin and Pechurkin (1968) tested a wide variety of flames impinging normal to a flat plate.¹¹⁰ Radiation was experimentally determined to be negligible. Purvis (1974) studied O_2/CH_4 and $\text{O}_2/\text{C}_3\text{H}_8$ flames, impinging normal to a flat plate.⁴⁷ The calculated nonluminous flame radiation was negligible compared to the total heat flux. Although the C_3H_8 flames were very fuel rich, there was no discussion of luminous flame radiation. Hoogendoorn et al. (1978) tested stoichiometric, laminar, air/natural gas flames impinging normal to a water-cooled flat plate.¹⁰⁵ Flame radiation effects did not exceed 5% of the total heat flux. No radiation measurements or calculations were given. Davies (1979) tested natural gas flames impinging normal to a water-cooled cylinder.⁴⁸ A range of stoichiometries and oxidizers was tested. Using an estimated flame emissivity of 0.01, the calculated nonluminous radiation was only 2% of the total heat flux. Van der Meer (1987) tested stoichiometric, laminar, and turbulent air/natural gas flames impinging normal to a water-cooled plate.⁴⁹ The flame radiation was said to be negligible because of the very low emissivity of a thin hot gas layer. No supporting calculations were cited.

Baukal and Gebhart (1997) studied total flame radiation from oxygen-enhanced natural gas flames.⁵⁰ A total, narrow-angle radiometer was used to measure the flame radiation as a function of position from the burner as shown in Figure 2.12. Five parameters were investigated, including the burner firing rate (q_f), the oxidizer composition (Ω), the equivalence ratio (ϕ), the axial position along the flame (L_r), and the radial position from the flame (D_r). The parameters of primary interest were Ω , q_f , and L_r . Most of the tests were done at $\phi = 1.0$ and $D_r = 0.5$. The equivalence ratio was only varied through a narrow range for two reasons. The first is that the equivalence ratio operates in a narrow band around stoichiometric conditions in nearly all industrial heating and melting processes. The second reason is that only nonluminous radiation was studied here. If the burner had been operated at very fuel-rich conditions ($\phi \rightarrow \infty$), luminous radiation would have become important. Figure 2.13 shows the effects of both the firing rate and the axial distance along the flame length. The minimum L_r was 0.5. For $L_r < 0.5$, the radiometer would have viewed the flame and part of the burner itself. At low firing rates, the flame radiation decreased rapidly with the distance from the burner exit. At intermediate and higher firing rates, the flame radiation was

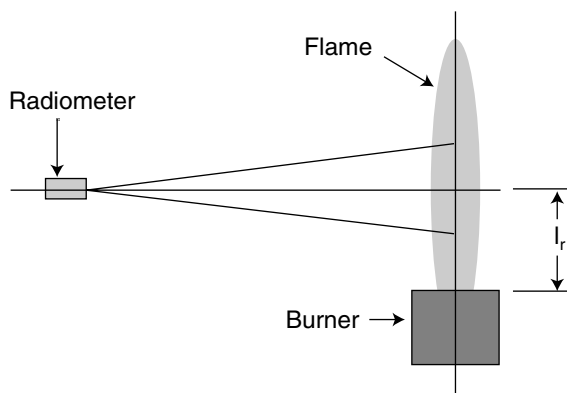


FIGURE 2.12 Experimental setup for measuring total nonluminous gaseous radiation from an open-flame diffusion burner as a function of the distance from the burner outlet.⁵⁰

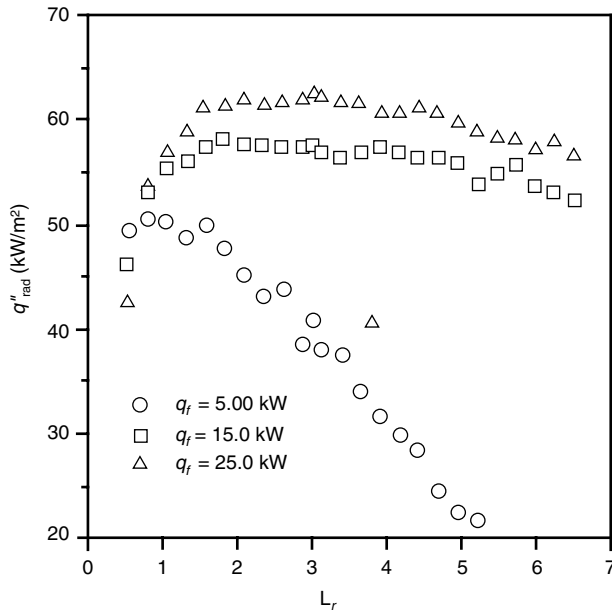


FIGURE 2.13 Total nonluminous gaseous radiation (q''_{rad}) as a function of distance from the burner (L_r) and firing rate ($q_f = 5.0, 15.0, 25.0$ kW) for $\Omega = 1.00$, $\phi = 1.00$, and $D_r = 0.5$.⁵⁰

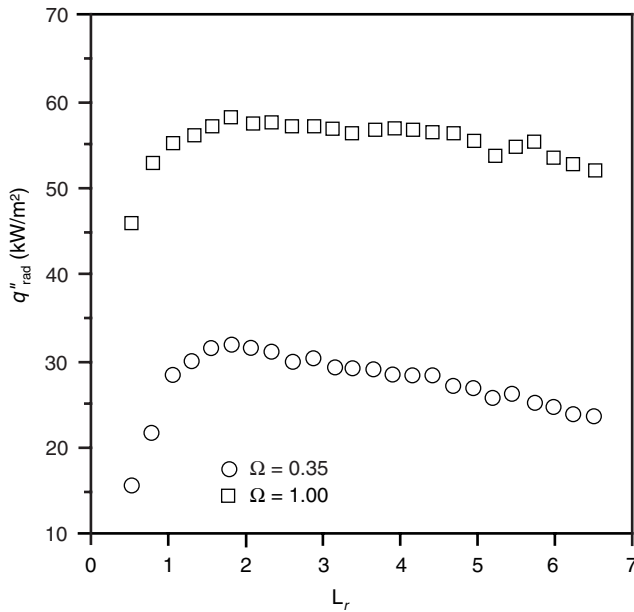


FIGURE 2.14 Total nonluminous gaseous radiation (q''_{rad}) as a function of distance from the burner outlet (L_r) and oxidizer composition ($\Omega = 0.35, 1.00$) for $q_f = 15.0$ kW, $\phi = 1.00$, and $L_r = 3.0$.⁵⁰

relatively constant for L_r from about 2 to 5. Figure 2.14 is a similar graph except that the oxidizer composition, instead of the firing rate, was varied. The flame radiation was relatively constant over a wide range of L_r , for $\Omega = 1.0$. At $\Omega = 0.35$, the peak flame radiation occurred at about $L_r = 1.5$ and then slowly decreased with L_r . Figure 2.15 shows how the radial location of the radiometer

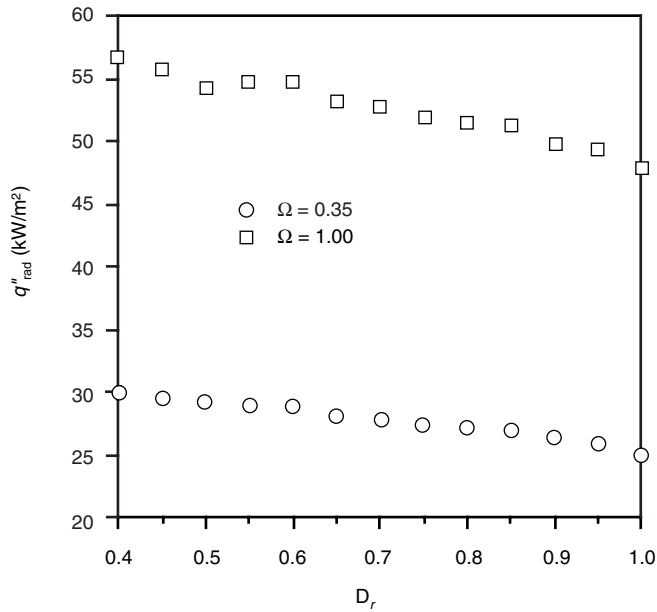


FIGURE 2.15 Total nonluminous gaseous radiation (q''_{rad}) as a function distance from the burner outlet (D_r) and oxidizer composition ($\Omega = 0.35, 1.00$) for $q_f = 15.0$ kW, $\phi = 1.00$, and $L_r = 3.0$.⁵⁰

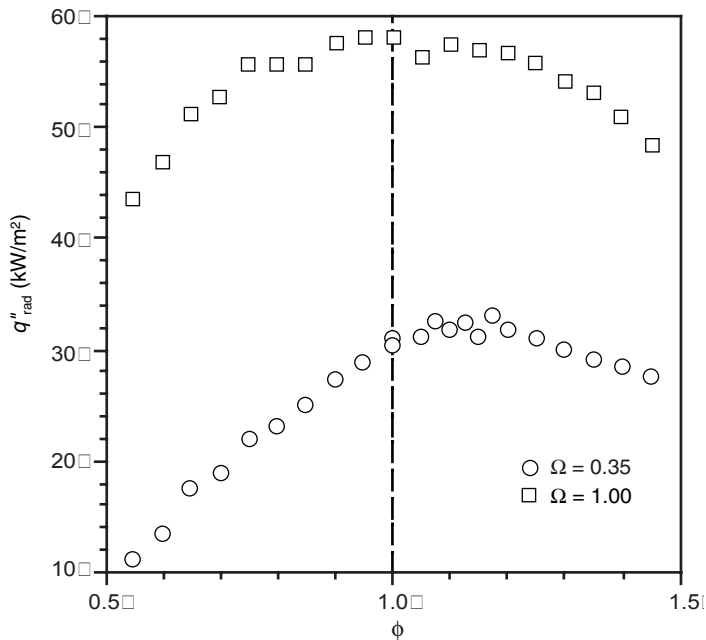


FIGURE 2.16 Total nonluminous gaseous radiation (q''_{rad}) as a function of the equivalence ratio (ϕ) and oxidizer composition ($\Omega = 0.35, 1.00$) for $q_f = 15.0$ kW, $L_r = 3.0$, and $D_r = 0.5$.⁵⁰

affected the measurements. At $D_r = 1.0$, the radiometer viewed the entire width of the flame. The effective path length decreased as D_r increased, because the flame had a circular cross section. This reduced the average flame radiation as shown in the plot. However, the reduction was relatively small. Figure 2.16 shows how the equivalence ratio and the oxidizer compositions affected the

