

Section II

Burner Fundamentals

6 Heat Transfer from Burners

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6.1 INTRODUCTION

The primary purpose of a burner is to transfer heat from the combustion products to some type of load. Therefore, it is important to match the burner to the load to ensure heat transfer is maximized in order to maximize the system thermal efficiency. This reduces operating costs and indirectly reduces pollutant emissions because less fuel needs to be combusted for a given production rate. The heat transfer from burners is often a complicated process because of the turbulent fluid flow, high-temperature chemical reactions, and spectral gaseous radiation.

The purpose of this chapter is to consider the heat transfer from different types of burners, without much consideration for the applications of the burners that are discussed in other chapters as they pertain to different burner types. The emphasis here is to give the reader an idea of the general behavior of different types of burners. This information should be useful when considering new applications or reevaluating existing technologies. Chapter 2 contains a general discussion of the modes of heat transfer. Much of this chapter was taken from Baukal (2000).¹

6.2 OPEN-FLAME BURNERS

Open-flame burners are defined here as ones where the flame is not confined, as is the case in radiant tubes (see Chapter 13), nor are the flames primarily attached to a surface as in porous refractory burners (see Chapter 14). Open-flame burners are normally visible to the naked eye where the radiant heat from the flame, rather than from a surface heated by the flame, can directly heat the load. Some of the important factors that affect the heat transfer from open-flame burners are considered in this section.

6.2.1 MOMENTUM EFFECTS

There are two aspects to the momentum effects on flames. The first involves the forward momentum normally associated with the average outlet velocity of the combustion products. The second aspect is the lateral momentum caused by swirl. The swirl number is defined as the ratio of the lateral momentum to the forward momentum. Burners with no swirl have a swirl number of zero. Beér and Chigier (1972) defined weak swirl as a swirl number below 0.6 and strong swirl as a swirl number greater than 0.6.⁹ Villasenor and Escalera (1998) studied swirl effects on heavy fuel oil combustion in an air-cooled, high-temperature research furnace.² The experimental results showed that there was a strong dependence between the incident radiation flux and the swirl number. Burners with intermediate swirl numbers (0.1 and 0.4) produced more uniform heat flux profiles than burners with either no swirl, or with higher swirls (0.75 and 1.0). As expected, the burner swirl has an impact on the heat flux distribution from the flame.

6.2.2 FLAME LUMINOSITY

An example of a high-luminosity flame is shown in Figure 6.1.³ An example of a low-luminosity flame is shown in Figure 6.2. The flame luminosity is a function of many variables but is especially dependent on the fuel. Solid and liquid fuels tend to make more luminous flames than gaseous fuels because of particles in the flame that radiate like graybodies. A recent trend in the glass industry has been to make more luminous flames, with natural gas as the fuel, to improve the thermal efficiency of the glass melting process.⁴ The burner design also plays a large role in how luminous the flame will be and how heat is transferred from the flame to the load.⁵

The radiant energy from the flame can be approximated by:⁶

$$q_{\text{rad}} \propto a_p V_f T_f^4 \quad (6.1)$$

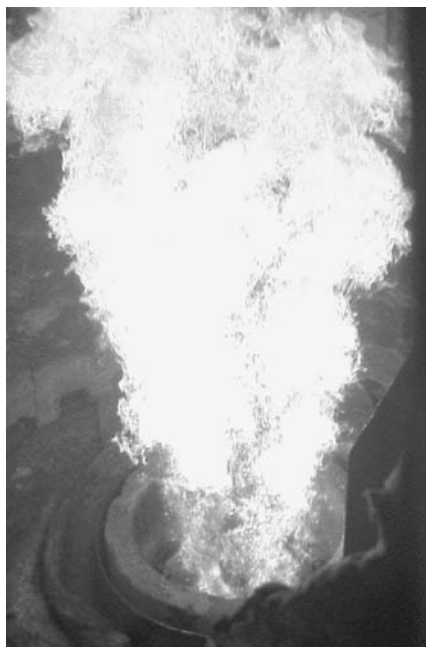


FIGURE 6.1 Example of a high-luminosity oil flame. (Courtesy of John Zink Company, LLC.³)

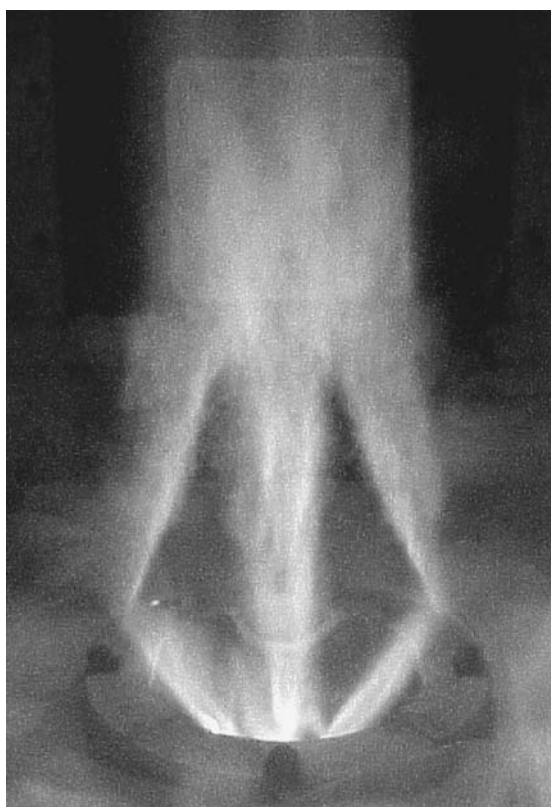


FIGURE 6.2 Example of a low-luminosity gas flame. (Courtesy of John Zink Company, LLC.³)

where a_p is the Planck-mean absorption coefficient for an optically thin flame, V_f is the flame volume, and T_f is the absolute temperature of the flame. The total heat release by combustion is:

$$q_{\text{total}} = \dot{m}_{\text{fuel}} \Delta H_c \quad (6.2)$$

where $\dot{m}_{\text{fuel}}$ is the fuel mass flow rate and ΔH_c is the heat of combustion. The radiant fraction was then defined as the ratio of the radiant heat transfer to the total heat released by combustion:

$$\chi_{\text{rad}} = q_{\text{rad}} / q_{\text{total}} \approx a_p T_f^4 d / u \quad (6.3)$$

where d is the diameter of the nozzle outlet diameter, u is the outlet velocity, and the flame volume was assumed proportional to d^3 and the fuel flow rate to $d^2 u$. Turns and Myhr (1991) studied the interaction between flame radiation and NOx emissions from turbulent jet flames.¹⁶ Radiant fractions were calculated from radiant heat flux measurements made with a transducer having a 150° view angle with a window for free jet flames of C₂H₄, C₃H₈, CH₄, and a blend of 57% CO/43% H₂. A global residence time was defined as:

$$\tau_g = \frac{\rho_f w_f^2 l_f f_s}{3 \rho_0 d^2 u} \quad (6.4)$$

where ρ_f is the flame density, w_f is the flame width, l_f is the flame length, f_s is the fuel mass fraction, and ρ_0 is the cold fuel density. Figure 6.3 shows a plot of the calculated radiant fraction as a function of the global residence time for all four fuels. At small residence times, the radiant fraction is somewhat independent of the fuel composition because the flames are momentum-dominated and nonluminous. As the residence time increases, the radiant fraction becomes dependent on the sooting tendency of the fuel.

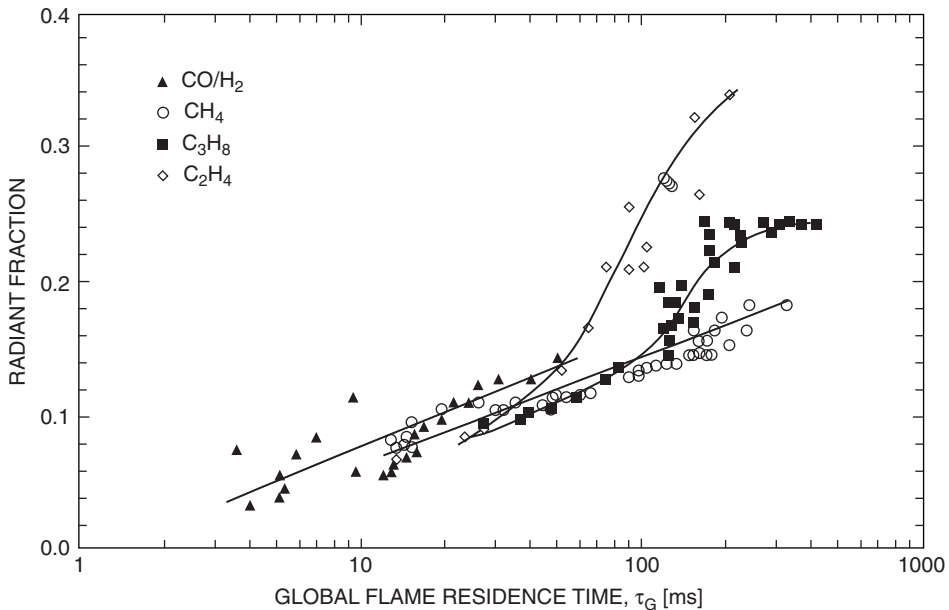


FIGURE 6.3 Flame radiant fractions as functions of global residence time. (Courtesy of The Combustion Institute.¹⁶)

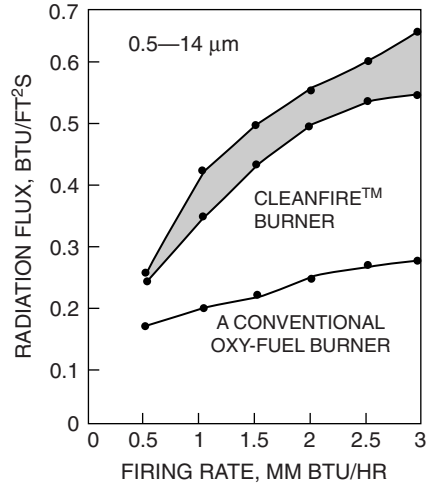


FIGURE 6.4 Comparison of measured flame radiation for a new style oxy/fuel burner Cleanfire™ Burner and a conventional oxy/fuel burner.⁷

Slavejkov et al. (1993) showed that a new style oxy/fuel burner can produce significantly more radiant flux than conventional, older-style designs.⁷ A comparison of the radiant flux as a function of firing rate is shown in Figure 6.4. In some cases, the measured radiant flux was more than doubled using the new burner design.

6.2.3 FIRING RATE EFFECTS

The main concern in industrial combustors is normally to maintain a certain temperature profile in the material being heated, which often equates to a specific temperature profile inside the combustor for a given burner type. It is often necessary to adjust the firing rate to meet the needs of a given application. For example, for an existing combustion system, it may be desirable to increase the material processing rate, which normally means that the firing rate must be increased. The design question to be answered is by how much, because this may or may not be a linear relationship. Another example is the modification of a well-known system design for higher or lower throughput rates. Again, the question is how to do the scaling from the known design.

There are many possible ways to scale a burner according to changes in the firing rate, which is the primary characteristic of interest in most industrial applications. Spalding (1963),⁸ Beér and Chigier (1972),⁹ and Damköhler (1936)¹⁰ looked at numerous dimensionless groups based on considerations of the momentum, energy, and mass balances. Some of these groups include the Reynolds, Froude, and Damköhler numbers. However, it is not possible to maintain all of the dimensionless groups constant simultaneously.

The two most common methods used to scale industrial burners are constant velocity and constant residence time. Constant velocity scaling is, by far, the most popular. The burner thermal input (Q) can be calculated using:

$$Q_0 = K\rho_0 u_0 d_0^2 \quad (6.5)$$

where ρ_0 , u_0 , and d_0 are the inlet average fluid density, characteristic fluid velocity, and characteristic diameter, respectively, and K is a proportionality constant. Assuming that the inlet fluid density is constant, and the characteristic inlet fluid velocity u_0 is held constant, then the constant velocity scaling law can be written as:

$$d_1 \propto (Q_1)^{0.5} \quad (6.6)$$

where the new characteristic burner diameter d_1 is proportional to the square root of the new firing rate. This law simply says that the outlet area of the burner is directly proportional to the firing rate for a constant-velocity scaling law. For round outlets, the area is proportional to the square of the diameter.