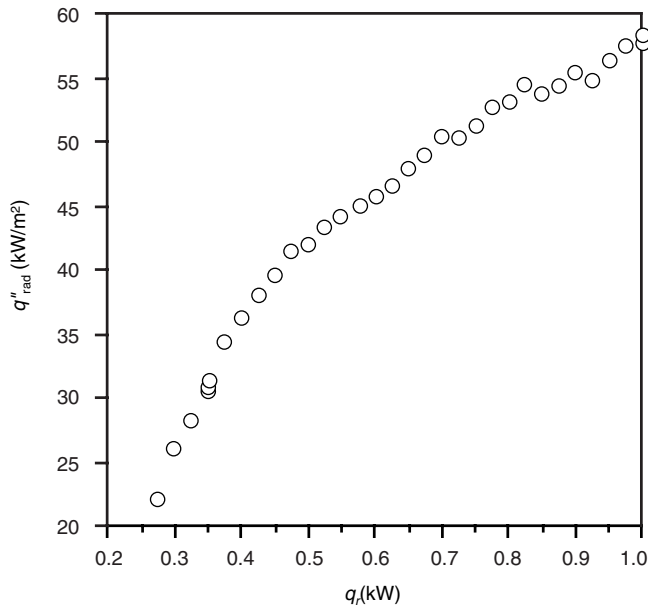


FIGURE 2.17 Total nonluminous gaseous radiation (q''_{rad}) as a function of the oxidizer composition (Ω) for $q_f = 15.0 \text{ kW}$, $\phi = 1.00$, $L_r = 3.0$, $D_r = 0.5$.⁵⁰

flame radiation. For the smaller Ω , the highest flame radiation occurred at slightly fuel-lean conditions. For a pure O_2 oxidizer, the highest flame radiation occurred in a band around stoichiometric conditions ($\phi = 1.0$). Figure 2.17 shows how the flame radiation varied as a function of the oxidizer composition for a fixed firing rate and equivalence ratio. The only thing that was varied was the amount of N_2 in the oxidizer. The fuel and O_2 flow rates were constant. Re_n decreased from 6800 to 2400 as Ω increased from 0.28 to 1.00. The plot shows that the flame radiation increased by more than 2.5 times by removing N_2 from the oxidizer. This is a result of higher flame temperatures and partial pressures of CO_2 and H_2O . Figure 2.18 shows how the flame radiation increased as the firing rate increased and as the equivalence ratio increased. Figure 2.19 is a plot of flame radiation as a function of the equivalence ratio and the firing rate for a constant Reynolds number of 4500. As N_2 was removed from the oxidizer (Ω increasing), the firing rate had to be increased to maintain a fixed Re_n . This shows that, for a given nozzle diameter, a higher heat release density can be achieved by increasing the O_2 concentration in the oxidizer. It also implies that Re_n by itself is not a sufficient parameter to indicate the performance of oxygen-enhanced flames. The oxidizer composition Ω must also be specified. Figure 2.20 shows the peak flame radiation measured for a given firing rate and oxidizer composition. For $\Omega = 1.00$, the peak radiation increased with the firing rate. For $\Omega = 0.35$, the peak radiation was relatively constant over a wide range of firing rates. Figure 2.21 shows the approximate axial location for the peak flame radiation. Initially, the peak flame locations were closer to the burner for $\Omega = 1.00$ compared to $\Omega = 0.35$. At higher firing rates, this trend reversed. For $\Omega = 1.00$, these locations were fairly well defined. For $\Omega = 0.35$, there was a range of positions along the flame length where the maximum flame radiation values were obtained. The locations given in the figure are approximately in the center of the range. Figure 2.22 is a plot of the location for the peak flame radiation that has been normalized to the visible length of the flame. This shows that for high Ω flames, the axial location of the peak flame radiation was at about 14% of the visible flame length. For the lower Ω flames, this location was between about 9 and 15% of the flame length.

The important conclusions of the study included: (1) thermal radiation increased dramatically by removing N_2 from the oxidizer; (2) the flame radiation increased with the firing rate with a more

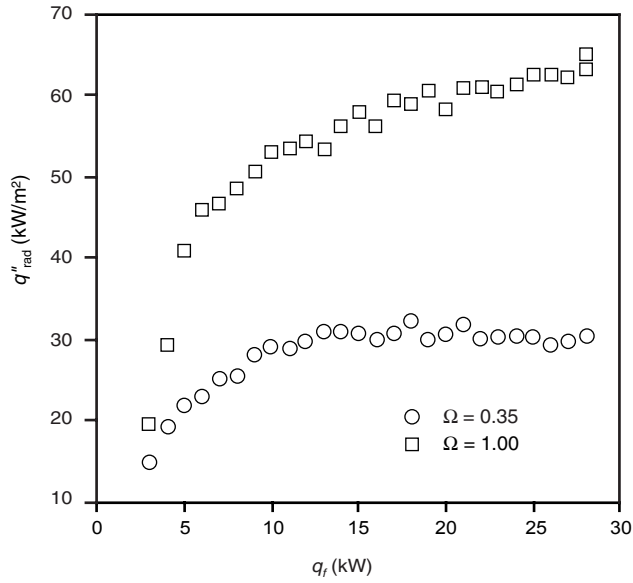


FIGURE 2.18 Total nonluminous gaseous radiation (q''_{rad}) as a function of the firing rate (q_f) and oxidizer composition ($\Omega = 0.35, 1.00$) for $\phi = 1.00$, $L_r = 3.0$, and $D_r = 0.5$.⁵⁰

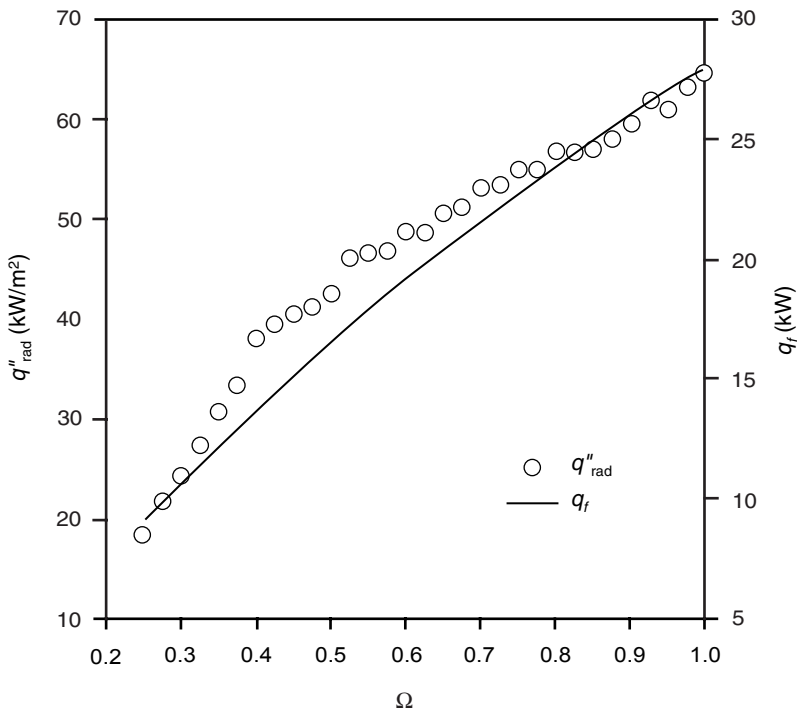


FIGURE 2.19 Total nonluminous gaseous radiation (q''_{rad}) as a function of the oxidizer composition (Ω) for $\text{Re}_n = 4500$, $\phi = 1.00$, $L_r = 3.0$, and $D_r = 0.5$.⁵⁰

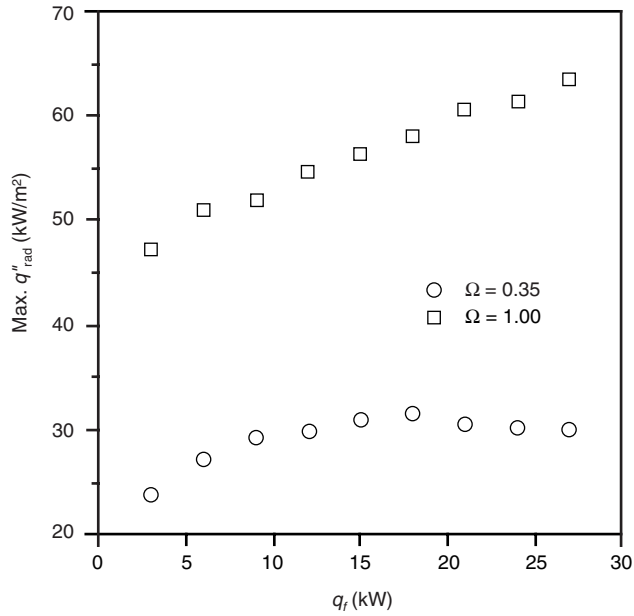


FIGURE 2.20 Maximum total nonluminous gaseous radiation (q''_{rad}) as a function of the firing rate (q_f) and oxidizer composition ($\Omega = 0.35, 1.00$) for $\phi = 1.00$ and $D_r = 0.5$ with L_r variable.⁵⁰

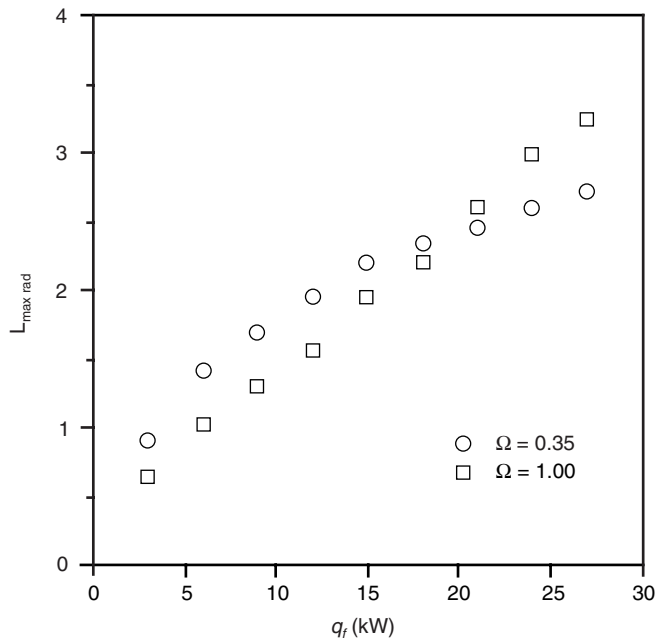


FIGURE 2.21 Axial distance of maximum radiation ($L_{\text{max rad}}$) as a function of the firing rate (q_f) and oxidizer composition ($\Omega = 0.35, 1.00$) for $\phi = 1.00$ and $D_r = 0.5$.⁵⁰

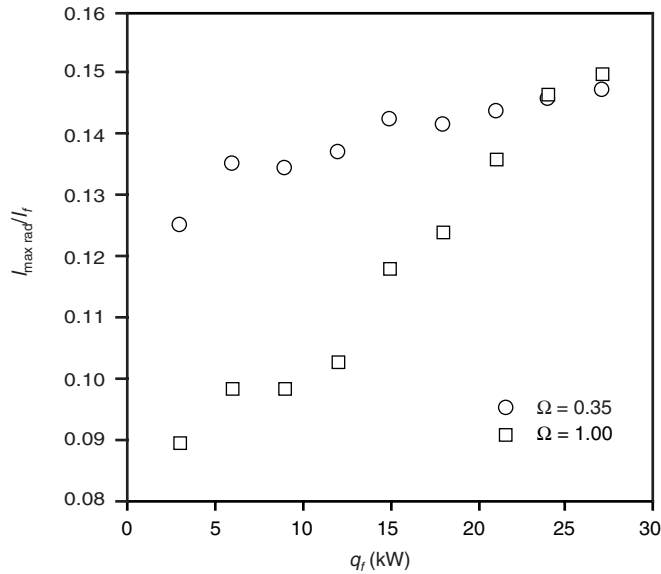


FIGURE 2.22 Normalized axial location of maximum radiation ($I_{\max \text{ rad}}/I_f$) as a function of firing rate (q_f) and oxidizer composition ($\Omega = 0.35, 1.00$) for $\phi = 1.00$ and $D_r = 0.5$.⁵⁰

dramatic increase at higher O_2 concentrations in the oxidizer; (3) higher flame radiation was measured at or near stoichiometric conditions where typical industrial heating processes operate; (4) for higher firing rates, the radiation was nearly constant over a wide range of axial locations in the flame; and (5) at lower firing rates, the flame radiation decreased with the axial distance from the burner outlet. For higher O_2 concentrations, the peak flame radiation increased with the firing rate. For lower O_2 concentrations, the peak flame radiation was nearly constant over a wide range of firing rates. The location of the peak flame radiation varied from 9 to 15% of the visible flame length. This location was more defined for the higher O_2 flames in terms of both its absolute position and the position normalized by the flame length.