The principle for constant residence-time scaling is to maintain the ratio of d_0/u_0 , which has the units of time (typically seconds). This scaling law, sometimes known as the inertial or convective time scale, can then be written as:

$$d_1 \propto (Q_1)^{0.33} \quad (6.7)$$

This approach is not as commonly used because it leads to very low windbox pressures for smaller burners and excessive pressures for larger burners.¹¹

The heat transfer from the flame to the furnace and load is often dominated by radiation close to the burner outlet. Further from the burner outlet, radiation and convection may both be important. Another approach to scaling is based on the radiative characteristics of the flame. Markstein (1976) developed a correlation for buoyancy-dominated turbulent flames.¹² Other correlations have been developed for laminar flames.^{13,14} Works by Buriko and Kuznetsov (1978),¹⁵ Turns and Mayr (1991),¹⁶ Faeth et al. (1989),¹⁷ and Delichatsios et al. (1988, 1992)^{18,19} have quantified radiation from flames for different fuels of different sooting tendencies, using the radiant fraction as a scaling parameter. All of the above were for free jets, without confinement and not inside a hot combustor. The radiation scaling inside a combustor is more complicated because of recirculation effects and reradiation from the furnace walls to the flame gases.

Weber (1996) reported on burner scaling effects for firing rates ranging from 7 kW to 14 MW (24×10^3 to 48×10^6 Btu/hr) or a 2000:1 turndown range.²⁰ The goal of the study, known as the Scaling 400 Study for the nominal turndown of 400:1,²¹ was to determine the proper method for scaling burner sizes. The burner used in that study was a staged mixing burner for primary, secondary, and tertiary air, with variable swirl capability. The primary conclusions of the study were that:

- The fuel-to-air momentum ratio must be maintained.
- The geometrical burner similarity must be maintained (small departures are acceptable).
- The confinement effects of the furnace are secondary if the confinement ratio (furnace-to-burner diameter ratio) is larger than 3.
- The inlet swirl should be reduced by 20 to 30% for laboratory-scale burners with characteristic diameters less than 2 in. (5 cm).

Baukal and Gebhart (1998) studied oxygen-enhanced natural gas flames impinging normal to a water-cooled flat metal disk.²² The firing rate of the burner was one of the parameters studied to determine its effect on the heat transfer to the target. The heat flux was determined by calorimetry to concentric water-cooled rings on the target. The firing rate was varied from 5 to 25 kW (17,000 to 85,000 Btu/hr). The upper limit was a function of the flow control equipment. Figure 6.5 shows contours of the heat flux as functions of both the axial and radial distances from the burner, with 35% total O_2 in the oxidizer ($\Omega = 0.35$). This was the lowest oxidizer composition that could be used which produced an acceptable flame throughout the entire range of firing rates for the burner that was used (see Figure 2.23). These plots show that for small L , the peak heat flux did not occur at the stagnation point, but at about $R_{eff} = 0.5$ to 0.7 . This was caused primarily by a slightly higher heat output from the burner at that radial location.²³ The heat flux increased by 40 to 230%, depending on the axial and radial position, by increasing the firing rate 400% by going from 5 to 25 kW (17,000 to 85,000 Btu/hr). The smallest percentage improvements were for the closest axial ($L = 0.5$) and radial ($R_{eff} = 0.16$) locations. This was because the heat flux rate was already relatively high there, compared to farther axial and radial locations. The largest percentage improvements

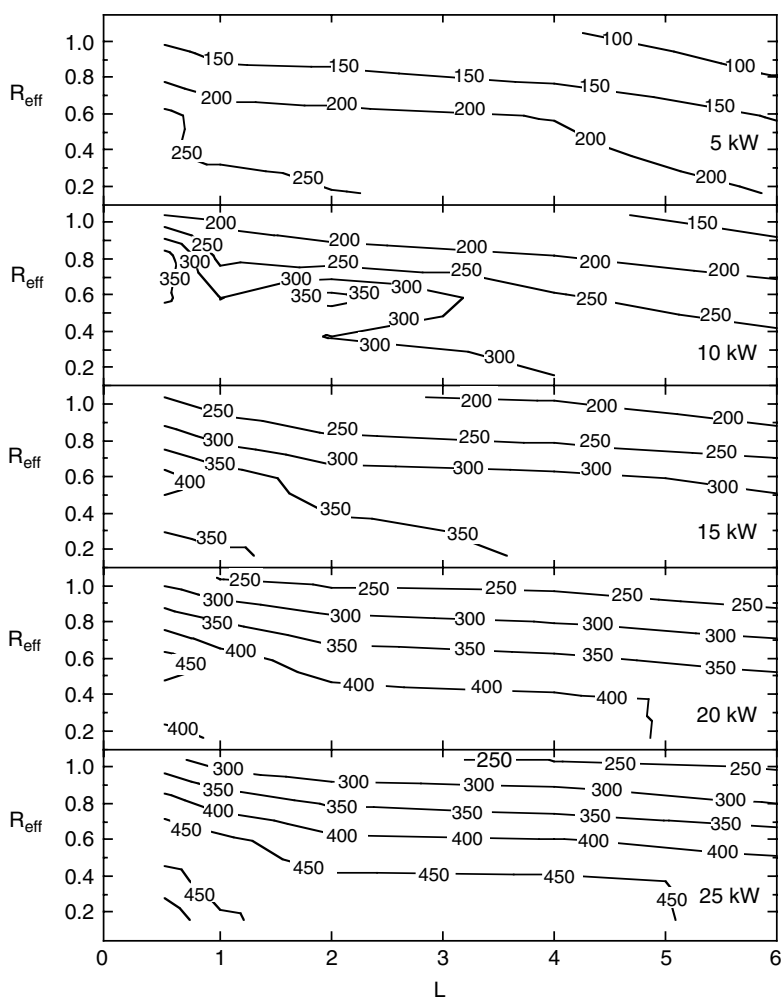


FIGURE 6.5 Contours of the total heat flux, q'' (kW/m²), from stoichiometric natural gas flames ($\Omega = 0.35$) with various firing rates ($q_f = 5$ to 25 kW) impinging on the surface of an untreated stainless target.²²

occurred at the farthest axial ($L = 6$) and radial ($R_{\text{eff}} = 1.04$) locations, because the heat flux rate was initially so low there. At $q_f = 5$ kW and $R_{\text{eff}} = 1.04$, the heat flux increased by 91% by decreasing the axial spacing from $L = 6$ to $L = 0.5$. However, at $q_f = 25$ kW (85,000 Btu/hr) and $R_{\text{eff}} = 0.16$, the heat flux decreased by 14% by decreasing the axial spacing from $L = 6$ to $L = 0.5$. In general, at higher firing rates, the heat flux did not have a strong dependence on the axial distance L between the flame and the target. At the farthest axial location ($L = 6$), the heat flux to the inner calorimeter was approximately double the heat flux to the outer calorimeter. However, at the closest axial spacing ($L = 0.5$), the peak flux occurred at about the middle calorimeter ($R_{\text{eff}} = 0.59$).

Figure 6.6 shows heat flux contours similar to Figure 6.7, but for a pure O oxidizer. The peak fluxes occurred at the stagnation point, except for $q_f = 20$ and 25 kW (68,000² and 85,000 Btu/hr) for $L = 0.5$, where a slightly higher flux was measured at $R_{\text{eff}} = 0.37$. The heat flux increased by 78 to 470%, depending on the axial and radial locations, by increasing the firing rate from 5 to 25 kW (17,000 to 85,000 Btu/hr). Again, the largest percentage improvements occurred at the farthest locations while the smallest improvements occurred at the closest locations. Unlike the $\Omega = 0.35$ data in Figure 6.6, the heat flux strictly increased, by 42 to 230%, as the axial spacing decreased

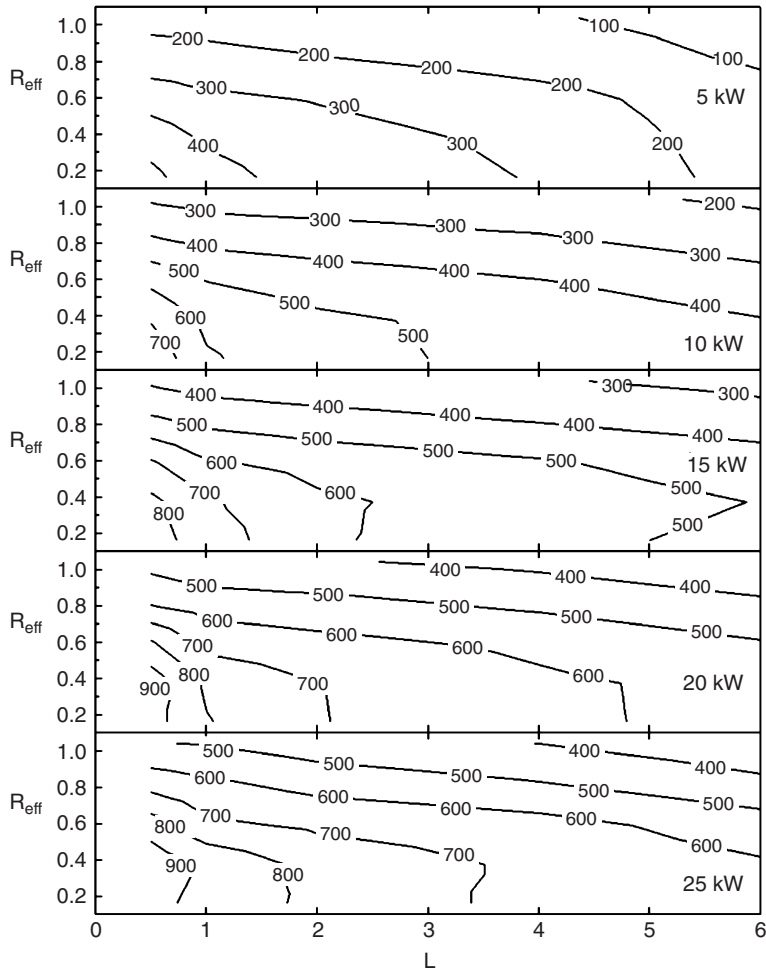


FIGURE 6.6 Contours of the total heat flux, q'' (kW/m^2), from stoichiometric natural gas flames ($\Omega = 1.00$) with various firing rates ($q_f = 5$ to 25 kW) impinging on the surface of an untreated stainless target.²²

for the $\Omega = 1.00$ data in Figure 6.6. Like the $\Omega = 0.35$ data, at large axial spacings, the heat flux to the innermost calorimeter was approximately double the flux to the outermost calorimeter.

Figure 6.7 shows how the heat flux to the innermost calorimeter was affected by the axial location of the target and by the burner firing rate. The heat flux increased with the firing rate and decreased with the axial distance. The shape of the contour lines was significantly different for the $\Omega = 0.35$ flames, compared to the $\Omega = 1.00$ flames. The peak flux occurred at an intermediate axial location ($L \approx 2$) for the $\Omega = 0.35$ flames. For the $\Omega = 1.00$ flames, the heat flux strictly increased as the axial spacing decreased.

Figure 6.8 shows that for $\Omega = 1.00$ and large L , the heat flux increased rapidly with the firing rate. The shape of each curve is similar. In all cases, the heat flux increased with the firing rate and decreased with the axial spacing. The thermal efficiency η was defined as:

$$\eta = \frac{\sum_{i=1}^5 q''_i A_i}{q_f} \quad (6.8)$$

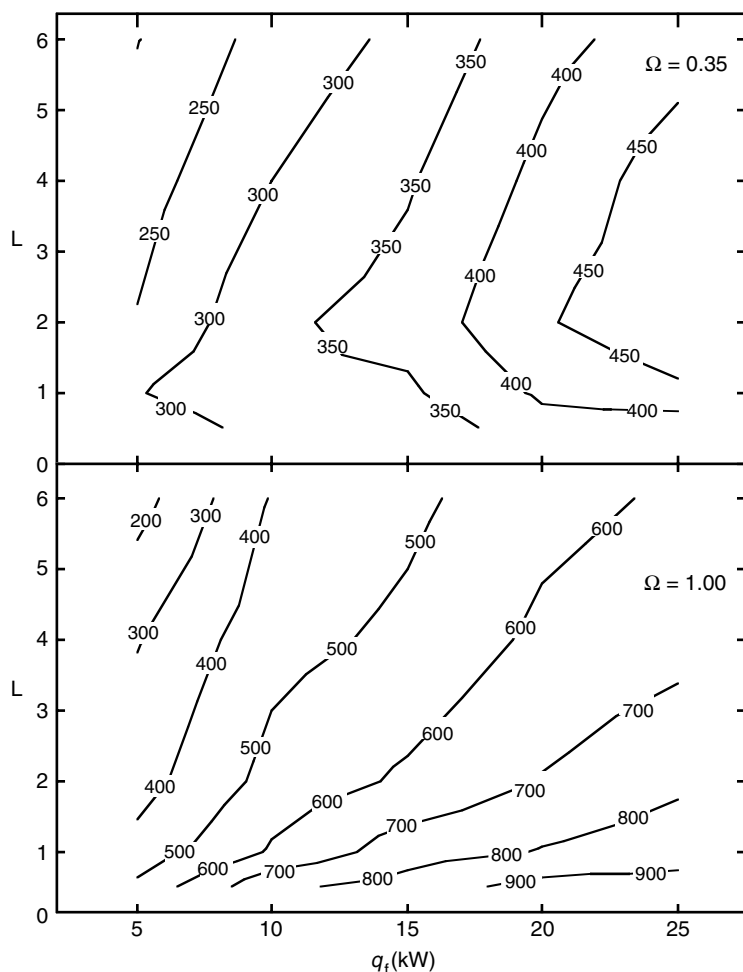


FIGURE 6.7 Contours of the total heat flux, q'' (kW/m²), from stoichiometric natural gas flames of various firing rates ($q_f = 5$ to 25 kW) and oxidizer compositions ($\Omega = 0.35$ and 1.00) to the stagnation point ($R_{\text{eff}} = 0.16$) of an untreated brass target.²²

where q''_i was the calculated heat flux to ring i , A_i was the impingement surface area of ring i , and q_f was the burner firing rate. The efficiency was then the total energy absorbed by the first five calorimeters divided by the burner firing rate. Figure 6.9 shows how the thermal efficiency varied with the firing rate, oxidizer composition, and axial location. The efficiency decreased with L , increased with Ω , and decreased with the firing rate. At $\Omega = 1.00$ and $q_f = 5$ kW (17,000 Btu/hr), the efficiency decreased rapidly from $L = 4$ to $L = 6$, because, at $L = 6$, the visible flame length was less than the distance between the burner and the target. Buoyancy effects were visually evident, as the flame shape was highly transient and wavy. Air infiltration may have significantly reduced the heat flux to the target for that set of conditions.