2.2.2.2.2 Spectral Radiation

Ji and Baukal (1998) did the first systematic experimental study of the nonluminous and luminous spectral radiation from oxygen-enhanced flames.⁵¹ Two common quantities are measured in spectral radiometry: spectral radiance and spectral irradiance. Spectral radiance quantifies the energy flux in a unit wavelength interval that is emitting from a source surface per unit area and unit solid angle. The spectral irradiance is a measure of the energy flux in a unit wavelength interval that is incident on a target surface per unit area. For an optical source with a well-defined uniform emitting surface, spectral radiance can be measured accurately and converted into spectral irradiance. The lack of such well-defined and uniform radiating surfaces for flames renders the measurement of spectral radiance impractical. However, spectral irradiance is still a valid quantity to measure in flames and has more practical meaning in heat transfer to targets. For this reason, spectral irradiance was measured, despite the fact that spectral radiance has been used in some previous theoretical studies on flame radiation.³

A custom-built version of the commercial ICSM burner from Nordsea Gas Technology Ltd. (Cheshire, England) was used for the study as shown in Figure 2.23, where $d_n = 38.5$ mm. It was a round form of the rectangular burners⁵² used in industrial heating and melting processes. Details of this burner and its operating characteristics, as well as fuel and oxidizer flow specifications, are given elsewhere.⁵³ A key feature of the burner was the wide variety of oxidizers that can be used,

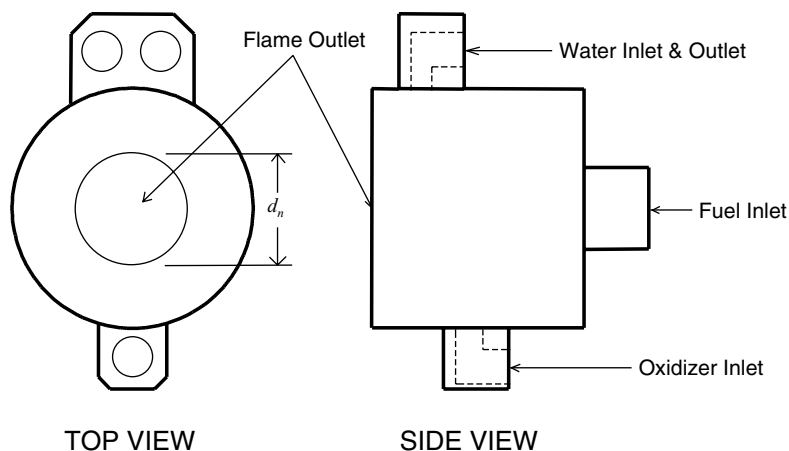


FIGURE 2.23 Burner used for flame radiation study using oxygen-enhanced burners.⁵³

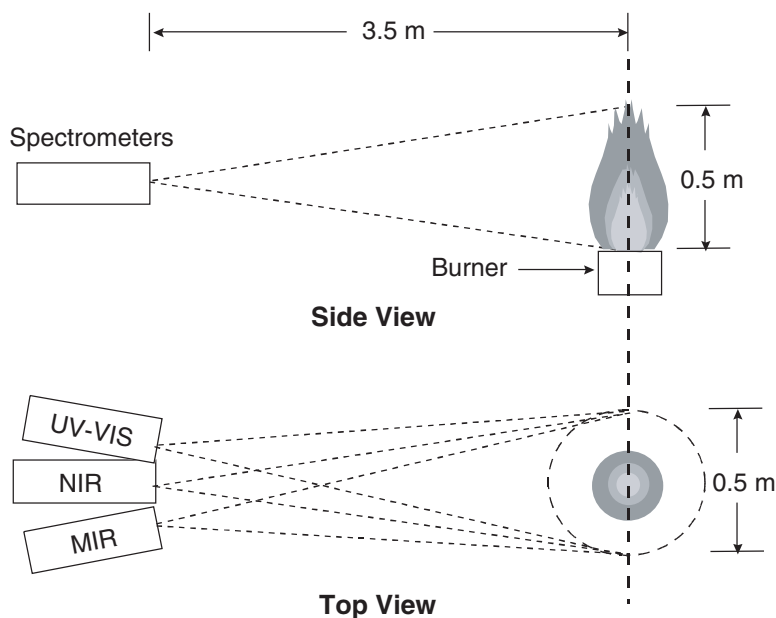


FIGURE 2.24 Experimental setup of the Monolight spectral radiometer (UV-VIS = UV and visible, NIR = near-IR, MIR = mid-IR).⁵¹

ranging from air to pure oxygen, for a range of firing rates. Another key feature was that the burner produced a uniform, nearly one-dimensional flame, so that the flame radiation was more uniform compared to diffusion flames. A firing rate of 17,000 Btu/hr (5.0 kW) was used for most of the measurements, so that there was no significant radiation at a height 1.6 ft (0.5 m) above the burner surface.

As shown in Figure 2.24, three spectral monochromators were set 11 ft (3.5 m) away from the burner centerline, and 0.82 ft (0.25 m) above the burner surface so that their field of view covered the entire radiation length of the flames. Because the flames were in open air, without any furnace enclosure and without impinging on any targets or nearby walls, ambient emission and reflection from the surroundings were negligible. Three parameters were varied to study their effects on the

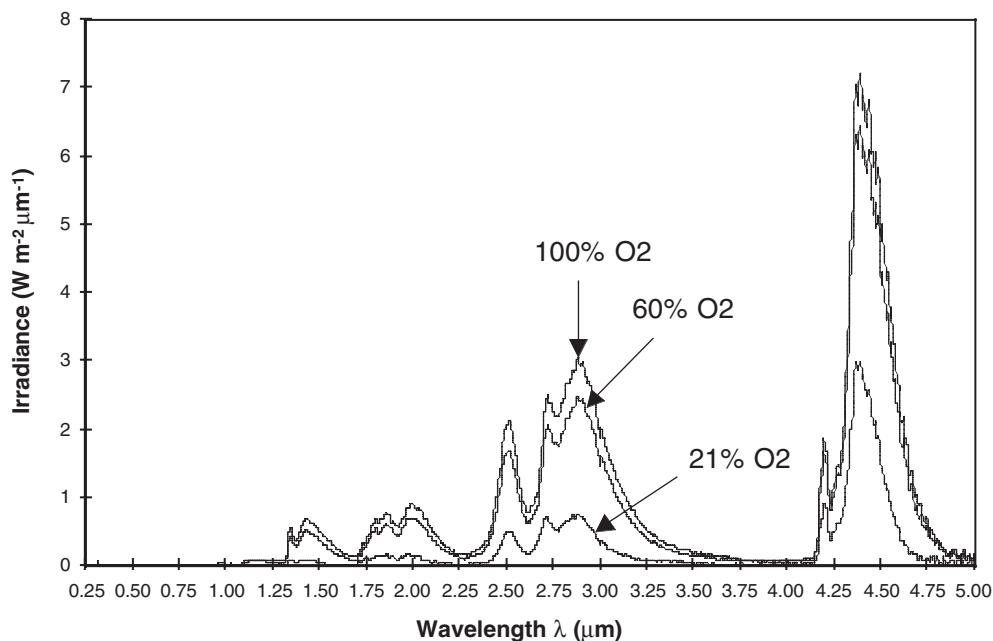


FIGURE 2.25 Overall spectra for flame radiation dependence on oxygen enrichment ratio Ω .⁵¹

spectral radiation from oxygen-enhanced/natural gas flames: (1) oxygen enrichment ratio (Ω), (2) fuel equivalence ratio (ϕ), and (3) firing rate or fuel input (q_f). The oxygen enrichment ratio ranged from $\Omega = 0.21$ (air) to $\Omega = 1.00$ (pure O_2). For the specific natural gas composition used, the stoichiometric ratio of oxygen to fuel was 2.08. Fuel-rich and fuel-lean flames had fuel equivalence ratios greater than 1.0 and less than 1.0, respectively.

As will be shown in the flame radiation spectra, most of the flame radiation fell within the infrared region. Ultraviolet (0.25 to 0.38 μm) radiation, while important in some flame diagnostics, contributed little to the total radiation intensity. Visible (0.38 to 0.78 μm) radiation also made only a small contribution to the total radiation intensity. However, as shown in [Chapter 4](#), the visible radiation is a direct indicator of the penetrating radiation that is most effective for glass heating and melting. For a relative comparison among the flames studied, the radiation spectra over the entire wavelength region (0.25 to 5.0 μm) was numerically integrated and then normalized to the integrated radiation intensity of the flame with $q_f = 5.0$ kW, $\Omega = 1.0$, and $\phi = 1.0$. This normalization eliminated possible errors due to the uncertainties in the spectral radiometer calibration. Normalization also helped reveal the general trends as the flame parameters changed. The results are summarized in [Figure 2.25](#), which shows three flame radiation spectra with $q_f = 5.0$ kW, $\phi = 1.0$, and $\Omega = 1.00, 0.60$, and 0.21 , respectively. The integrated results are listed in [Table 2.5](#). The radiation intensity from the pure oxygen/natural gas flame ($\Omega = 1.00$) was about four times that from the air/natural gas flame ($\Omega = 0.21$), for the same fuel consumption. This is because nitrogen in the air absorbs energy from the combustion reaction but N_2 does not radiate. [Figure 2.26](#) is the UV portion of the spectra plotted on a logarithmic scale. Despite the relatively small contribution of UV radiation to the total intensity, the presence of UV emissions (predominantly from the electronically excited OH radicals) can be explored as a way to visualize nonluminous flames in hot and highly luminous furnaces. The UV radiation from a pure oxygen flame was about three orders of magnitude stronger than that from the air/fuel flame. Such a dramatic difference cannot be simply accounted for by thermodynamic equilibrium at different flame temperatures. Rather, this was an indication that significantly more nonequilibrium chemiluminescence of OH radicals exists in

TABLE 2.5
Flame Radiation Intensity Measurements⁵¹

Ω	ϕ	$I_{\text{nm(total)}}$	$I_{\text{nm(pen.)}}$	$I_{\text{nm(BK7)}}$	γ
1.00	0.67	0.90 ± 0.06	0.89 ± 0.06	0.82 ± 0.06	0.15 ± 0.01
1.00	0.80	0.99 ± 0.07	0.98 ± 0.07	0.95 ± 0.07	0.13 ± 0.01
1.00	1.00	1.00 ^a	1.00 ^a	1.00 ^a	0.14 ± 0.01
1.00	1.33	0.93 ± 0.07	0.98 ± 0.07	0.99 ± 0.07	0.17 ± 0.01
1.00	1.50	0.91 ± 0.06	0.93 ± 0.07	0.87 ± 0.06	0.20 ± 0.01
1.00	1.60	0.84 ± 0.06	0.83 ± 0.06	0.88 ± 0.06	0.21 ± 0.01
1.00	1.79	0.80 ± 0.06	0.83 ± 0.06	0.83 ± 0.06	0.37 ± 0.03
1.00	1.89	0.75 ± 0.05	0.87 ± 0.06	0.96 ± 0.07	0.55 ± 0.04
1.00	2.00	0.80 ± 0.06	1.12 ± 0.08	1.10 ± 0.08	1.02 ± 0.07
0.60	1.00	0.73 ± 0.05	0.70 ± 0.05	0.75 ± 0.05	0.12 ± 0.01
0.21	1.00	0.26 ± 0.02	0.19 ± 0.01	0.19 ± 0.01	0.005 ± 0.001

The $\Omega = 1.0$, and $\phi = 1.0$ intensities were used as the reference in the normalization.