6.2.4 FLAME SHAPE EFFECTS

Hutchinson et al. (1975) experimentally and numerically showed that a quarl (burner block or tile) shortened the flame on an air/natural gas burner with swirl firing into a water-cooled cylindrical furnace.²⁴ The recirculation zone in the furnace was pulled closer to the burner with the

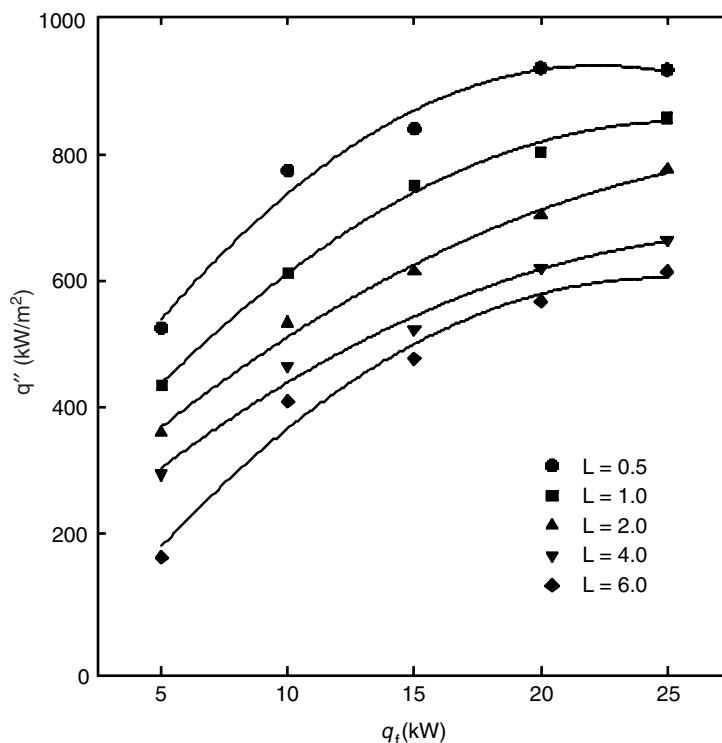


FIGURE 6.8 Total heat flux (q'') from stoichiometric natural gas flames ($\Omega = 1.00$) of various firing rates (q_f) impinging on the stagnation point ($R_{\text{eff}} = 0.16$) of an untreated stainless target.²²

quarl, compared to the same burner without a quarl. Higher temperatures and heat fluxes were measured closer to the burner with the quarl, than without the quarl. The quarl shortened the flame and moved the heat release closer to the burner. The calculations showed that the convective heat transfer and the radiative heat transfer to the furnace wall were of the same order with or without the quarl.

One recent burner development trend in about the last decade has been the development of so-called flat flame burners. This can mean different things, but here it refers to a rectangular flame shape (see Figure 16.29), as opposed to the traditional round flame shape. The goal is to have a higher flame surface area to increase the area that radiates from the flame to the load. This trend in burner design has been particularly evident in the use of oxygen-enriched burners (see Chapters 20 and 21) in the glass and aluminum industries. If the trend continues, this type of flame will likely be applied to other industrial heating and melting applications.

There are numerous examples of the use of flat flames in glass melting (e.g., Figure 20.24). Kirilenko et al. (1988) described a flat flame burner used in a regenerative glass-melting furnace.²⁵ Ibbotson (1991) described a technique for lancing a fan-shaped jet of oxygen under a fuel port in a glass-melting furnace to improve temperature uniformity and minimize hot spots.²⁶

Flat shape flames have also been used in aluminum melting. Yap and Pourkashanian (1996) described a flat flame oxy/fuel burner that they describe as a large aspect-ratio flame.²⁷ The burner was claimed to have more uniform heat transfer rates, compared to conventional round flame burners. The burner also had excellent flame stability, resulting in a wide operating range, low NO_x emissions, and high flame luminosity. The combination of low NO_x emissions and high luminosity, both of which are important in most heating applications, is a unique aspect of the burner.

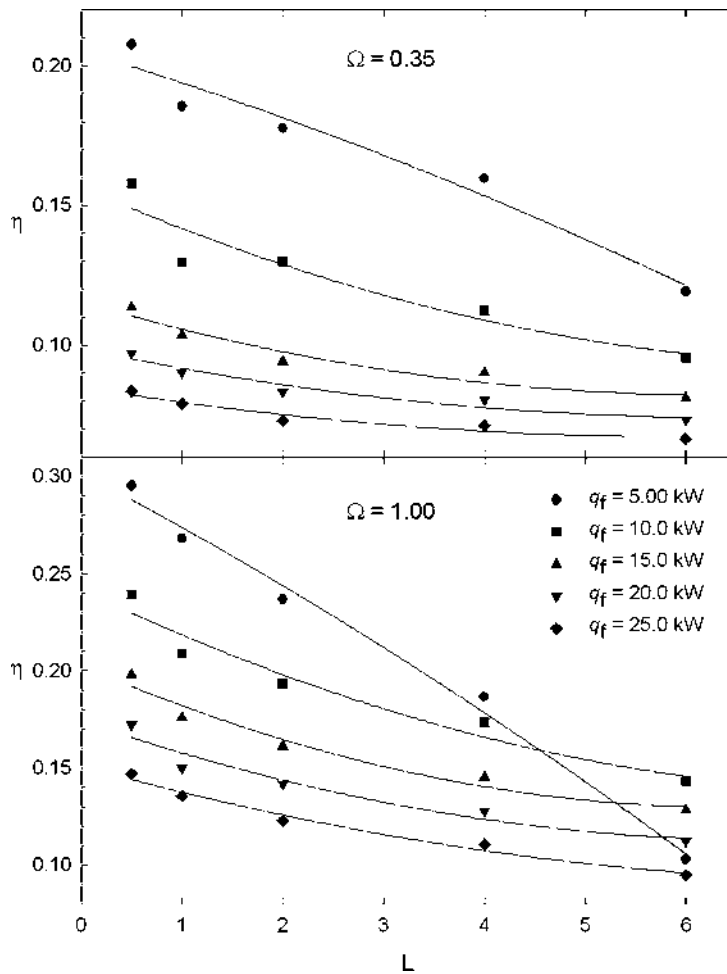


FIGURE 6.9 Thermal efficiency (η) for stoichiometric natural gas flames of various firing rates ($q_f = 5$ to 25 kW) and oxidizer compositions ($\Omega = 0.35$ and 1.00).²²

In addition, the burner minimized oxidation of the molten aluminum, which maximizes the product yield.

The length of the flame influences the heat transfer as longer flames may be more luminous and lower temperature compared to similar flames at the same operating conditions. Blake and McDonald (1993) have correlated visible flame length data for vertical turbulent diffusion flames as follows:²⁸

$$\frac{l_f}{d_f} = \alpha_1 \left(\frac{4J}{\pi \rho_\infty g d_f^3} \right)^{\alpha_2} \quad (6.9)$$

where l_f is the flame length, d_f is the flame diameter, J is the momentum of the source jet, ρ_∞ is the density of the ambient, g is the gravitational constant, and α_1 and α_2 are constants as given in Table 6.1. The fraction inside the parentheses in Equation 6.9 is referred to as the density weighted Froude number.

TABLE 6.1
Constants for Flame Length Equation

$4J/\pi\rho_{\infty}gd_f^3$	α_1	α_2
10^{-9} – 10^{-6}	6.73	0.209
10^{-6} – 10^{-4}	7.79	0.218
10^{-4} – 10^{-1}	7.25	0.184

Source: Adapted from T.R. Blake and M. McDonald, *Comb. Flame*, 94, 426–432, 1993.

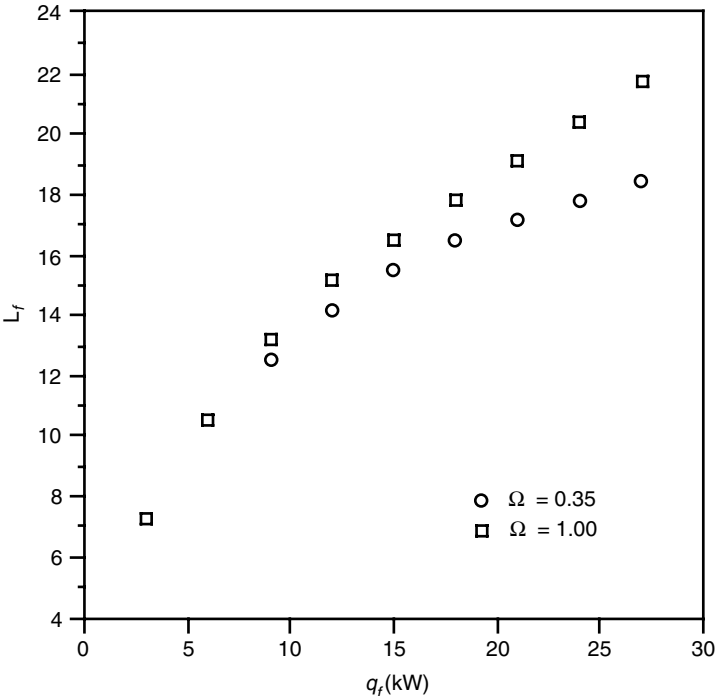


FIGURE 6.10 Flame length as function of firing rate and oxidizer composition for stoichiometric natural gas flames.²⁹

Figure 6.10 shows how the visible flame length for an open-flame diffusion burner varies as a function of the firing rate and oxidizer composition.²⁹ The flame length was nearly the same for lower firing rates. At higher firing rates, the pure O₂ flames ($\Omega = 1.00$) were longer than the lower purity flames ($\Omega = 0.35$). The flame length affects the flame shape, which influences the radiating area of the flame.

6.3 RADIANT BURNERS

Radiant burners operate by combusting a fossil fuel, which heats a solid surface that radiates infrared energy to a load. These burners are used in a number of lower temperature heating and drying applications, including, for example:

- Drying paper and cardboard in a paper mill
- Paper finishing
- Drying wood
- Porcelain frit drying
- Curing ceiling tiles
- Teflon curing
- Drying and curing coatings on paper and metals
- Curing ink on paper and powder coat paints
- Baking in mass food preparation
- Setting dyes in textile and carpet production, sometimes referred to as pre-drying
- Curing lens assemblies in automotive headlamp assembly manufacturing
- Plastics curing

Both gas-fired and electric radiant heaters are commonly used, however, gas-fired radiant burners have lower operating costs due to the difference between the cost of electricity and natural gas in most locations. Pettersson and Stenström (1998) compared gas-fired and electric IR heaters in two paper-coating drying processes.³⁰ The efficiency of the gas-fired and electric dryer was 30% and 40%, respectively. However, there was considerable uncertainty in the measurements due to the difficulty in obtaining accurate moisture measurements of the paper being dried. Even if the efficiencies are correct, the gas-fired system may still be more economical, depending on the cost of electricity vs. natural gas.