Courtesy of GRI Press.

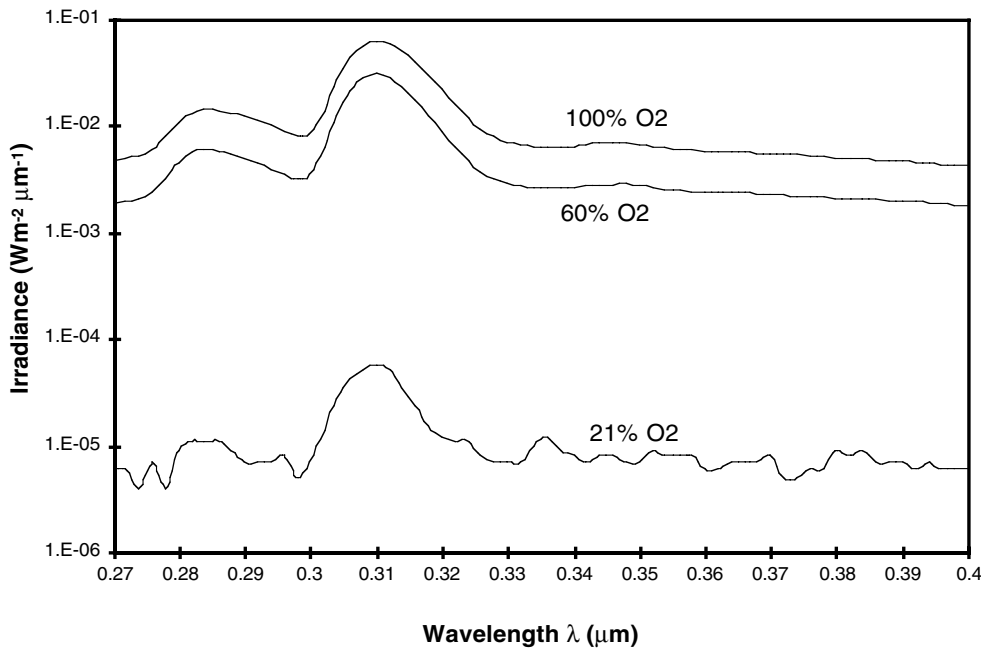


FIGURE 2.26 UV region of the spectra on a logarithmic scale for the flame radiation dependence on oxygen enrichment ratio Ω .⁵¹

oxygen-enhanced flames. It can be inferred that the OH emission-based band-reversal method, while successfully performed in $\text{CH}_4/\text{N}_2\text{O}$ flames,⁵⁴ will not yield the correct flame temperature in oxygen-enhanced flames. Figure 2.27 shows three flame radiation spectra with $q_f = 5.0 \text{ kW}$, $\Omega = 1.0$, and $\phi = 0.67, 1.00$, and 2.00 , respectively. The normalized total radiation intensities for ϕ in the range of 0.67 to 2.00 are summarized in Table 2.7. The reduced radiation intensities in the fuel-lean cases ($\phi < 1.0$) are due to the fact that excessive oxygen carries away heat but does not radiate. Although the radiation intensities in the fuel-rich cases ($\phi > 1.0$) are apparently less than that in the stoichiometric case ($\phi = 1.00$), it should be noted that not all of the fuel was consumed, due

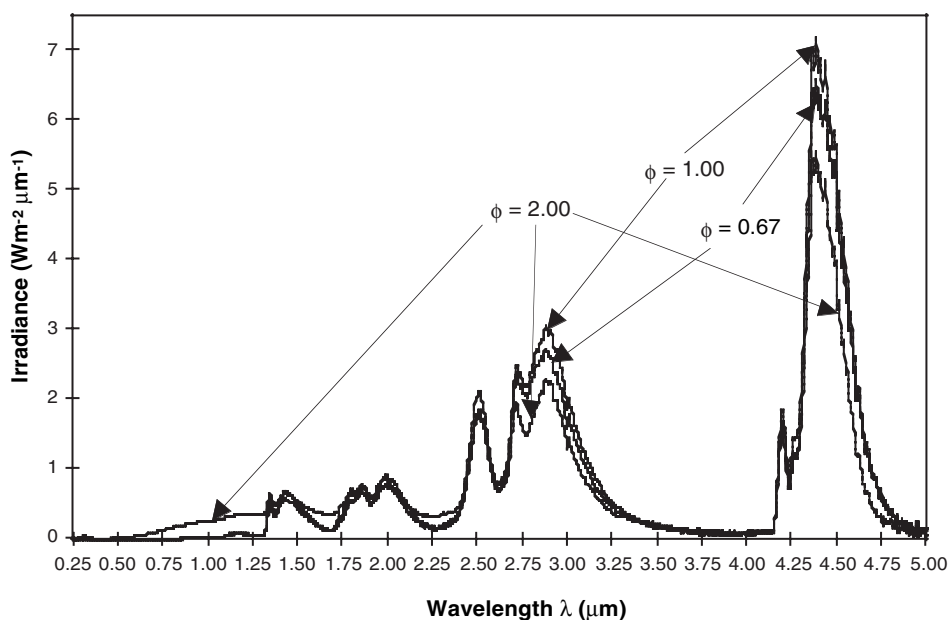


FIGURE 2.27 Flame radiation dependence on fuel equivalence ratio ϕ . Spectra were plotted with $q_f = 5.0$ kW, $\Omega = 1.0$, and $\phi = 0.67, 1.00$, and 2.00 , respectively.⁵¹

to lack of available oxygen. In a more realistic combustion system, oxygen would be introduced in a later stage to complete the combustion. With this in mind, the fuel-rich cases would actually provide more radiation for the same fuel consumption. Because the flames were in open air, actual equivalence ratios were different from the control settings due to air entrainment. Nevertheless, the general trends remain the same.

The integrated spectral radiation intensities were nearly the same for stoichiometric flames with $\Omega = 1.00$, $q_f = 5$ kW and $\Omega = 0.60$, $q_f = 7$ kW. For a given O_2 enrichment level, the radiation intensity increased more slowly than the increase in firing rate. Figure 2.28 shows radiation spectra from stoichiometric flames with different oxygen enrichment ratios and firing rates. The radiation intensity did not increase for air-fired flames ($\Omega = 0.21$) when the firing rate increased from 5 to 10 kW. These trends for the change in radiation as a function of the firing rate are consistent with the findings in another study.⁵³ While a quantitative comparison may depend on the specific burner design, the qualitative trend is generally true. Nitrogen in the air reduces radiation intensity. Because increasing the firing rate in air/natural gas flames is accompanied by increasing nitrogen flow, the increase in combustion heat cannot be as efficiently released by radiation as in the oxygen-enriched flames.

2.2.3 LUMINOUS RADIATION

2.2.3.1 Theory

Luminous flames are produced by the continuous radiant emission of particles in the flame, such as soot, that radiate approximately as blackbodies. Yagi and Iino (1961) studied both the luminous (q_{rs} = radiation from soot) and nonluminous (q_{rg} = gaseous radiation) from turbulent diffusion flames.⁵⁵ A comparison of the two types of radiation is shown in Figure 2.29, which shows that the soot radiation is greater than the gaseous radiation. Echigo et al. (1967) studied luminous radiation from flames.⁵⁶ They noted that a rigorous definition of luminous and nonluminous flames does not exist. It had long been assumed that soot remains in the solid phase during the combustion process and emits a continuous spectrum of visible and infrared radiation. Based on experiments,

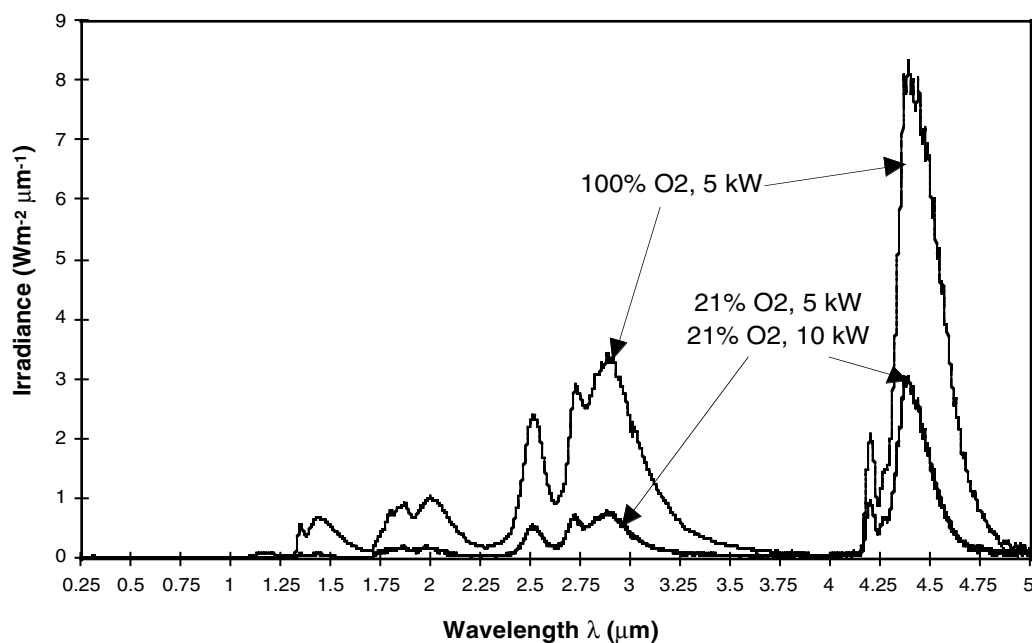


FIGURE 2.28 Flame radiation dependence on firing rate and oxygen enrichment ratio.⁵¹

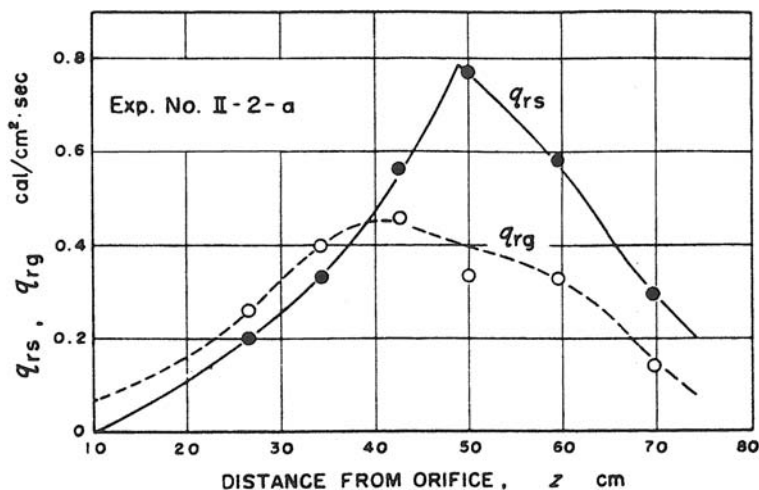


FIGURE 2.29 Soot (q_{rs}) and gaseous (q_{rg}) radiation from a turbulent diffusion flame with a fuel consisting of 11.9% CH_4 , 7.5% C_3H_8 , 22.5% C_3H_6 , 2.8% heavy hydrocarbons, 8.4% CO , 18.9% H_2 , 3.5% O_2 , 2.1% CO_2 , 22.4% N_2 , and a nozzle diameter of 7 mm. (Courtesy of The Combustion Institute.⁵⁵)

they hypothesized that the dehydrogenation and polymerization of hydrocarbon fuels occurs in the liquid phase, and that the decomposed and polymerized compound (“presoot substance”) emits banded spectra. Then the soot particles agglomerate after dehydrogenation and polymerization are complete. Gray et al. (1976) noted three size categories for particles: large, small, and intermediate.⁵⁷ Particles absorb, diffract, or attenuate radiation to varying degrees, depending on their size. The soot generated in a flame is highly dependent, among other things, on the fuel composition. Luminous radiation is usually important when liquid and solid fuels (e.g., oil and coal) are used.