Radiant burners (see [Chapter 13](#)) are designed to produce a uniform surface temperature heat source for heating and melting a variety of materials. The uniform surface temperature produces more homogeneous heating of the materials, which normally improves the product quality compared to conventional burners that may produce hot spots. Other advantages of these burners may include:

- High thermal efficiencies
- Low pollutant emissions
- Directional heating
- Very fast response time to load changes
- Very fast heating compared to convective heating
- Burner shape can be tailored to the shape of the heat load to optimize heat transfer
- Ability to segment a burner to produce a nonuniform heat output profile, which may be useful in certain types of heating and drying applications
- Certain types of radiant burners have very rapid heat-up and cool-down times
- No open flames that could ignite certain types of materials (e.g., paper or textiles)
- More control over the heating process because of the known and measurable surface temperature of the radiant surfaces compared to open flames, where the flame temperature is very difficult to measure
- Burners are very modular and can be configured in a wide variety of geometries to accommodate the process heating requirements

The primary parameters of interest for radiant burners are the power density (firing rate per unit area), radiant efficiency (fraction of fuel heating value converted to thermal radiation), heat-up and cool-down time, and pollutant emissions. Other factors of importance include cost, durability, and longevity.

There are also some important limitations of porous refractory burners, compared to more conventional open-flame burners, which may include:

- Relatively low temperature limit for the radiant surface due to the limits of the refractory material
- Fuel and oxidizer must be clean to avoid plugging the porous radiant surface, which essentially precludes the use of fuels such as coal or heavy fuel oil

- Some of the radiant surfaces can be damaged by water or by contact with solid materials that may be prevalent in certain applications
- Holes in the radiant surface can cause flashback because these burners use premix
- Some designs may have high pressure drops, which means that more energy is needed for the blower to flow the combustion air through the ceramic burner material
- Due to the limits in radiant surface temperatures, the firing rate density is usually limited
- Some types of radiant burners using hard ceramic surfaces may have high heat capacities that could ignite certain load materials upon a sudden line stoppage

In these burners, fuel and air are premixed and combusted, either just inside a radiating surface or just above the surface, depending on the operating conditions and specific radiant burner design. If the mixture velocity is too low, flashback or flame extinguishment can occur, depending on the design of the burner. In addition to the operational considerations, flashback is an obvious safety concern. If the mixture velocity is too high, the flame may blow off or the radiant performance may be severely reduced because the burner surface is not being directly heated by the hot exhaust products. Depending on the specific design of the burner, optimum performance is achieved when the flame is stabilized just inside or just above the outer burner outlet.

Howell et al. (1995) refer to the burner material as “porous inert media” and present a review of these types of burners that have been made from a wide variety of ceramics, which often include alumina, zirconia, or silicon carbide.³¹ Hsu and Howell (1993) developed the following equation for the effective thermal conductivity of partially stabilized zirconia (PSZ) (zirconium oxide plus <3% magnesium oxide) for temperatures in the range of 290 to 890K (63 to 1140°F) and for sample pore sizes from 10 to 65 pores per inch (ppi):³²

$$k \text{ (W/m-K)} = 0.188 - 0.0175 \, d \tag{6.10}$$

where d is the actual pore size of the material (in mm) and $0.3 < d < 1.5$ mm.

Speyer et al. (1997) studied the performance of four different types of commercial gas-fired radiant burners: a metal (Fe-Cr-Al) fiber (~40 μm diameter) burner, a reticulated ceramic burner made of a porous cordierite ($\text{Mg}_2\text{Al}_4\text{Si}_5\text{O}_{18}$), a porous single-unit mullite ($\text{Al}_6\text{Si}_2\text{O}_{13}$) tile burner, and a flame impingement burner where the products of combustion are forced around a metal screen that radiates to the load.³³ The study was sponsored by the Gas Research Institute (Chicago, IL). The thermal efficiencies as a function of the combustion intensity are shown in Table 6.2. The efficiency was determined by measuring the radiant output or radiosity from the burner surface and

TABLE 6.2
Comparison of Thermal Efficiencies for Radiant Burners

Radiant Burner Type	Combustion Intensity (kW/m ²)	Thermal Efficiency Range (%)
Metal fiber		
No perforations	150–540	18–26
With perforations	150–540	16–27
With perforations and front metal screen	100–440	22–38
Reticulated ceramic	150–630	27–39
Ported ceramic	140–520	37–54
Flame impingement	150–430	46–52

Source: Adapted from R.F. Speyer, W.-Y. Lin, and G. Agarwal, *Exp. Heat Transfer*, 9, 213–245, 1996.³³

was calculated using:

$$e = \frac{R_T}{\dot{V}_g \Delta H_c / A} \quad (6.11)$$

where e is the calculated efficiency, R_T is the measured radiosity, $\dot{V}_g$ is the volume flow rate of the fuel, H_c is the heating value of the fuel, and A is the surface area of the burner. The experimental results showed that the peak radiosity measurements as a function of the air-to-fuel mixture ratio occurred for percent excess fuels ranging from 6 to 7.5%, depending on the specific burner design. Similar results were calculated for the radiant efficiencies. Preheating the air/fuel mixture increased both the radiosity and thermal efficiencies essentially in a linear manner for mixture temperatures ranging from 40 to 155°C (100 to 311°F). In most cases, the efficiency declined with combustion intensity, except for the reticulated ceramic and metal fiber with perforations and a front metal screen, both of which had peak efficiencies at intermediate combustion intensities. The flame impingement and ported ceramic burners were the most efficient, while the metal fiber with and without perforations was the least efficient. The study also showed that the thermal efficiency generally increased as the fraction of closed area increased on the burner surface.

The heat transfer coefficient between the hot exhaust products and the radiant burner material is difficult to predict and measure due to the uncertainty in the surface area of the ceramic structure. These coefficients have traditionally been presented in terms of a volumetric coefficient (Btu/hr-°F-ft³ or kW/°C-m³). Chen et al. (1987) assumed the coefficient was large enough that the temperature of the solid was essentially the same as the temperature of the hot flowing gas.³⁴ Sathe et al. (1990) used a value of 2×10^9 W/m³-K (1×10^8 Btu/hr-°F-ft³) computed for cylinders in cross flow.³⁵ Hsu et al. (1993) used 10^7 W/m³-K (5×10^5 Btu/hr-°F-ft³).³⁶ Younis and Viskanta (1993) empirically determined the following correlation for alumina and cordierite ceramic foams with pore diameters ranging from 0.29 to 1.52 mm (0.011 to 0.0598 in.):³⁷

$$\text{Nu} = 0.819 [1 - 7.33 (d/L)] \text{Re}^{0.36[1 + 15(d/L)]} \quad (6.12)$$

where d is the actual pore diameter and L is the thickness of the specimen in the flow direction. Fu et al. (1998) noted the relationship between the volumetric heat transfer coefficient h_v and the convection coefficient h :³⁸

$$h_v = a_v h \quad (6.13)$$

where a_v can be computed using the following empirical equation:

$$a_v = 169.4 \text{ PPC (m}^2/\text{m}^3) \quad (6.14)$$

where PPC is the number of pores per centimeter for cellular ceramics. Four different choices for the characteristic dimension were identified:

$$\text{The reciprocal of the surface area, } d_a = 1/a_v \quad (6.15)$$

$$\text{The hydraulic diameter, } d_h = 4\phi/a_v \quad (6.16)$$

$$\text{The mean pore diameter, } d_m = \frac{\sqrt{4\phi/\pi}}{\text{PPC}} \text{ (cm)} \quad (6.17)$$

$$\text{The ratio of the inertial to the viscous friction coefficients, } d_r = \beta/\alpha \quad (6.18)$$

where ϕ is the porosity, β is the inertia coefficient in the Reynolds-Forchheimer equation, and α is the viscous coefficient in the Reynolds-Forchheimer equation. The following empirical correlations,

using each of these characteristic dimensions, were developed for specimens having a PPC ranging from 4 to 26:

$$\text{Nu}_v = \left[0.0426 + \frac{1.236}{L/d_m} \right] \text{Re}_{d_m} \quad \text{for } 2 \leq \text{Re}_{d_m} \leq 836 \quad (6.19)$$

$$\text{Nu}_v = \left[0.0730 + \frac{1.302}{L/d_h} \right] \text{Re}_{d_h} \quad \text{for } 3 \leq \text{Re}_{d_h} \leq 1594 \quad (6.20)$$

$$\text{Nu}_v = \left[0.0252 + \frac{1.280}{L/d_a} \right] \text{Re}_{d_a} \quad \text{for } 1 \leq \text{Re}_{d_a} \leq 480 \quad (6.21)$$

$$\text{Nu}_v = \left[0.000267 + \frac{1.447}{L/d_r} \right] \text{Re}_{d_r} \quad \text{for } 0.02 \leq \text{Re}_{d_r} \leq 2.4 \quad (6.22)$$

where Nu_v is the volumetric Nusselt number, defined as:

$$\text{Nu}_v = \frac{h_v l_c^2}{k} \quad (6.23)$$

where l_c is the characteristic pore length and k is the thermal conductivity of the gas. The uncertainty in the correlations was largest for the smaller Reynolds numbers.

One of the challenges of gas-fired infrared burners is determining the radiant efficiencies. Mital et al. (1998) noted that there is a wide discrepancy in the reported radiant efficiencies (varying by more than 200%) for radiant burners.³⁹ Part of the discrepancy was attributed to a lack of a standard measurement technique to determining the efficiency. Other problems include nondiffuse radiation from the burners and nonuniform burner surfaces. They presented a technique that is not sensitive to burner surface nonuniformities. Typical results for a reticulated ceramic foam radiant burner are shown in Figure 6.11. A calorimetric method was used to check the consistency of the data. It was shown that single-point radiation measurements can deviate considerably from more rigorous multi-point measurement techniques. Yetman (1993) described a simple technique for measuring the total radiant output of an infrared burner using a narrow-angle pyrometer.⁴⁰ Johansson (1993) presented a method for measuring the spectral output of radiant burners using an infrared spectrometer.⁴¹

Madsen et al. (1996) measured the spectral radiation from several types of radiant burners.⁴² As can be seen, the spectra are fairly similar that have distinctive peaks around 3 and 4.5 μm . This shows the selective emittance of certain types of radiant burners and the possibility of matching the burner to the load spectral absorptivity to optimize heat transfer efficiency.

6.3.1 PERFORATED CERAMIC OR WIRE MESH RADIANT BURNERS

Examples of these types of burners are shown in Figure 6.12. Perforated or ported ceramic burners may consist of a pressed ceramic plate that may include prepunched holes, where the flames heat the surface directly.⁴³ The surface can be textured to further enhance the radiant efficiency of the burner. New developments in ceramic foams are being applied to this type of burner. These foams are often less expensive to make than perforated ceramics. They provide a higher surface area for radiation and a more uniform heating surface, compared to perforated ceramics. Many shapes are possible with the ceramic foams and the pore size is adjustable. Flanagan et al. (1992) described the use of a ported ceramic burner to achieve low NO_x emissions.⁴⁴ However, the burner was

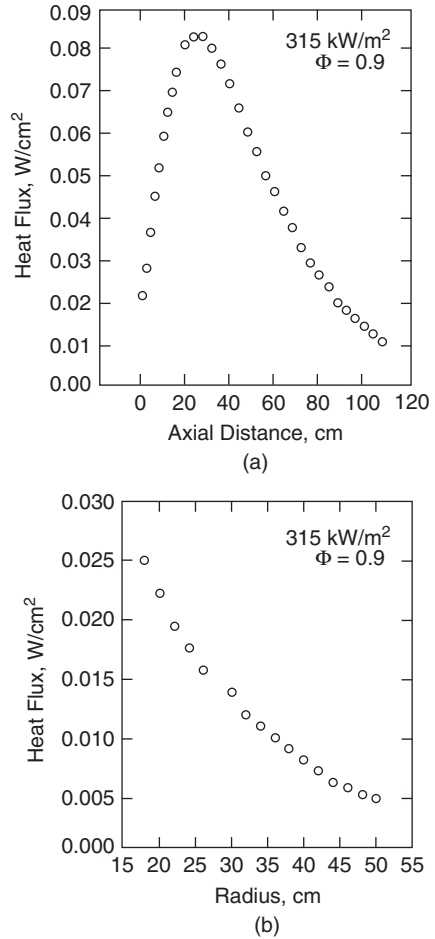


FIGURE 6.11 Heat flux measurements from a ceramic foam radiant burner as a function of (a) the axial distance from the burner surface and (b) the radial distance from the center of the burner. (Courtesy of The Gas Research Institute.³⁹)

actually fired at up to 15 times its normal maximum design firing rate so that the flames were highly lifted and therefore did not radiate from the surface as in normal operation. Mital and Gore (1994) discussed the use of a reticulated ceramic insert to enhance radiation heat transfer in direct-fired furnaces.⁴⁵ The insert was placed downstream of the outlet of a laminar diffusion flame. Experimental results showed up to a 60% improvement in radiative heat flux compared to the case with no insert. There was a significant difference in radiative heat flux with and without the insert. Kataoka (1998) described a new type of high-temperature porous ceramic burner made of aluminum titanate (Al_2TiO_5) capable of surface temperatures up to 1100°C (2000°F).⁴⁶

Wire mesh burners are made from high-temperature-resistant metals, such as stainless steels or inconel. The open area in the mesh serves as the port area for the burner. However, due to the high thermal conductivity of metals, several layers of mesh are often required to prevent flashback. The thermal conductivity between the layers is much less than through the mesh itself because of the contact resistance between the layers. An important problem with wire mesh radiant burners are the lower temperature limits compared to ceramic burners, due to the temperature limits of the metals.

6.3.2 FLAME IMPINGEMENT RADIANT BURNERS

This burner is sometimes referred to as a direct-fired refractory burner. In this type of radiant burner, the flame impinges on a hard ceramic surface, which then radiates to the load. The heat transfer from the flame to the tile is mostly by convection due to the direct flame impingement. The heat

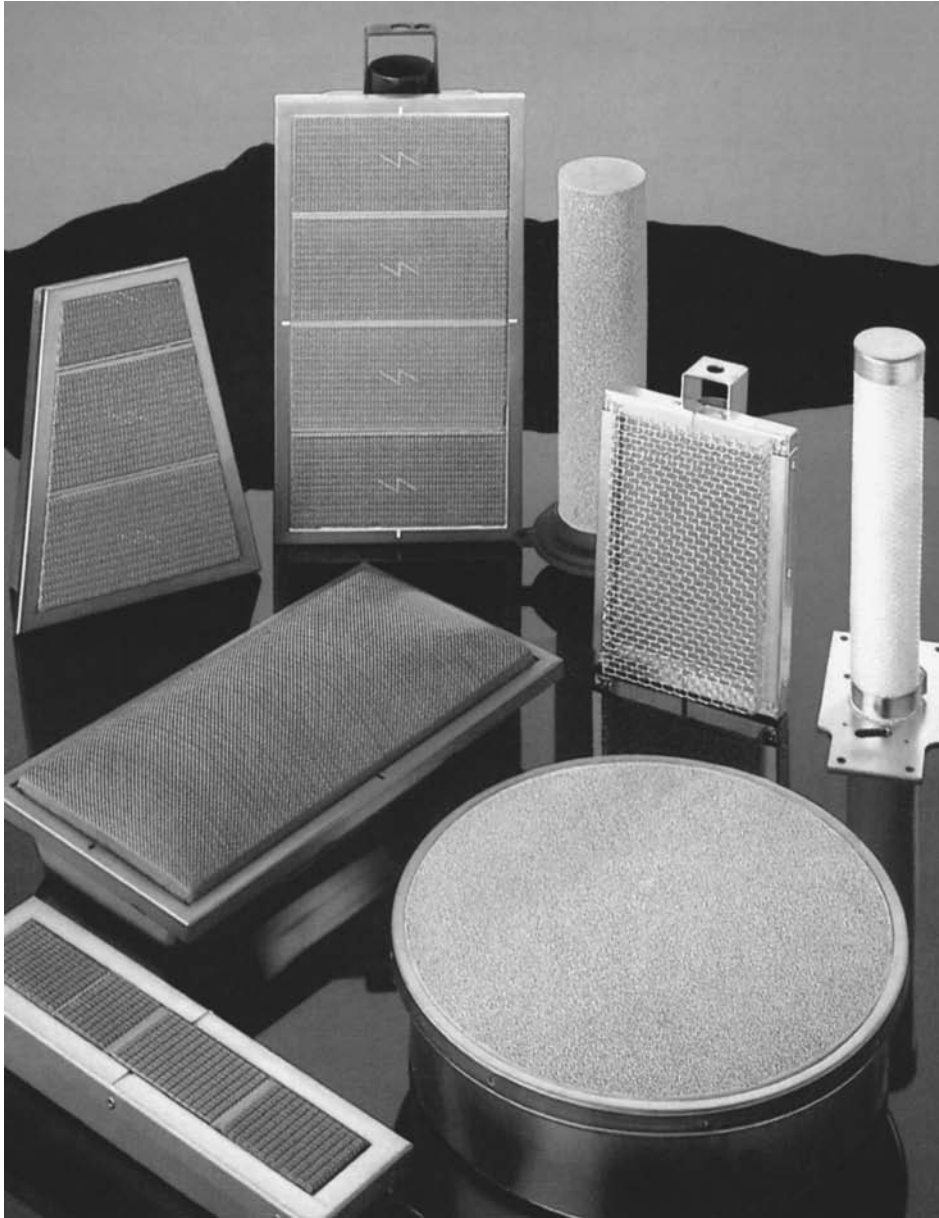


FIGURE 6.12 Examples of porous ceramic and wire mesh radiant burners. (Courtesy of Solaronics.⁴³)

transfer from the burner tile to the load is purely by radiation. The hot exhaust gases from the burner heat up the surrounding wall by convection, but at a much lower rate than to the burner tile. One advantage of this type of burner is that there is no metal matrix with lower temperature limits compared to ceramic that has high temperature limits. Therefore, this type of burner can often be used in higher temperature applications. There also is no matrix of porous ceramic fiber refractory that could get plugged up as in the next type of burner. This type of burner can also fire a liquid fuel as opposed to many other radiant burners that use only gas. A disadvantage is that the heat flux is not as uniform as other types of radiant burners. Another problem is that the burner tile has the typical problem of thermal cycling, which can cause the tile to disintegrate.

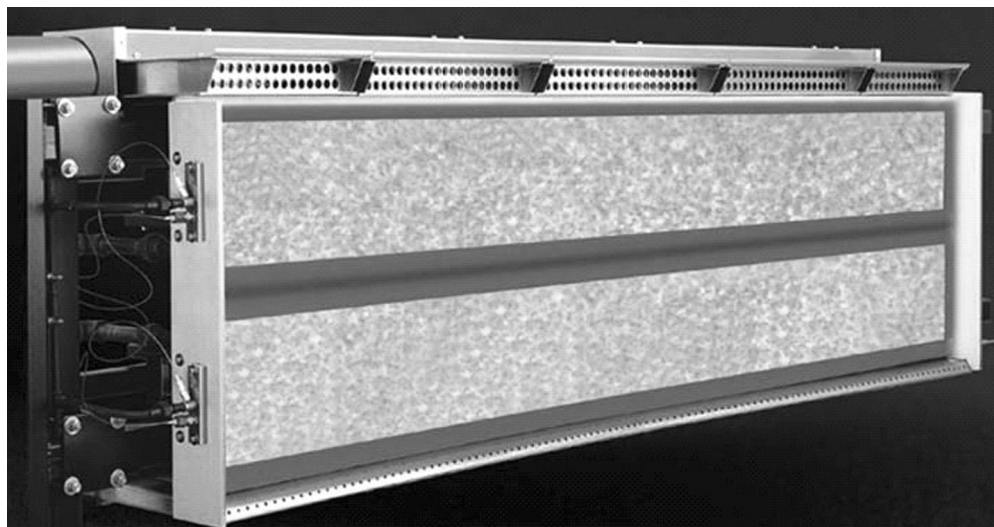


FIGURE 6.13 Flat-panel infrared burner. (Courtesy of Marsden, Inc., Pennsauken, NJ.)

6.3.3 POROUS REFRACTORY RADIANT BURNERS

In this type of burner, the surface is made of a porous ceramic fiber that is often made in a vacuum-forming process. A relatively new type is now available, made from a woven ceramic fiber mesh, similar to the wire mesh radiant burners except that ceramic fiber is used instead of metal. The predominant shape used in porous refractory burners is a flat panel. An example is shown in [Figure 6.13](#).

In addition to flat panels, other shapes are also available. Alzeta Corp. (Santa Clara, CA) makes many cylindrically shaped porous refractory burners.^{47,48} For their Duratherm™ burners, sizes range from as small as 2 in. (5 cm) in diameter by 4.5 in. (11 cm) long, to as large as 30 in. (76 cm) in diameter by 180 in. (460 cm) long. Firing rates range from 23,000 Btu/hr (6.7 kW) to 16.5×10^6 Btu/hr (4.83 MW). Bartz et al. (1992) describe the use of an Alzeta porous refractory radiant burner to incinerate volatile organic compounds (VOCs).⁴⁹ The unique aspect of the combustion system was that the burner was formed into a cylinder and fired inwardly while the VOCs flowed into that inner core with the combustion products where the VOCs were then destroyed. Very high VOC destruction efficiencies were measured.

The American Gas Association made an extensive study of radiant burners, both gas fired and electric.⁵⁰ For the gas-fired burners, they determined what was termed a “Gas Infrared Radiation factor” (GIR), which was determined as follows:

$$\text{GIR} = \frac{W}{\sigma T_b^4} \quad (6.24)$$

where $W = Q/A$ was the total normal infrared radiation from the burner measured by a spectrophotometer, σ is the Stefan-Boltzmann constant, and T_b is the absolute brightness temperature of the burner surface measured with an optical pyrometer. The GIR factor was similar to a burner emissivity. However, for some of the burners, the GIR factor exceeded 1.0. The GIR factor was approximately 15% higher than the true burner surface emissivity. This is due to the gaseous non-luminous radiation in addition to the surface radiation from the burner. For the burners tested, the GIR factor ranged from 0.36 to 1.17, depending on the burner type.

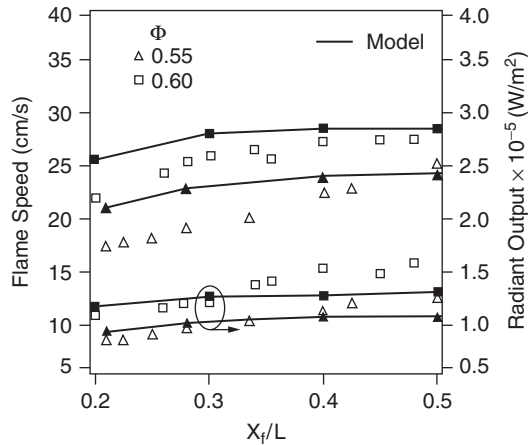


FIGURE 6.14 Measured and predicted flame speed and radiant output for a porous radiant burner. (Courtesy of The Combustion Institute.⁵¹)

Sathe et al. (1990) made experimental measurements and numerical predictions on a porous radiant burner.⁵¹ They used the following correlation for flow over circular cylinders for the convection heat transfer coefficient from the flame to the porous refractory:

$$Nu = 0.989 Re^{0.33} \quad (6.25)$$

where the characteristic length d is the equivalent particle diameter. Figure 6.14 shows a comparison of the measured and predicted result as a function of the axial location of the flame (x_f) normalized by the porous layer length (L). The figure shows that both the radiant output and the flame speed are fairly independent of the flame location, but are dependent on the equivalence ratio. Zabielski et al. (1991) used an optical technique to measure the radiant characteristics of a porous ceramic fiber radiant burner.⁵² The macroscopic emittance of the burner was estimated to be 0.70. Xiong and Viskanta (1992) measured the heat flux and calculated the thermal efficiency of a porous matrix ceramic combustor.⁵³

Jugjai and Sanitjai (1996) presented a concept for a new type of porous radiant burner incorporating internal heat recirculation to increase thermal efficiency.⁵⁴ They studied the effects of the optical thickness of the porous medium on the radiant output of porous radiant burners. The optical thickness was increased by added layers of stainless steel wire mesh that formed the porous medium for the burners tested. The added layers improved the heat transfer from the hot gases to the porous medium.

Mital et al. (1996) claimed to have made the first measurements of temperature and species distributions inside submerged flames stabilized inside porous ceramic burners.⁵⁵ They note the wide discrepancy in the performance of these burners in the literature and in manufacturers' literature.

Rumminger et al. (1996) developed a one-dimensional model of a bilayered reticulated ceramics radiant burner.⁵⁶ The model results showed that for a "submerged flame" where the flame is anchored inside the porous refractory, nearly all of the important reactions occur inside the porous medium. This complicates the study for this type of burner because of the difficulty in making measurements inside the porous medium. They speculate that it is possible the large internal surface area of the porous medium could affect the chemistry, but there is no data at this time to confirm that possibility.

Van der Drift et al. (1997) studied various coatings on porous foam ceramic infrared burners to determine the effects on heat transfer to wet paper and to three colors (white, blue, and black).⁵⁷ They showed that there can be up to a 10% improvement using coated burners compared to a base case metal fiber burner.

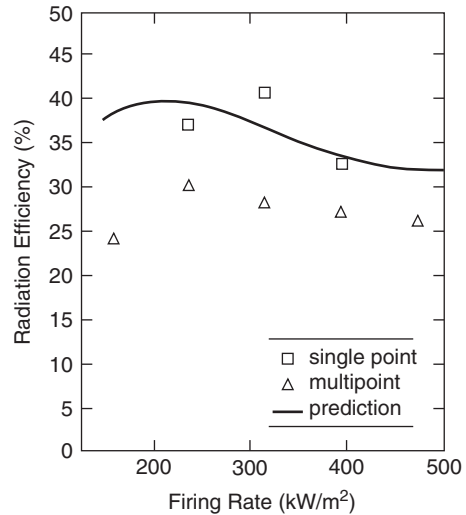


FIGURE 6.15 Predicted and measured radiation efficiency for a porous radiant burner at an equivalence ratio of 0.9. (Courtesy of The Combustion Institute.⁵⁸)

Mital et al. (1998) made experimental measurements on a bilayered reticulated ceramic foam made of Cordierite.⁵⁸ Measurements included radiation efficiency using a heat flux gage and burner surface measurements with a type R thermocouple. Radiation efficiency results are shown in Figure 6.15, which include both single and multi-point measurements and model predictions.