TABLE 2.6
Sooting Tendency of Common Gaseous Fuels¹

Fuel	Formula	C	H	C/H Mass Ratio
Hydrogen	H ₂	0	2	0.00
Methane	CH ₄	1	4	2.97
Ethane	C ₂ H ₆	2	6	3.96
Propane	C ₃ H ₈	3	8	4.46
Butane	C ₄ H ₁₀	4	10	4.76
Propylene	C ₃ H ₆	3	6	5.95
Acetylene	C ₂ H ₂	2	2	11.89
Carbon monoxide	CO	1	0	∞

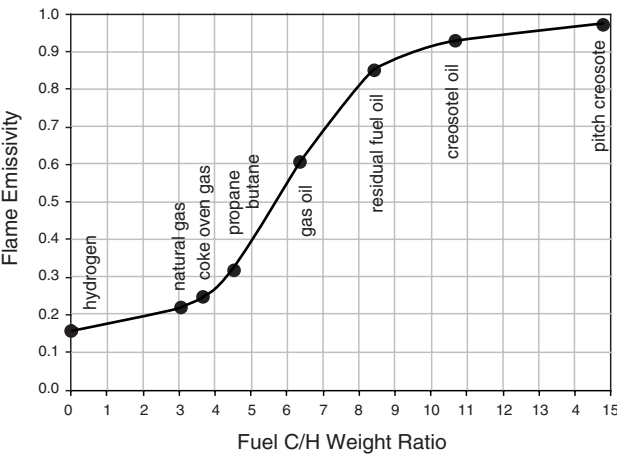


FIGURE 2.30 Emissivities of industrial-scale, turbulent diffusion flames of various fuels. (Courtesy of Gulf Publishing.⁶⁰)

It is usually not significant for gaseous fuels (e.g., natural gas). Fuels with higher carbon-to-hydrogen (C:H) weight ratios (see Table 2.6) tend to produce sootier flames. In a survey article, Wagner (1978) reviews soot formation in combustion.⁵⁸ It is noted that soot forms at temperatures ranging from 1000 to 2500°C (1800 to 4500°F). Soot consists primarily of carbon, formed into long chains. The total amount of soot formed is usually small in comparison to the amount of available carbon. Tien and Lee (1982) reviewed some of the various models available for calculating the emissivity of luminous flames, including the homogeneous nongray model, the homogeneous gray model, and the nonhomogeneous nongray model.³⁸ Glassman (1988) discussed the detailed chemistry of soot formation as it relates to fuel composition.⁵⁹ The graph in Figure 2.30 shows how the flame emissivity varies with fuel type.⁶⁰ For fuel gases with C:H weight ratios between 3.5 and 5.0, the data were correlated by either of two correlations:

$$\epsilon = 0.20 \sqrt{\frac{\text{LHV}}{900}} \tag{2.35}$$

where LHV is the lower heating value of the fuel in Btu/ft³, or

$$\varepsilon = 0.048\sqrt{MW_{\text{fuel}}} \quad (2.36)$$

where MW_{fuel} is the molecular weight of the fuel. For liquid fuels with C:H ratios between 5 and 15, the following correlation was determined:

$$\varepsilon = 1 - 68.2e^{-2.1\sqrt{C/H}} \quad (2.37)$$

where C/H is the weight ratio of carbon to hydrogen for the fuel. Haynes (1991) reviewed soot formation in flames, but did not specifically discuss the effects on heat transfer.⁶¹

The laminar smoke point for a fuel is another indicator of flame luminosity.⁶² Fuels with lower smoke point heights produce more luminous flames than high smoke point fuels. The laminar smoke point is determined by measuring the distance from the burner outlet where a vertical, laminar jet of burning pure fuel just begins to produce smoke. The lower the height where smoke starts to form, the higher tendency the fuel has to produce soot.

Lahaye and Prado (1983) have edited an extensive book on soot generated in combustion.⁶³ The focus of the book is on the chemistry of soot formation and destruction, with no real consideration of the heat transfer from luminous flames. Propane and acetylene have been two commonly used fuels to study soot. Longwell reports that soot in the form of polycyclic aromatic hydrocarbons (PAHs) can come from the fuel, fuel nonflame pyrolysis, or from quenching fuel-rich mixture regions in the flame.⁶⁴ Howard and Bittner (1983) note the formation of high-molecular-weight species prior to and during the appearance and growth of soot.⁶⁶ Bittner et al. (1983) note that soot formation is strongly related to fuel composition and that aromatic fuels have a much higher propensity to soot.⁶⁶ Prado et al. (1983) discuss soot inception and growth through nucleation and agglomeration.⁶⁷ Calcote (1983) notes the steps involved in soot formation:⁶⁸

1. Formation of precursors
2. Nucleation, which transforms molecules into particles
3. Soot particle growth, which increases the molecular weight
4. Coagulation of soot particles to form single, larger particles where the identity of the original colliding particles has been lost
5. Agglomeration of particles that adhere to each other to form a chain
6. Aggregation of colliding particles to form a cluster of individual particles that are still individually distinguishable
7. Oxidation of particles in any of the above steps to reduce the particle size and C:H ratio

Blokh (1988) has a lengthy treatment of luminous flame radiation produced by the burning of pulverized coal, oil, natural gas, and combinations of these.⁶⁹

Chemiluminescence is light emission produced when an atom or molecule is elevated to an electronically excited state in a chemical reaction. Stambuleanu (1976) notes that visible luminosity in flames is often actually chemiluminescence, and not luminosity at all.⁷⁰

Blokh (1974) discusses theoretical and experimental research on radiative properties of soot particles in luminous flames and coal particles in pulverized coal flames.⁷⁰ Kunitomo (1974) developed a method for calculating luminous flame radiation from a liquid fuel at pressures up to 20 atm (20 bar).⁷² Hammond and Beér (1974) made spectroradiometric measurements in sooty oil flames to determine spectral attenuation coefficients.⁷³ Ku and Shim (1990) developed models to predict the radiative properties of soot particles in flames.⁷⁴ To calculate luminous radiation from soot particles, the extinction coefficient, single-scattering albedo, and phase function are needed in the visible and near-infrared wavelengths. The formation processes and physical properties of soot,

light scattering and extinction by small particles, the effect of complex refractive index, the effects of particle size distribution, and the effect of agglomeration are all discussed. In addition to the discussions of models available and suggested refinements for soot particle properties, diagnostic techniques for making these measurements are also discussed. Bockhorn (1994) edited a book on soot formation in combustion, including several sections on modeling, although nothing specifically on industrial combustion.⁷⁵ Fridman et al. (1997) showed that flame luminosity can be enhanced by adding pyrene ($C_{16}H_{10}$).⁷⁶ Farias et al. (1998) computed the soot radiation properties of phase function, albedo, extinction coefficient, and emissivity using the integral equation formulation for scattering (IEFS).⁷⁷

The properties of a luminous gas can be expressed as:^{78,79}

$$\epsilon_{\lambda} = 1 - \exp(-\kappa_L L) \quad (2.38)$$

where ϵ_L is the monochromatic emittance of the luminous gas, κ_L is the absorption coefficient of the luminous gas, and L is the equivalent length of the radiating system. Hottel has recommended the following forms for the absorption coefficient:⁸⁰

$$\kappa_L = \frac{Ck_1}{\lambda^{0.95}} \quad (2.39a)$$

in the infrared region (0.8 to 10 μm), or

$$\kappa_L = \frac{Ck_2}{\lambda^{1.39}} \quad (2.39b)$$

in the visible region (0.3 to 0.8 μm), where C is the soot concentration and k_1 and k_2 are constants specific to the flame under investigation. The need for empirically determined constants is undesirable because each flame must be tested prior to analysis of a given system.¹⁰

2.2.3.2 Combustion Studies

This section provides a sampling of combustion studies concerning luminous radiation. Leblanc and Goracci (1973) note the inverse relationship between flame luminosity and flame temperatures.⁵ That is, if the flame is very luminous, it radiates its energy more efficiently and therefore has a lower temperature. If it is very nonluminous, then the flame temperature will generally be much higher because it does not release its energy as efficiently. Leblanc and Goracci reported flame emissivities ranging from 0.2 to 0.7. Example 2.8 shows a comparison of how much the flame temperature can change, depending on the emissivity (luminosity) of the flame.

Example 2.8

Given: Flame 1: $\epsilon_1 = 0.2$, $t_1 = 3000^\circ\text{F}$, flame 2: $\epsilon_2 = 0.7$, load: $t_L = 300^\circ\text{F}$.

Find: Find temperature of flame 2 (t_2), assuming same total radiant outputs.

Solution: Because total radiant outputs are the same,

$$\epsilon_1(T_1^4 - T_L^4) = \epsilon_2(T_2^4 - T_L^4)$$

or,

$$T_2 = \left[\frac{\epsilon_1}{\epsilon_2}(T_1^4 - T_L^4) + T_L^4 \right]^{1/4}$$

$$T_2 = 2533^\circ\text{R}, \text{ or } t_2 = 2073^\circ\text{F}.$$

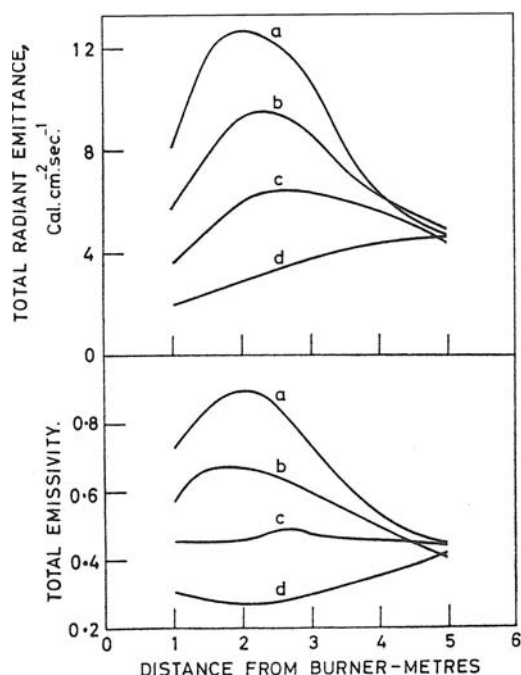


FIGURE 2.31 Effect of fuel composition and distance from an industrial burner for the following fuels: (a) 100% oil, (b) 40% oil/60% coke oven gas, (c) 20% oil/80% coke oven gas, and (d) 100% coke oven gas. (Courtesy of The Combustion Institute.)⁸¹

The flame temperature of the luminous flame is nearly 1000°F (560°C) lower than for the non-luminous flame. Although the assumptions made in the above example would not be strictly true for most flames, it shows how the flame emissivity affects the flame temperature.