6.3.4 ADVANCED CERAMIC RADIANT BURNERS

Tong et al. (1989) showed through computer modeling that the performance of porous radiant burners can be improved by as much as 109% using sub-micron diameter ceramic fibers.⁵⁹ Bell et al. (1992) described a staged porous ceramic burner that demonstrated low NO_x emissions.⁶⁰ Kendall and Sullivan (1993) studied enhancing the radiant performance of porous surface radiant burners using improved ceramic, high-temperature, high-emissivity fibers.⁶¹ Selective emissivity was achieved by coating a standard ceramic fiber burner with an outer layer of ytterbia. The output of the uncoated vs. the coated burner is shown in Figure 6.16. As can be seen, there is a large spike in the output of the coated burner at around 1 μm . However, there was not a significant improvement in performance and the durability was in question. In certain applications, there may be a need to have a burner with selective radiant emissions corresponding to selective absorption in the load. Potential applications include glass melting, glass bending and lamination, thin film drying, and indirect heating. Xiong et al. (1993) described a porous ceramic surface combustor-heater with a built-in heat exchanger for improving the thermal efficiency.⁶² Ruiz and Singh (1993) described an advanced infrared burner.⁶³ The new burner design has considerably higher radiant outputs than the previous style design. The new burner increased a powder paint drying process by 40% and a paper drying process by 200%. Severens et al. (1995) modeled porous radiant burners.⁶⁴ The one-dimensional model results compared favorably with experimental data for the radiant fraction as a function of the gas velocity through the burner. Bogstra (1998) described a new type of infrared radiant burner, referred to as CHERUB, with a closed surface to separate the combustion exhaust gases from the product being heated.⁶⁵ The burner had an enclosed, flat, multi-burner system that heats a ceramic radiant plate. The burner was reported to have a radiant efficiency of 80%, compared to an efficiency of 40% for conventional high-temperature radiant burners.

As previously discussed, one of the limitations of porous refractory burners is the burner surface temperature. If the firing rate density is to be increased, new refractory materials are needed that can withstand continuous operation at higher temperatures, while maintaining their integrity during thermal cycling. One example of research to develop improved radiant burner materials is by

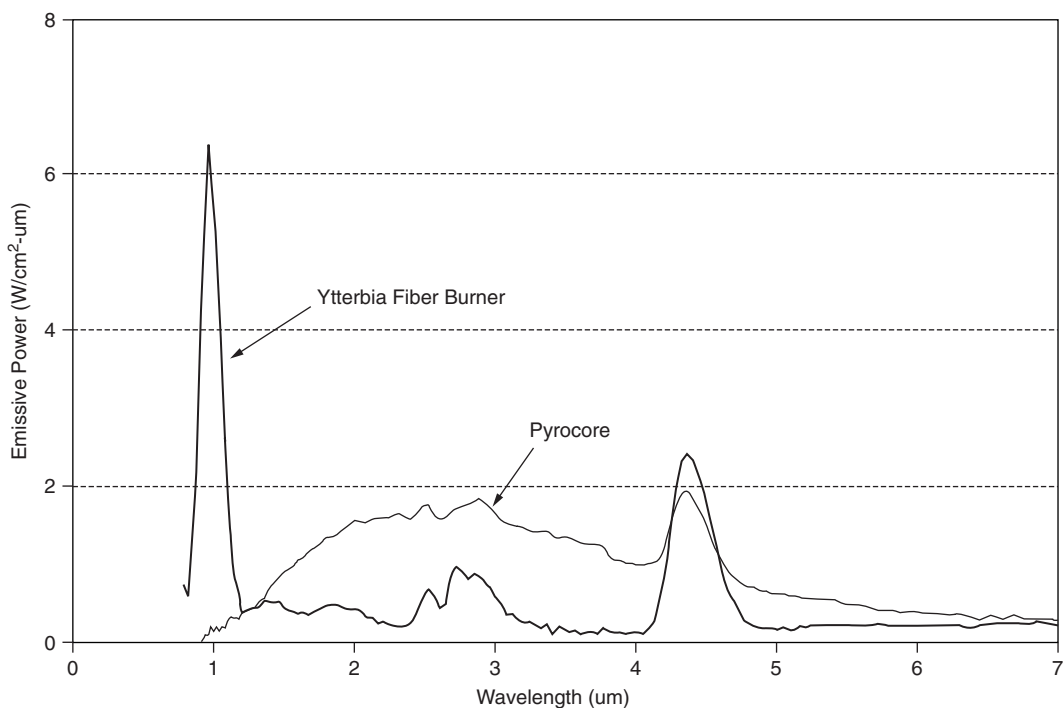


FIGURE 6.16 Emission spectra of ytterbia fiber burner compared to Alzeta (Santa Clara, CA) Pyrocore burner, both operating at 127,000 Btu/hr-ft² (400 kW/m²). (Courtesy of The Gas Research Institute.⁶¹)

Superkinetic, Inc. (Albuquerque, NM) with funding from the U.S. Dept. of Energy.⁶⁶ Three single-crystal ceramic fibers were produced and two fiber materials were successfully made into felt for testing as radiant burner screen surfaces. The materials were alpha-alumina and alpha-silicon carbide, which were successfully bonded with a high-temperature ceramic to form the burner screens that were 95% porous. The purpose of this project was to develop materials capable of radiant burner service near 3000°F (1900K), compared to conventional radiant burner surface temperatures of about 1800°F (1300K). The new materials performed well in actual burner operation, but more research was recommended.

6.3.5 RADIANT WALL BURNERS

A typical radiant wall burner is shown schematically in [Figure 6.17](#). This type of burner is commonly used in process heaters where they heat a refractory wall that radiates heat to tubes parallel to the wall.⁶⁷ The tubes contain a fluid, typically a hydrocarbon, which is being heated. These burners are similar to flame impingement radiant burners except that the flame in a radiant wall burner is directed along the wall and not at the wall or burner tile in the case of impingement burners. The object of a radiant wall burner is to distribute heat as evenly as possible over a fairly wide area. The impingement burner primarily heats its burner tile, which then radiates to the load. This burner is discussed in more detail in [Chapter 15](#).

6.3.6 RADIANT TUBE BURNERS

In some heating processes, it is not desirable to have the products of combustion come in contact with the load. One example is in certain types of heat treating applications where the exhaust gases from a combustion process could contaminate the surface of the parts being heated. In those cases,

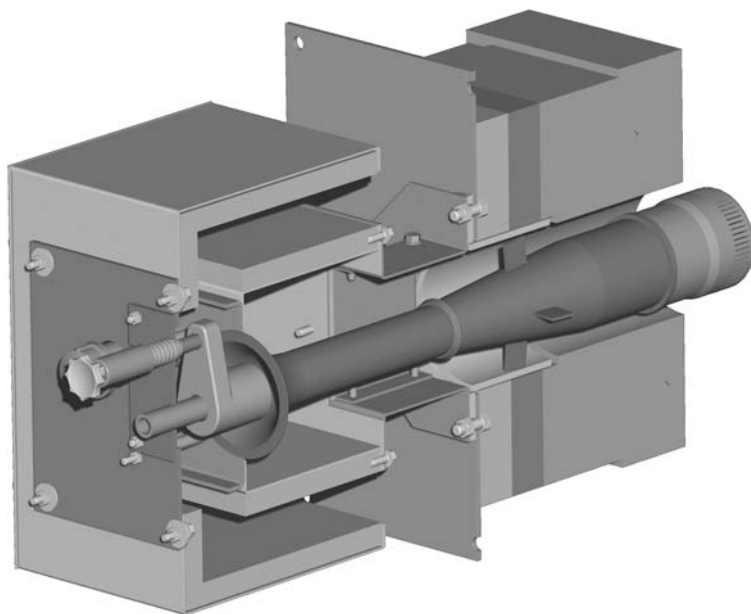


FIGURE 6.17 Typical radiant wall burner. (Courtesy of John Zink Co. LLC.³)

an indirect method of heating is needed. Electric heaters are sometimes used; however, the energy costs are considerably higher than the cost of fossil-fuel-fired heaters. The burner of choice for indirect heating is typically a radiant tube (see [Chapter 14](#)).

The objective of the radiant tube burner is to efficiently transfer heat from the combustion gases to the radiant tube, and then to efficiently radiate that energy to the load. There are several challenges that need to be considered when using radiant tubes. The biggest challenge is the material of construction for the tube itself. Typical tubes are constructed of high-temperature metal alloys or ceramics. Metal alloys can be expensive and typically do not have as high a continuous operating temperature as ceramics. Metal tube burners typically operate at temperatures around 2000°F (1400K). These tubes can fail due a variety of metallurgical problems related to the high-temperature operation and thermal cycling of the burners. Metal tubes are more commonly used than ceramic tubes and research continues into higher temperature metals for use in metallic radiant tube burners.^{68,69} Ceramic tubes tend to have higher temperature operating limits compared to the metal tubes, but are even more susceptible to thermal shock. Research also continues on new ceramic materials for radiant tube burners.^{70–72} There may also be problems joining the ceramic tubes to the metal burner body due to the differences in thermal expansion that can cause the ceramic tubes to crack. The Institute of Gas Technology (Chicago, IL) is working on a new composite material for radiant tubes that is silicon carbide based and has a working temperature that exceeds 2450°F (1620K).⁷³ These new tubes have excellent shock resistance and are capable of heat flux rates up to 150 Btu/hr-in.² (68 kW/m²), compared to rates of 55 Btu/hr-in.² (25 kW/m²) for metallic radiant tubes. Tube lives can be increased by as much as three times with the new tubes. The Gas Research Institute (1998) has funded many projects related to the use of advanced ceramic composite materials for radiant tube burners.⁷⁴ The GRI estimated that about 40% of the more than 50,000 heat treating furnaces are indirectly heated using an estimated 250,000 radiant tube burners. The new ceramic composites promise higher operating temperatures and longer lives.

There are three common configurations for radiant tube heaters. The first and simplest is known as a straight-through tube, where there is a burner at one end of a tube with the exhaust gases from

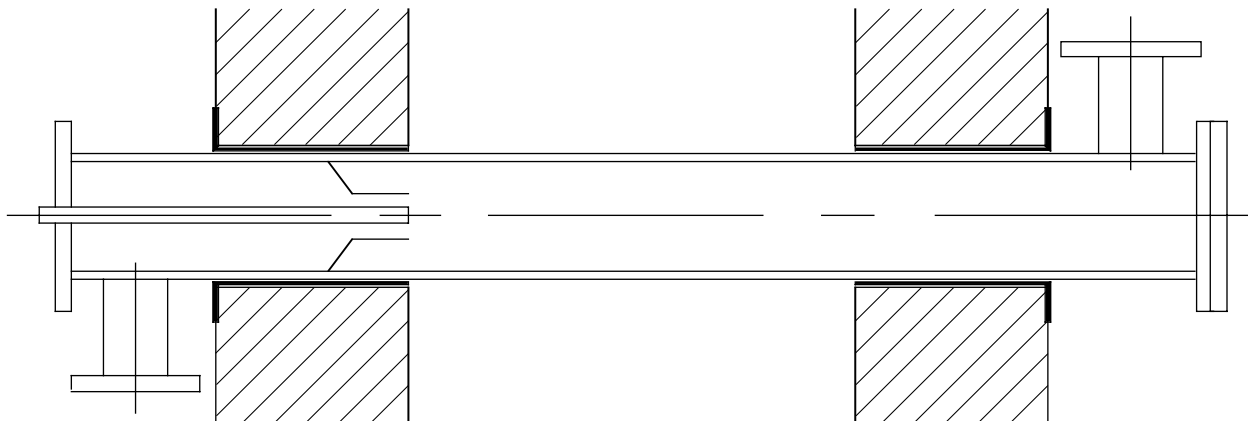


FIGURE 6.18 Straight-through radiant tube burner. (From Ceramic Single Ended Recuperative Radiant Tube (Phase 1), Final Report, NTIS PB91222554, 1990.⁷⁰)

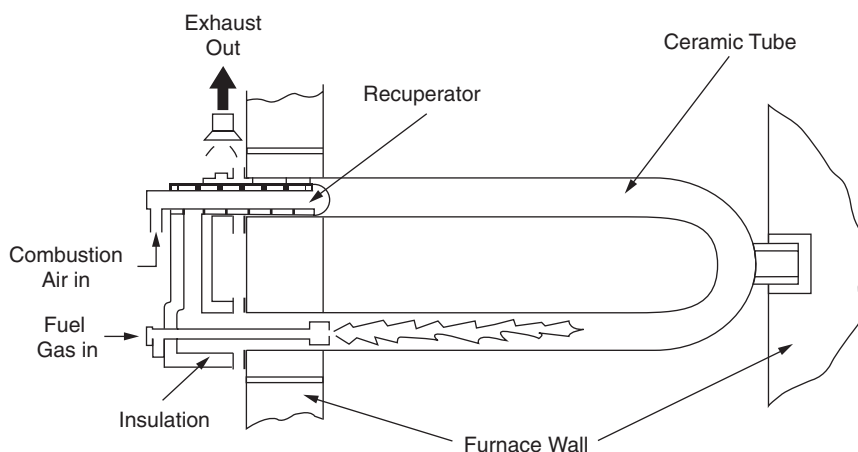


FIGURE 6.19 U-tube radiant tube burner. (Courtesy of American Society of Metals.⁶⁸)

the burner traveling through the tube and exiting at the other end (see Figure 6.18). The challenge when using this geometry is to get efficient heat transfer from the flame gases to the tube, which then radiates to the load. Another potential disadvantage of this type compared to the other two types is that connections for the supply gases are on one side, while the connections for the exhaust gases are on the other side. The second type of radiant tube heater is known as a single-ended recuperative burner. Here, the burner and exhaust are located on the same side of the tube. The burner fires down an inner tube and then returns back to the starting end through an outer annulus. The exhaust gases are used to preheat the incoming fuel and oxidizer, which increases the overall efficiency. A potential difficulty with this design is that the tube is cantilevered from the wall, which puts some additional stress on the tube. Louis et al. (1998) described a single-ended recuperative radiant tube burner for use in a steam-cracking ethylene production plant.⁷⁵ The third radiant tube design is known as a U-tube because of its shape (see Figure 6.19). Again, the supply inlets and exhaust gas outlet are on the same side. One of the difficulties of this design is making a single monolithic U-tube that avoids the need for the 180° elbow where leaks and failures might occur. Abbasi et al. (1998) described an advanced, high-efficiency, low-emissions U-tube radiant tube burner made of either a metal alloy or a ceramic composite.⁷⁶ The new burner promises to have better temperature uniformity and lower pollutant emissions than conventional burners by using internal exhaust gas recirculation.

The heat transfer from radiant tube burners is important because it not only affects the energy transfer to the load, but also the life and performance of the radiant tube. If the heat flux profile along the radiant tube is highly nonuniform, with high- and low-temperature regions along the length of the tube, then the life of the tube will be significantly reduced due to the high thermal stresses and possible overheating. A more uniform heat flux from the tube will give a higher tube life and is normally also more desirable for the load as well to produce uniform heating. The fuel burn-up in the tube is needed to calculate the heat flux from the tube. Ramamurthy et al. (1997) developed fuel burn-up and wall heat transfer correlations for gas-fired radiant tube burners with turbulent flows inside the tube.⁷⁷ They studied several parameters, including the burner area ratio (ratio of the outer annular air inlet port area to the inner circular fuel port inlet area), excess combustion air ratio, combustion air preheat temperature, and the fuel firing rate. The fuel burn-up ratio increased for both high and low burner area ratios, where there was a higher difference between the fuel and combustion air velocities compared to the base case burner area ratio where the velocities were more similar. The fuel burn-up ratio increased with the excess air ratio and with

the air preheat temperature, but decreased as the fuel firing rate increased. The following correlation was developed for the fuel burn-up:

$$\kappa(z) = 1 - \exp \left[-8.24 \times 10^{-4} \frac{A_b^{0.54}}{\phi^{1.53}} \left(\frac{\dot{m}_{\text{air}} \bar{v}_{\text{air}, z=0}}{\dot{m}_{\text{fuel}} \bar{v}_{\text{fuel}, z=0}} \right)^{0.41} \left(\frac{z}{d} \right)^{1.6} \text{Re}_{\text{AFT}}^{-0.08} \right] \quad (6.26)$$

where κ is the fuel burn-up coefficient (fraction of the fuel that has been consumed), z is the distance from the burner outlet, A_b is the burner area ratio (defined above), ϕ is the fuel equivalence ratio, $\dot{m}_{\text{air}}$ is the combustion air mass flow, $\dot{m}_{\text{fuel}}$ is the fuel mass flow, $\bar{v}_{\text{air}, z=0}$ is the average combustion air velocity at the burner outlet, $\bar{v}_{\text{fuel}, z=0}$ is the average fuel velocity at the burner outlet, d is the tube inner diameter, and Re_{AFT} is the Reynolds number of the combustion products at stoichiometric adiabatic flame temperature conditions. They also developed a correlation for the one-dimensional heat flux from the tube to an isothermal bounding wall:

$$q_{\text{rad,1D}} = \frac{\epsilon_w \sigma (\epsilon_{\text{CP}} T_{\text{CP}}^4 - \alpha_{\text{CP}} T_w^4)}{1 - (1 - \epsilon_w)(1 - \alpha_{\text{CP}})} \quad (6.27)$$

where ϵ_w is the emissivity of the wall, σ is the Stefan-Boltzmann constant, ϵ_{CP} is the emissivity of the combustion products, T_{CP} is the absolute temperature of the combustion products, α_{CP} is the absorptivity of the combustion products, and T_w is the absolute temperature of the wall. This correlation was then modified for two dimensions by calculating two factors, depending on the temperature of the combustion products compared to the temperature of the isothermal bounding wall:

$$F_1 = \frac{\int_{z=0}^{z=l_1} q_{\text{rad,2D}} dz}{\int_{z=0}^{z=l_1} q_{\text{rad,1D}} dz} \quad \text{for } T_{\text{CP}} < T_w \quad (6.28)$$

$$F_2 = \frac{\int_{z=l_1}^{z=l_2} q_{\text{rad,2D}} dz}{\int_{z=l_1}^{z=l_2} q_{\text{rad,1D}} dz} \quad \text{for } T_{\text{CP}} > T_w \quad (6.29)$$

where $0 \leq l < l_1$ is the length along the tube where $T_{\text{CP}} < T_w$ and $l_1 \leq l < l_2$ is the length along the tube where $T_{\text{CP}} > T_w$. The average values for F_1 and F_2 were found to be 1.39 and 1.26, respectively. There was excellent agreement between the measured and predicted heat fluxes using the two-dimensional correction factors.

Schultz et al. (1992) reported the use of a single-ended radiant tube burner in a vacuum furnace used for heat treating metals.⁷⁸ The radiant flux from the burner was 150 Btu/hr-in.² (68 kW/m²). Mei and Meunier (1997) tested and modeled the single-ended radiant tube.⁷⁹ They used the Reynolds Stress Model to simulate the burner. The results showed that detailed flame chemistry modeling in the near flame region is important to get an accurate representation of the heat release pattern. The model was in good agreement with the measurements for the total heat flux from the outer tube to the environment, but that the choice of a turbulence model was important for getting agreement on the tube wall temperatures.

There are a number of new developments being made to improve radiant tube burners. One concept involves coating the inside of the tube with a catalyst so that the premixed preheated air

and fuel partially burn on the catalyst surface and thermally conduct that energy to the outside of the tube to radiate to the load.⁸⁰ Advantages include more uniform tube heating and lower NO_x emissions. Further research was recommended to find a platinum catalyst with a higher melting point. Huebner et al. (1986) described the results of tests using oxygen-enriched air to enhance the performance of radiant tube burners.⁸¹ The oxygen content in the oxidizer ranged from 21% (air) to 80%, by volume. The furnace temperatures ranged from 1300 to 2400°F (700 to 1315°C) and radiant tube diameters of 4, 7 $\frac{1}{8}$, and 10 in. (10.2, 18.1, and 25.4 cm) were tested. Large improvements in thermal efficiency and relatively little change in the tube temperature were found. Larger efficiency improvements were found for the larger diameter tubes.

6.4 EFFECTS ON HEAT TRANSFER

There are many variables that affect the heat transfer from flames. In this section, the following effects are briefly discussed: fuel and oxidizer composition and temperature, fuel and oxidizer staging, burner orientation, heat recuperation, and the use of pulse combustion.

6.4.1 FUEL EFFECTS

The fuel can have a significant impact on the heat transfer from flames. The three major types of fuels (solids, liquids, and gases) are considered in this section.

6.4.1.1 Solid Fuels

Solid fuels such as coal and coke are well-known for producing luminous flames because of the particles in the flame that radiate (see [Chapter 2](#)). Solid fuels are not often used in industrial combustion applications and are most used in large power generation plants. The solid particles in solid-fuel combustion radiate as graybodies and give a more uniform spectral radiation output than gaseous flames that tend to have discrete wavelength bands for water and carbon dioxide radiation. The luminous radiation from solid fuel combustion tends to make these flames more efficient, although there are challenges to completing combusting the fuel and handling any residues that are generated from the flame.

Another aspect of solid fuels that indirectly impacts the heat transfer in a combustor is the deposition of ash, produced by the combustion of the solid fuel, onto the inside of the combustor.⁸² These deposits may improve the radiant absorptivity of the materials to which they adhere because the emissivity of the deposit is often high. However, the deposits may have a lower thermal conductivity than the base material, particularly when the substrate is a metal tube containing a fluid to be heated, such as water. Then, the deposits can significantly impede the heat transfer to the load and must be periodically removed. Wall et al. (1994) noted that there can actually be a slight increase in the thermal conductivity of the deposit with time as its physical and chemical nature change, but the conductivity is still considerably less than that of the metal tube.⁸²

Books are available on solid fuel combustion, particularly coal combustion, although most of them have very little on heat transfer and tend to focus more on the physics of the combustion process itself.^{83–90}

6.4.1.2 Liquid Fuels

Boersma (1973) showed that the heat transfer from liquid oil flames can be considerably higher than for comparable gas flames (see [Figure 6.20](#)).⁹¹ This is caused by particle generation in the flames where the particles radiate like graybodies, similar to solid fuel flames. An example of an oil flame is shown in [Figure 6.1](#).

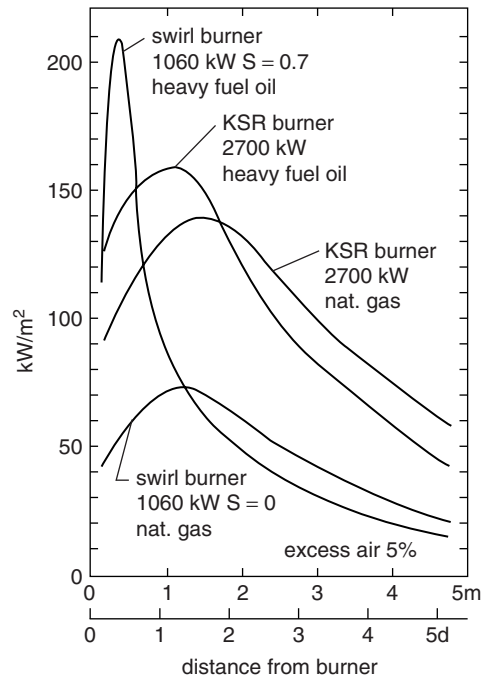


FIGURE 6.20 Measured heat flux rates from different fuels and burners in a cylindrical furnace. (Courtesy of Academic Press.⁹¹)

Zung (1978) has edited a book about the evaporation and combustion of liquid fuels.⁹² However, the book has almost nothing on heat transfer from flames or in furnaces. Williams (1990) has written a book that has a chapter that specifically addresses liquid fuel combustion in furnaces, although there is very little on heat transfer.⁹³

6.4.1.3 Gaseous Fuels

One of the benefits of the gaseous fuels used in industrial combustion applications is that they are very clean-burning and normally generate very few particulates. However, this is a detriment when it comes to heat transfer from flames because gaseous flames are often very nonluminous and may only radiate in a few narrow wavelength bands as discussed in [Chapter 2](#). New burner designs have been introduced in recent years to increase the flame luminosity from natural gas flames. This has improved thermal efficiencies and increased product throughputs. One major way this has been achieved is through staged combustion by making the inner part of the flame closest to the burner to be fuel rich and then to burn out the fuel downstream with staged oxidizer injection. This technique is well-known for NO_x emission reduction as well. Other techniques for increasing the luminosity of gaseous flames include particle injection into the flames, oil injection into the flames, and preheating and cracking the fuel prior to combustion. As discussed in [Chapter 1](#), this continues to be an important area of combustion research.