2.2.3.2.1 Total Radiation

Beér and Howarth (1968) presented radiation measurements from industrial flames produced by fuels having a variable composition of oil and coke oven gas as shown in Figure 2.31.⁸¹ As expected, the more oil in the flames, the more radiation from the flame. Maesawa et al. (1968) reported radiation measurements from industrial residual oil flames in a vertical cylindrical furnace.⁸² As shown in Figure 2.32, the peak emissivity and radiant heat flux occurred at about 1 m (3 ft) from the burner in a 3-m (10-ft)-long furnace. Gill et al. (1968) measured and predicted the radiant heat transfer from pulverized coal flames in a 6-m (20-ft)-long $\times$ 1-m (3-ft)-wide rectangular, water-cooled furnace.⁸³ The peak heat flux occurred at approximately the midpoint of the furnace length. You (1985) measured the radiation from pure diffusion ($\phi = \infty$), natural gas flames.¹⁰⁷ Much of this radiation may have been luminous, because these are very fuel-rich flames. Radiation, to the stagnation point, was 13 to 26% of the total heat flux. Radiation was negligible at the edge of the plate ($R = 7.3$). Hustad calculated the radiant flux to be 7 to 14% and 20 to 40% of the total flux for CH_4 and C_3H_8 pure diffusion flames ($\phi = \infty$), respectively.¹²¹ Hustad assumed that these long flames were optically thick radiating cylinders. The convective flux was calculated using correlations for flow over a cylinder. This was subtracted from the measured total flux to determine a calculated radiant flux. This calculated radiation compared favorably with the measured radiation. The following empirical correlation was derived:

$$q''_{b,\text{rad}} = 5 + 3.9(l_j - x_{\text{liftoff}}) \quad (2.40)$$

where $q''_{b,\text{rad}}$ is in kW/m^2 and x_{liftoff} is the axial distance from the nozzle exit to the start of the flame. Luminous radiation was not considered in other studies, where the flames were very fuel rich (e.g., see Reference 112), although it may have been important.

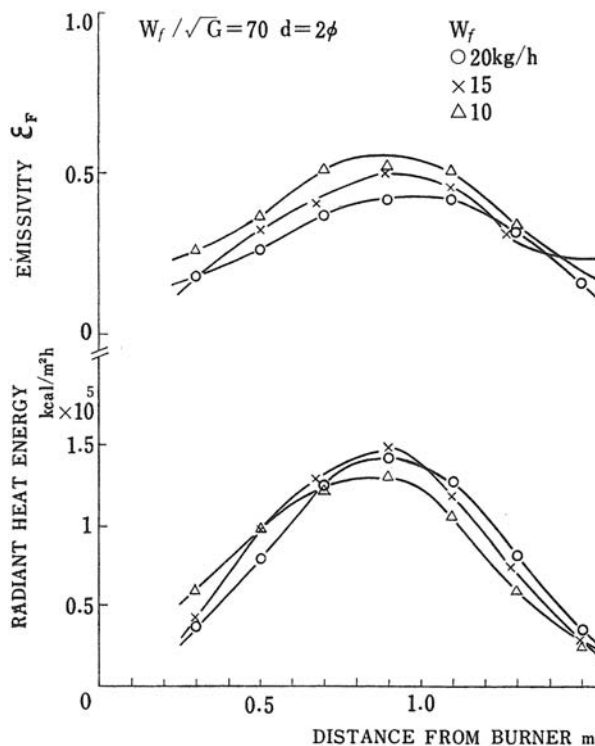


FIGURE 2.32 Measured radiant heat flux and calculated flame emissivity as a function of the distance from the burner for a residual oil flame in a vertical cylindrical furnace. (Courtesy of The Combustion Institute.⁸²)

2.2.3.2.2 Spectral Radiation

The Ji and Baukal (1998) study discussed above also investigated luminous flame radiation.⁵¹ The increased penetrating radiation in the extreme fuel-rich case ($\phi = 2.00$) reported above comes from the soot radiation. Incomplete combustion causes the fuel to crack, which forms soot particles at high temperatures. These hot soot particles emit blackbody-type radiation. The total radiation in this case is composed of discrete molecular band spectra (from hot combustion products) superimposed onto a blackbody continuum (from soot particles). The continuum radiation in the $\phi = 2.00$ flame spectra can be accounted for by a blackbody at 2318 K (3713°F), which has peak radiation at 1.25 μm . Figure 2.33 decomposes the $\phi = 2.00$ flame spectra into its soot blackbody continuum and its molecular band spectra components. The discrete band spectra are dominated by the hot combustion products: H_2O bands at 1.14, 1.38, 1.87, and 2.7 μm , and CO_2 bands at 2.7 and 4.3 μm . The CH_4 fuel molecules (2.3 and 3.3 μm bands) and combustion intermediate CO molecules (4.7 μm band) do not make a significant contribution to the flame radiation.

For quantitative characterization of the relative contributions from soot and molecular radiation, a soot radiation index γ was defined as:

$$\gamma = \frac{I_{(\text{soot at } 1.250 \mu\text{m})}}{I_{(\text{molecule at } 1.346 \mu\text{m})}} \quad (2.41)$$

Because the soot blackbody radiation peaks at 1.250 μm and there is little molecular radiation at this wavelength, it was therefore assumed that:

$$I_{(\text{soot at } 1.250 \mu\text{m})} = I_{(\text{total at } 1.250 \mu\text{m})} \quad (2.42)$$

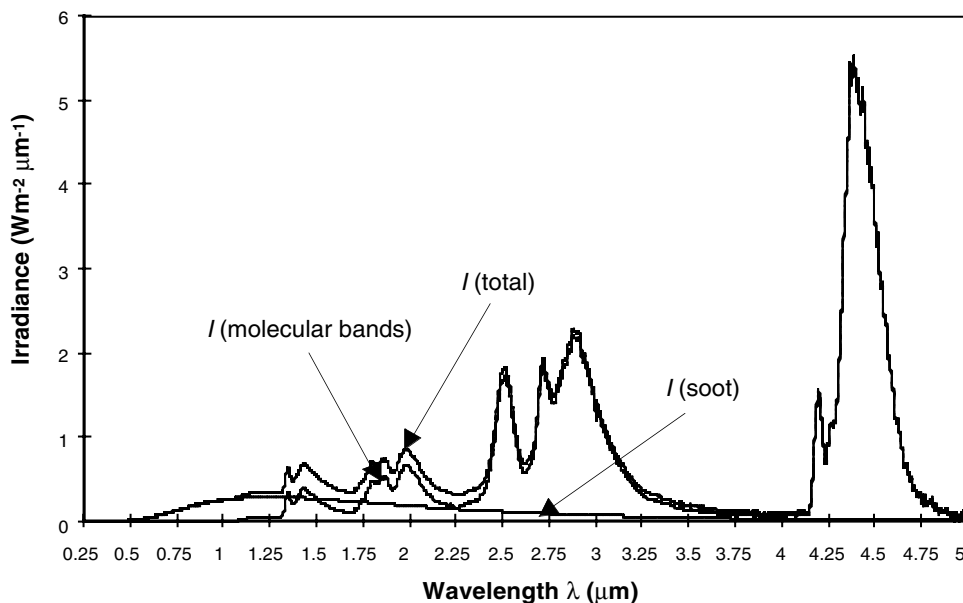


FIGURE 2.33 Soot radiation from a fuel-rich flame ($\phi = 2.00$). (Courtesy of The Combustion Institute.)⁵¹

The H_2O molecular radiation has a sharp rising (band head) peak at $1.346 \mu\text{m}$; therefore, it can be approximated that:

$$I_{(\text{molecule at } 1.346 \mu\text{m})} = I_{(\text{total at } 1.346 \mu\text{m})} - I_{(\text{soot at } 1.250 \mu\text{m})} \quad (2.43)$$

And, because the Monolight spectrometer had a nearly flat response sensitivity over the narrow wavelength range from 1.250 to $1.346 \mu\text{m}$, the value of soot radiation index γ reported here is independent of the spectrometer calibration. Table 2.5 lists the soot radiation index γ for $q_f = 5.0$ kW flames.

Figure 2.34 shows the normalized total radiation intensity $I_{\text{nm}(\text{total})}$, the normalized penetrating radiation intensity $I_{\text{nm}(\text{pen.})}$, and the soot radiation index γ as a function of the fuel equivalence ratio (ϕ) for $\Omega = 1.00$, $q_f = 5.0$ kW flames. As ϕ increased to fuel rich, both $I_{\text{nm}(\text{total})}$ and $I_{\text{nm}(\text{pen.})}$ decreased first due to incomplete combustion. At $\phi = 1.79$, $I_{\text{nm}(\text{pen.})}$ started to increase due to increased soot radiation. This increase in $I_{\text{nm}(\text{pen.})}$ corresponds to a rapid increase in the soot radiation index γ . Finally, at $\phi = 2.00$, $I_{\text{nm}(\text{total})}$ also increased because the much-increased soot radiation became a significant part of the total radiation. The soot radiation index γ correlates closely to the soot concentration in the flame. A new, commercial, high-radiation burner⁸⁴ has a soot index close to 4 compared to the highest value in this study, which was slightly greater than 1 (see Table 2.5).

2.3 CONVECTION

Convection heat transfer is caused by fluid motion past a material, where the fluid is either at a higher or lower temperature than the material. In industrial combustion applications, the fluid is usually at a higher temperature than the medium it is heating. At least one person has argued that convection is not actually a separate mode of heat transfer, but that it is a subset of conduction, because the energy must still conduct from the fluid to the material.⁸⁵ While that may be true on a microscopic scale in the boundary layer next to the material, convection is a fundamentally different process than conduction and is treated here as such, which is the convention in standard heat transfer texts. Forced convection is often a very important mode of heat transfer in industrial combustion

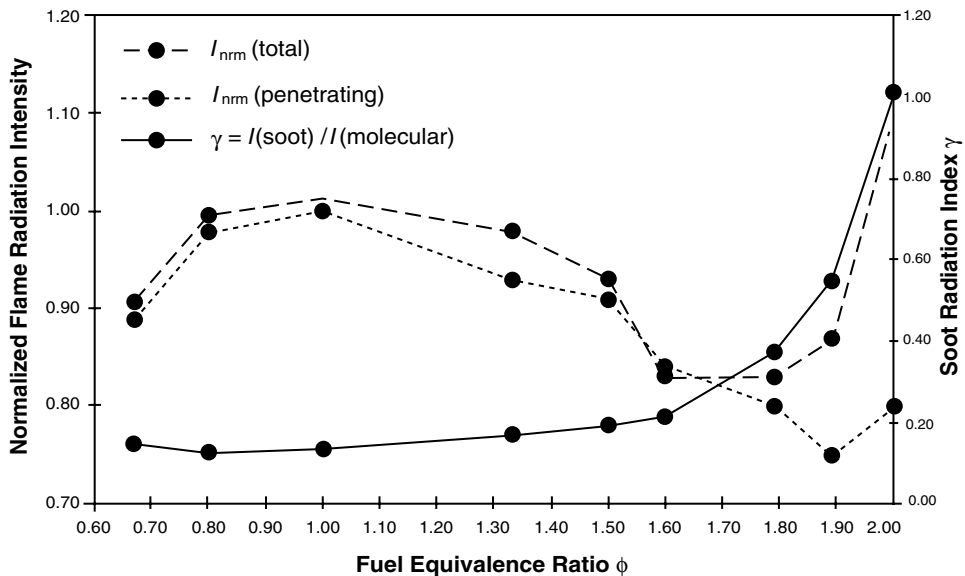


FIGURE 2.34 Normalized total radiation intensity $I_{nrm(\text{total})}$, normalized penetrating radiation intensity $I_{nrm(\text{pen.})}$, and soot radiation index γ as a function of fuel equivalence ratio ϕ for $\Omega = 1.00$ flames.⁵¹

systems. In some limited applications, natural convection may also be important because of the high temperature gradients that may exist. However, natural convection is not typically a significant mechanism in the majority of industrial combustion processes and is therefore not considered here. A number of books are available that specifically address convection heat transfer.⁸⁶⁻⁹⁴

2.3.1 FORCED CONVECTION

Forced convection heat transfer occurs when a fluid is forcefully directed at or over a medium (liquid or solid) and can be simply calculated using:

$$q = hA(t_f - t_m) \quad (2.44)$$

where q is the heat flux to the medium (Btu/hr or kW), h is the convective heat transfer coefficient (Btu/ft²-hr-°F or W/m²-K), A is the surface area of the medium in contact with the moving fluid, t_f is the fluid temperature (°F or K), and t_m is the temperature (°F or K) of the medium. A simple example is given below:

Example 2.9

Given: $h = 10 \text{ Btu/ft}^2\text{-hr-}^\circ\text{F}$, $A = 10 \text{ ft}^2$, $t_f = 300^\circ\text{F}$ and $t_m = 70^\circ\text{F}$.

Find: q .

Solution: $q = h A (t_m - t_p) = (10 \text{ Btu/ft}^2\text{-hr-}^\circ\text{F})(10 \text{ ft}^2)(300^\circ\text{F} - 70^\circ\text{F}) = 23,000 \text{ Btu/hr}$

This is an overly simplified example, because the convection heat transfer coefficient h has been given. In most problems, this value must be calculated from an appropriate correlation that may be fairly complicated, or worse, may not exist for the exact configuration under study. In the latter case, either experiments need to be done to determine this value or an approximate value must be used. The correlations for the convection coefficient are often complicated functions that normally depend on the flow geometry, the fluid velocity, and the fluid properties. In addition, for most real problems, the fluid and medium temperatures vary by location and are not constants over the entire medium surface.

Forced convection heat transfer from a flame to the load and inner combustor walls and from the outer combustor walls to the ambient are both potentially important mechanisms. Forced convection heat transfer in industrial combustion is often more difficult to analyze and calculate because of the large temperature difference between the combustion products and the surfaces (furnace walls and heat load) and because the gas properties vary widely as a function of both temperature and composition.

2.3.1.1 Forced Convection from Flames

In many conventional furnace heating processes, forced convection is only a small fraction of the total heat transfer to the product. Most of the heating comes from the radiation from the hot refractory walls. However, in flame impingement, with no furnace enclosure, forced convection can be 70 to 90% of the total heat flux.^{95,96} For flame temperatures up to about 2600°F (1700K), forced convection is the dominant mechanism in flame impingement heat transfer.⁹⁷

For low-temperature flames, as is common in air/fuel combustion systems, forced convection has generally been the only mechanism considered. In highly dissociated oxygen/fuel flames, a large fraction of the heat release is from exothermic reactions. However, even for those flames, forced convection is still an important contributor to the overall heat transfer to the target.

The turbulence level directly affects the importance of forced convection. The flow regime is determined by the Reynolds number, defined in Equation 2.1. For example, for flame jet impingement (see Figure 2.35), there are many possible choices for the length l . One is the burner outlet nozzle diameter, d_n ; another is the axial distance from the nozzle exit to the surface being heated, l_j ; and a third possibility is the width of the jet at the edge of the stagnation zone d_j . Yet another is some dimension of the material being heated. For a disk or cylinder, it might be the diameter d_b . For a plane surface, it might be the radial distance from the stagnation point r . In one case, the width of the water cooling channel in a target used in a flame impingement study was used.⁹⁷

Laminar flames have been used in many flame impingement studies.^{97–107} Sibulkin (1952) developed a semi-analytic solution for the heat transfer for laminar flow, normal to the stagnation point of an axisymmetric, blunt-nosed target.¹⁰⁸

$$q_s'' = 0.763(\beta_s \rho_e \mu_e)^{0.5} \text{Pr}_e^{-0.6} c_{p_e} (t_e - t_w) \quad (2.45)$$

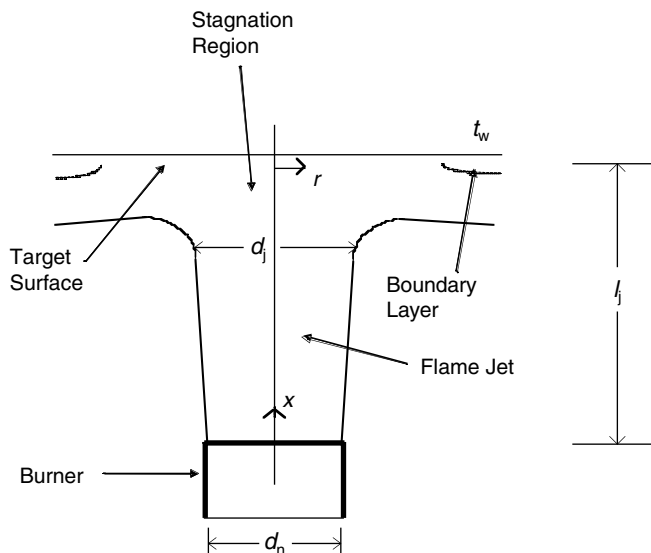


FIGURE 2.35 Flame jet impingement.¹³⁸

This result has been the basis for all other semi-analytical flame impingement heat transfer solutions.¹⁰⁹

Turbulent flames have also been commonly used.^{96,110–122} A typical example of an empirical equation, incorporating the turbulence intensity Tu , was given by Hustad and Sønju (1991)¹²¹ as:

$$q_s'' = \frac{k_e}{d_b} \left\{ 0.41 \text{Re}_{b,e}^{0.6} \text{Pr}_e^{0.35} Tu^{0.15} \left(\frac{\text{Pr}_e}{\text{Pr}_w} \right)^{0.25} \right\} (t_e - t_w) \quad (2.46)$$

This was developed for flames produced by jets of CH_4 and C_3H_8 , into ambient air. These are known as pure diffusion ($\phi = \infty$) flames. The flames impinged normal to uncooled steel pipes. These experiments were done to simulate fires caused by ruptured fuel pipes in the petrochemical industry.

Babiy (1974) presents a correlation for the convective heat transfer from combustion gases to carbon particles in pulverized coal combustion processes.¹²³ A commonly used equation for the convective heat transfer between a gas and a sphere is given by:

$$q_b'' = 31.2 \left(q_f / l_j^2 \right) \text{Ra}_e^{-1/6} \text{Pr}_e^{-3/5} \quad (2.47)$$

where d is the diameter of the sphere. For pulverized coal combustion, this equation is modified as follows:

$$q_b'' = 31.2 \left(q_f / l_j^2 \right) \text{Ra}_e^{-1/6} \text{Pr}_e^{-3/5} \quad (2.48)$$

where T_g is the absolute temperature (K) of the gas. The equation applies for $T_g = 1200$ to 1600K (1700 to 2400°F), $\text{O}_2 = 5\text{--}21\%$, $d = 150$ to $1000 \mu\text{m}$, and $\text{Re}_d < 1$.

2.3.1.2 Forced Convection from Hot Gases to Tubes

In the convection section of a heater, Monrad (1932)¹²⁴ recommended the following empirical correlation for the convective heat transfer from hot exhaust products to a bank of staggered tubes:

$$q_b'' = 31.2 \left(q_f / l_j^2 \right) \text{Ra}_e^{-1/6} \text{Pr}_e^{-3/5} \quad (2.49)$$

where h_c = convection coefficient ($\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$), G = gas mass velocity at minimum free cross-sectional area ($\text{lb/ft}^2\text{-s}$), T_g = absolute gas temperature ($^\circ\text{R}$), and d = tube diameter (in.).

2.3.2 NATURAL CONVECTION

Natural convection is sometimes referred to as buoyancy-induced flow.¹²⁵ Natural convection heat transfer from flames may be important in industrial combustion systems. This can occur when the gas velocities are very low.

One measure of the intensity of natural convection is the Rayleigh number:

$$\text{Ra} = g \beta_e q_f l_j^2 / \rho_e c_{p_e} \nu_e^3 \quad (2.50)$$

which is analogous to the Reynolds number for forced convection. Higher Rayleigh numbers indicate more natural convection. Another measure of intensity is the Grashoff number:

$$Gr = \frac{g\tilde{\beta}(t\omega - t_{\infty})\ell^3}{\nu^2} \quad (2.51)$$

where g is the gravity constant, $\tilde{\beta}$ is the volume coefficient of expansion ($=1/T$ for an ideal gas), and ν is the kinematic viscosity. The Richardson number Ri is one measure of the importance of buoyancy compared to forced convection. It is defined as:

$$Ri = \frac{Gr}{Re_n^2} \quad (2.52)$$

which is the ratio of the buoyant force to the inertial force. Conolly and Davies (1972) studied stoichiometric laminar flames impinging on a hemi-nosed cylinder.¹⁰³ The flame was parallel to the cylinder and impinging on the nose. A variety of fuels and oxidizers were used. Buoyancy effects were negligible. The criterion was that buoyancy could be neglected, compared to forced convection, for $Ri < 0.05$. Wang (1993) numerically modeled a nonreacting jet of ambient air impinging on an infinite flat plate.¹²⁶ It was concluded that natural convection is important only when Re_j is low, and that the temperature difference between the jet and the stagnation surface is large. The critical Ri value was estimated to be approximately 0.02. You (1985) determined the heat transfer from a buoyant flame to a flat plate, in terms of the Rayleigh number as:¹⁰⁷

$$q_b'' = 31.2 \left(q_f / l_j^2 \right) Ra_e^{-1/6} Pr_e^{-3/5} \quad (2.53)$$

The flames were produced by upward jets of pure natural gas ($\phi = \infty$) into ambient air. The jets impinged on a horizontal surface, to simulate a fire spreading over the ceiling of a room. Natural convection is more important in low-velocity flames. Both Beér (1968)⁹⁵ and Vizios (1971)¹¹¹ stated that the effects of buoyancy were negligible in their studies, due to the high burner exit velocities.

2.4 CONDUCTION

Thermal conduction is often overlooked when considering heat transfer in combustion systems. Although it is not an important heat transfer mode in the combustion space, it is important in determining the heat loss through the refractory walls. Conduction is important in the design of furnaces because of the thermal expansion that occurs as the furnace heats up, especially considering the difference in expansion between refractories and the metal shell.¹²⁷

Thermal conduction has played an important role in many flame impingement heating applications. In some processes, high thermal conduction rates are desired. An example is a rapid reheat furnace. There, the goal is to raise the temperature of metal parts. Because metals generally have high thermal conductivities, heat may be quickly conducted through the part. This reduces the temperature gradient between the surface and the interior of the part. High gradients may cause the part to warp or deform. In other applications, low thermal conduction is desired. An example is thermal spallation. In this process, a high-intensity flame impinges directly on a solid having a low thermal conductivity. The heat transfers slowly into the solid, due to its low conductivity. The surface is very hot. Just below the surface, the solid is near ambient temperature. This results in very large internal temperature gradients that produce high thermal stresses. These stresses cause the solid to fracture. Thermal spallation has been shown to be a cost-effective method to “drill” through rocks.¹¹⁹

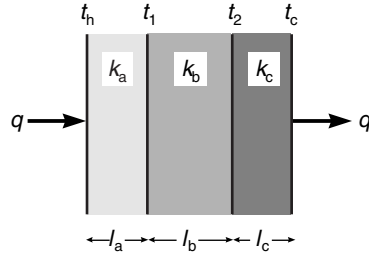


FIGURE 2.36 Thermal conduction through a composite wall.¹

In addition to transferring heat through the target, conduction has been used to measure the heat flux in flame impingement experimental studies. Both steady-state and transient methods have been used (see [Figure 7.4](#)). These are briefly discussed here. More detailed information on thermal conduction heat transfer is available in books specifically written on that subject.^{128–133}

2.4.1 STEADY-STATE CONDUCTION

Thermal conduction through refractory walls is important in determining the heat losses for a given process. The heat loss through a composite wall made of different refractory materials, shown in [Figure 2.36](#), can be calculated using:

$$q = \frac{t_h - t_c}{l_a/k_a A + l_b/k_b A + l_c/k_c A} \quad (2.54)$$

where t_h is the hotface temperature, t_c is the coldface temperature, l_i is the thickness of layer i , k_i is the thermal conductivity of layer i , and A is the cross-sectional area for conduction. Equation 2.54 assumes that the thermal conductivity of each layer does not change with temperature, which makes the analysis more complicated. It also assumes that there is no contact resistance between each layer. For example, if there is an air gap between each layer, this would give a large resistance to heat flow that should be included. The equation above is for a three-layer system, but any number of layers can be used by adding or subtracting the appropriate number of thermal conductances ($l_i/k_i A$). This equation can be used to solve for the hot or cold face temperatures, for the heat flux through the wall, or for a temperature inside the wall. It is assumed that the thicknesses (l_i), thermal conductivities (k_i), and cross-sectional area are known. Table 2.10 shows the thermal conductivity for various ceramics and other insulating materials.¹³⁴

Example 2.10

Given: Cold face temperature of 140°F, heat flux to the wall of 400 Btu/hr-ft², 8 in. of fireclay brick.

Find: Find the hot face temperature.

Solution: Assume that the conductivity through the steel shell is negligible. Note that the heat flux per unit area is given (q/A). Rearrange the equation above to solve for the hot face temperature:

$$t_h = t_c + \left(\frac{q}{A} \right) \left(\frac{l_{\text{firebrick}}}{k_{\text{firebrick}}} \right)$$

From [Table 2.7](#), $k_{\text{firebrick}} = 1.1 \text{ W/m-K} = 0.64 \text{ Btu/hr-ft-}^\circ\text{F}$,

$l_{\text{firebrick}} = 8/12 \text{ ft}$, $t_c = 140^\circ\text{F}$

solving for $t_h = 557^\circ\text{F}$

TABLE 2.7
Thermal Conductivity of Ceramics and Other Insulating Materials¹

Material	Dens. g/cm ³	<i>t</i> °C	Ther. cond, W/m K
Alumina (Al ₂ O ₃)	3.8	100	30
		400	13
		1300	6
		1800	7.4
Al ₂ O ₃ + MgO	3.5	100	17
		800	7.6
		100	15
		400	10
Asbestos	0.4	1000	5.6
		−100	0.07
		100	1
Asbestos + 85% MgO	0.3	0	0.09
Asphalt	0.3	30	0.08
Beryllia (BeO)	2.1	20	0.06
		100	210
		400	90
		1000	20
	1.85	1800	15
		50	64
		200	40
		600	23
Brick, dry	1.54	0	0.04
Brick, refractory:			
alosite		1000	1.3
aluminous	1.99	400	1.2
		1000	1.3
diatomaceous	0.77	100	0.2
		500	0.24
	0.4	100	0.08
		500	0.1
fireclay	2	400	1
		1000	1.2
silicon carbide	2	200	2
		600	2.4
vermiculite	0.77	200	0.26
		600	0.31
Calcium oxide		100	16
		400	9
		1000	7.5
Cement mortar	2	90	0.55
Charcoal	0.2	20	0.055
Coal	1.35	20	0.26
Concrete	1.6	0	0.8
Cork	0.05	0	0.03
		100	0.04
		0	0.06
		100	0.08
Cotton wool	0.08	30	0.04
Diatomite	0.2	0	0.05

(Continued)

TABLE 2.7
Thermal Conductivity of Ceramics and Other Insulating Materials (Continued)

Material	Dens. g/cm ³	<i>t</i> °C	Ther.cond, W/m K
		400	0.09
	0.5	0	0.09
		400	0.16
Ebonite	1.2	0	0.16
Felt, flax	0.2	30	0.05
	0.3	30	0.04
Fuller's earth	0.53	30	0.1
Glass wool	0.2	−200 to 20	0.005
		50	0.04
		100	0.05
		300	0.08
Graphite			
100 mesh	0.48	40	0.18
20–40 mesh	0.7	40	1.29
Linoleum cork	0.54	20	0.08
Magnesia (MgO)		100	36
		400	18
		1200	5.8
		1700	9.2
MgO + SiO ₂		100	5.3
		400	3.5
		1500	2.3
Mica:			
muscovite		100	0.72
		300	0.65
		600	0.69
phlogopite		100	0.66
Canadian		300	0.19
		600	0.2
Micanite		30	0.3
Mineral wool	0.15	30	0.04
Perlite, expanded	0.1	−200 to 20	0.002
Plastics:			
bakelite	1.3	20	1.4
celluloid	1.4	30	0.02
polystyrene foam	0.05	−200 to 20	0.033
mylar foil	0.05	−200 to 20	0.0001
nylon		−253	0.10
		−193	0.23
		25	0.30
polytetrafluoroethylene		−253	0.13
		−193	0.16
		25	0.26
		230	2.5
urethane foam	0.07	20	0.06
Porcelain		90	1
Rock:			
basalt		20	2
chalk		20	0.92

(Continued)

TABLE 2.7
Thermal Conductivity of Ceramics and Other Insulating Materials (Continued)

Material	Dens. g/cm ³	<i>t</i> °C	Ther.cond, W/m K
granite	2.8	20	2.2
limestone	2	20	1
sandstone	2.2	20	1.3
slate		95	1.4
slate		95	2.5
Rubber:			
sponge	0.2	20	0.05
92 percent		25	0.16
Sand, dry	1.5	20	0.33
Sawdust	0.2	30	0.06
Shellac		20	0.23
Silica aerogel	0.1	−200 to 20	0.003
Snow	0.25	0	0.16
Steel wool	0.1	55	0.09
Thoria (ThO ₂)		100	10
		400	5.8
		1500	2.4
Titanium dioxide		100	6.5
		400	3.8
		1200	3.3
Uranium dioxide		100	9.8
		400	5.5
		1000	3.4
Wood:			
balsa,	0.11	30	0.04
fir,	0.54	20	0.14
fir,	0.54	20	0.35
oak		20	0.16
plywood		20	0.11
pine,	0.45	60	0.11
pine,	0.45	60	0.26
walnut,	0.65	20	0.14
Wool	0.09	30	0.04
Zinc oxide		200	17
		800	5.3
Zirconia (ZrO ₂)		100	2
		400	2
		1500	2.5
Zirconia + silica		200	5.6
		600	4.6
		1500	3.7

Thermal conductivity values for ceramics, refractory oxides, and miscellaneous insulating materials are given here. The thermal conductivity refers to samples with density indicated in the second column. Since most of these materials are highly variable, the values should only be considered as a rough guide.

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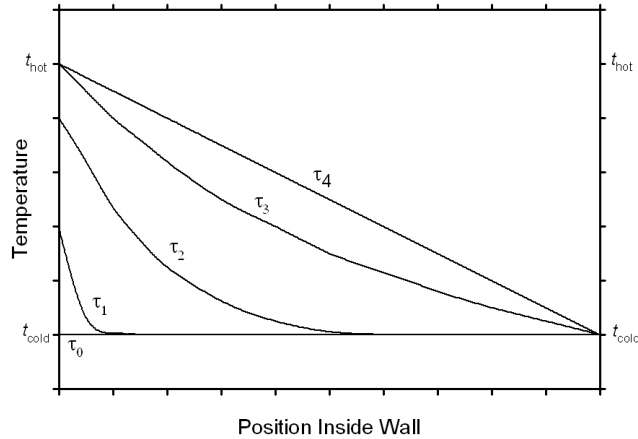


FIGURE 2.37 Transient thermal conduction through a wall.¹

Steady-state conduction has been important in flame impingement studies. Internally cooled targets have been used in many of the experimental studies.^{48,49,99,101,103,105,110–113,115,117,118,120} The flame impinges on the hot side of the target. The coolant flows past the cold side of the target. In other studies, the target has been an uncooled ceramic.^{95,97,98,102,111} In both the cooled metal and uncooled ceramic studies, the hot side and cold side temperatures were approximately constant, producing steady-state conduction through the target.

Steady-state conduction methods have been employed in the design of heat flux gages (see [Chapter 6](#)) used in flame impingement studies. These methods involve measuring the temperature gradient through a piece of metal in the gage. The gage is heated on one side by the flame. The other side is commonly maintained at a fixed temperature, using a coolant. Long, thin, solid rods;^{95,99,101,112,117,120} thin disks;¹¹⁰ and heat flux transducers^{107,113,122} have been used as gages.

2.4.2 TRANSIENT CONDUCTION

Transient conduction is an important aspect of heat transfer in industrial combustion that is often overlooked. It is significant during the start-up of a process where the heat distribution through the walls of a furnace has not reached steady-state. [Figure 2.37](#) shows how the temperature distribution through a refractory wall changes as a function of time. At t_0 , the temperature through the wall is uniform (t_{cold}), when heat is then applied to the left side of the wall. After some time (t_g), the wall temperature distribution reaches steady-state.

Another instance where transient conduction is important is during the heating of a load in a batch heating process. The load is initially cold after it is first charged into the furnace. As heat is applied to the surface, the surface temperature rises quickly, while the temperatures inside the load are still low until the heat has time to conduct through the load. The thermal conductivity of the load determines how fast the temperature profile equilibrates inside the load. For example, aluminum conducts very quickly compared to a refractory material such as bricks or ceramics. This can be an important consideration if the material conducts slowly because the surface can be overheated before the heat has a chance to conduct inside the load. This reduces the product quality and process yield.

Uncooled metal targets have been used in previous flame impingement experimental studies. The target materials have been steel,^{96,102,114,121} inconel,¹¹⁶ iron,⁴⁶ molybdenum,¹⁰⁰ brass,¹⁰⁴ and copper.¹¹⁹ These targets were exposed to the flame for only a short time. Kilham and Purvis (1971)¹⁰² and Nawaz (1973)¹⁰⁴ inserted the target into the flame for less than 1 second. Giedt and co-workers (1960, 1966) had test durations of between 20 and 50 seconds.^{46,100} The targets were instrumented

with an array of thermocouples. The heat flux to the surface was estimated, using an inverse transient heat conduction calculation. This method is discussed by Gebhart (1993).¹³²

Transient conduction has also been used in the design of slug calorimeter heat flux gages, as used in previous studies.^{102–104,116,118,120,121} Heat is received on a slug of metal inside the calorimeter. The slug is assumed to be internally isothermal at any instant in time. The heat flux is calculated from the rise in the thermal capacitance of the slug, as a function of time.

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