6.4.1.4 Fuel Temperature

Preheating the fuel influences the flame temperature by increasing the adiabatic flame temperature (see [Figure 6.21](#)). This influences both the radiation and convection from the combustion products to the load. The radiation is often increased because of the higher gas temperature. However, different gas species are often produced at higher temperatures, which also influences the radiant heat flux, especially if more or less soot is produced. Amin et al. (1995) experimentally determined that the radiant fraction from an oxygen enhanced methane flame decreased as the fuel preheat temperature was increased from 300 to 500°C (570 to 930°F).⁹⁴ The measured radiant fraction actually increased

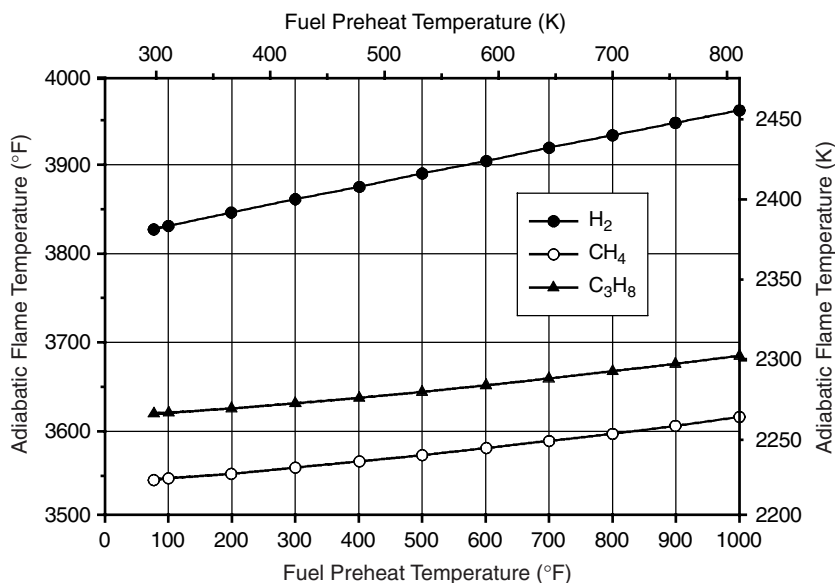


FIGURE 6.21 Adiabatic flame temperature as a function of fuel preheat temperature. (Courtesy of John Zink Co., LLC.³)

slightly by increasing the fuel preheat temperature from 500 to 600°C (930 to 1100°F), due to increased soot production. The convective heat transfer from the combustion products to the load is also influenced by fuel preheating. By preheating the fuel, the transport properties change with both temperature and composition, which directly influence the convective heat transfer coefficient. The gas velocity increases with temperature because of the gas expansion at higher temperatures, which can increase the convective heat transfer to the load.

6.4.2 OXIDIZER EFFECTS

The oxidizer composition and temperature both play important roles in the heat transfer from flames to the load and are briefly discussed next.

6.4.2.1 Oxidizer Composition

This section briefly discusses the effects of the oxidizer composition on the heat transfer from flames. Related discussions are also given in Chapters 2, 20, and 21. Chedaille (1965) showed that an air/oil flame can be enhanced by injecting pure oxygen between the flame and the load.⁹⁵ The experiments were conducted in a tunnel furnace where the load consisted of water-cooled tubes located 7 cm (0.3 in.) below the top of the hearth. The flame was angled, at angles ranging from 16 to 30°, from an end wall down toward the hearth. The total combined oxygen content of the combustion air and lanced O₂ was 30% by volume. The experiments showed that the total heat transfer was highest by injecting the oxygen under the flame, but the heat transfer was more uniform by premixing the oxygen in with the combustion air. Arnold (1967) calculated the heat transfer rates for burners fired on town gas with either air or pure oxygen as shown in Figure 6.22.⁹⁶ The oxy/fuel flames produced heating rates as much as six times higher than the air/fuel flames when the surface temperature of the load was the highest. Kobayashi et al. (1986) showed theoretically how the heat flux to the furnace heat load can be significantly improved using higher levels of O₂ in the oxidizer, as shown in Figure 6.23.⁹⁷ De Lucia (1991) showed that using oxygen to enhance the performance of industrial furnaces can result in some dramatic fuel reductions per unit of output.⁹⁸

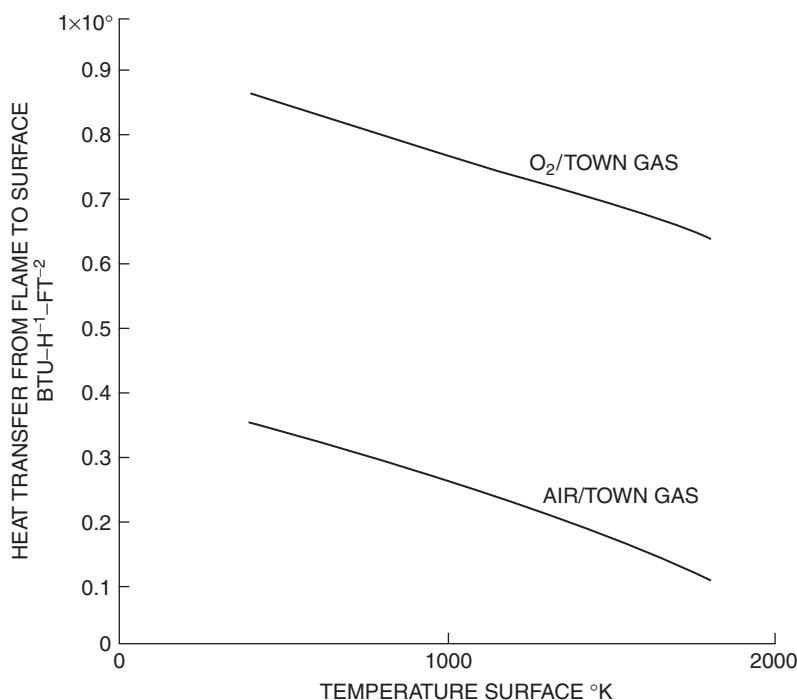


FIGURE 6.22 Calculated heat transfer rates for town gas flames combusted with either air or pure O₂. (Courtesy of Institute of Energy.⁹⁶)

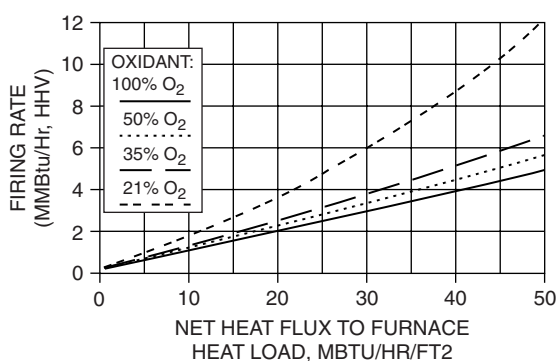


FIGURE 6.23 Calculated heat flux vs. firing rate for different oxidant compositions. (Courtesy of American Society of Metals.⁹⁷)

- 30 to 50% in glass melting
- More than 50% in pig iron melting
- More than 30% in copper alloy production
- As much as 39% in ceramic production

In the study by Baukal and Gebhart discussed above under firing rate considerations, the oxidizer composition was also a parameter of interest.²² Figure 6.24 shows how the heat flux to the target varied as a function of the oxidizer composition and as a function of the geometry. The heat flux intensity increased by 54 to 230% as Ω increased from 0.30 to 1.00. The average improvement was approximately 80%. The largest improvement occurred at $L = 0.5$ and $R_{\text{eff}} = 0.16$. At higher

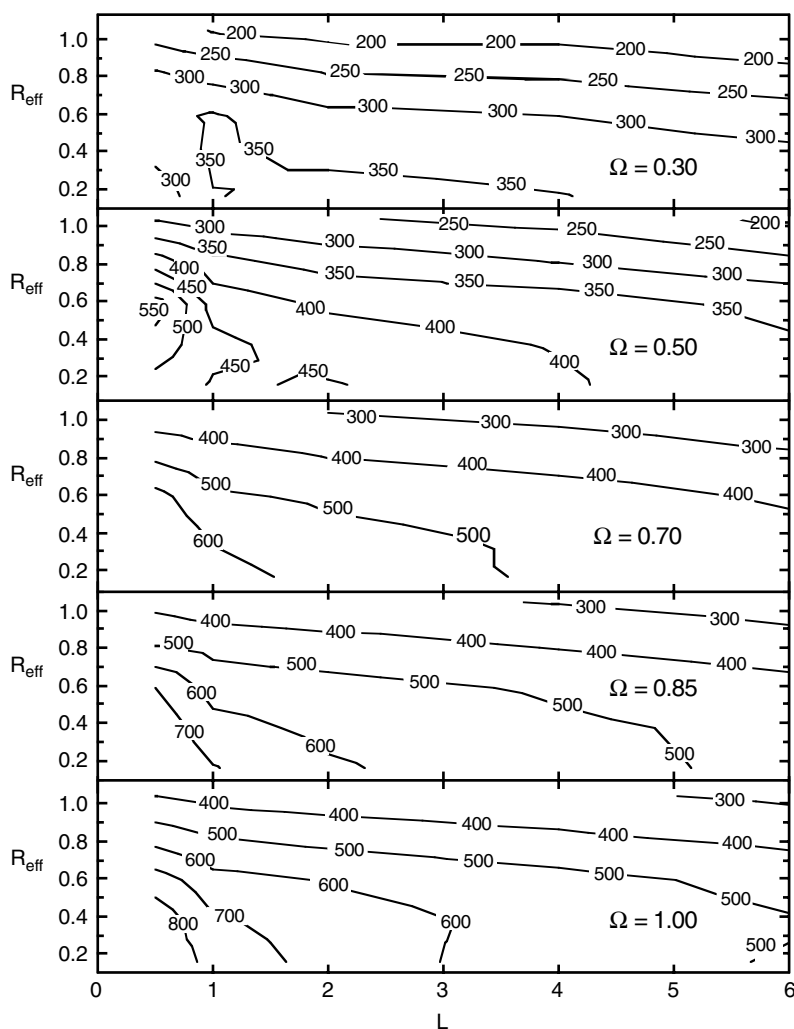


FIGURE 6.24 Contours of the total heat flux, q'' (kW/m²), for stoichiometric natural gas flames ($q_f = 15.0$ kW), with a variable oxidizer composition, impinging on an untreated stainless target.²²

values of Ω , the peak heat flux was measured at the closest axial and radial locations. At lower values of Ω , the peak flux occurred at intermediate axial and radial positions. Therefore, at lower Ω , there was clearly an optimum position to maximize the heat transfer and the thermal efficiency. Figure 6.25 depicts the increase in heat flux intensity as the O_2 content in the oxidizer (Ω) increased. For $L = 0.5$, the heat flux increased by 192% as the O_2 in the oxidizer increased from $\Omega = 0.30$ to $\Omega = 1.00$. As L increased, the oxidizer composition had less influence on the heat flux intensity. The slope of each curve increased as L decreased. Therefore, the oxidizer composition was more important for closer axial spacings. Figure 6.26 shows that the thermal efficiency increased with Ω . This was a consequence of removing the diluent N_2 from the oxidizer. The efficiency decreased with L . The shapes of the curves are similar.

6.4.2.2 Oxidizer Temperature

Preheating the oxidizer is commonly done to recover energy from the exhaust products and to increase the adiabatic flame temperature of the flame as shown in Figure 6.27. Preheating the

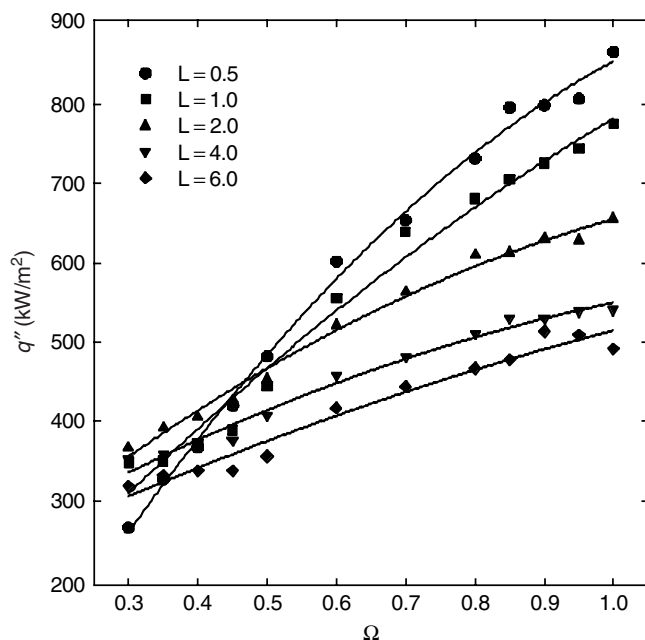


FIGURE 6.25 Total heat flux (q'') to the stagnation point ($R_{\text{eff}} = 0.16$) of an untreated stainless target, for stoichiometric natural gas flames ($q_f = 15.0$ kW) with a variable oxidizer composition (Ω).²²

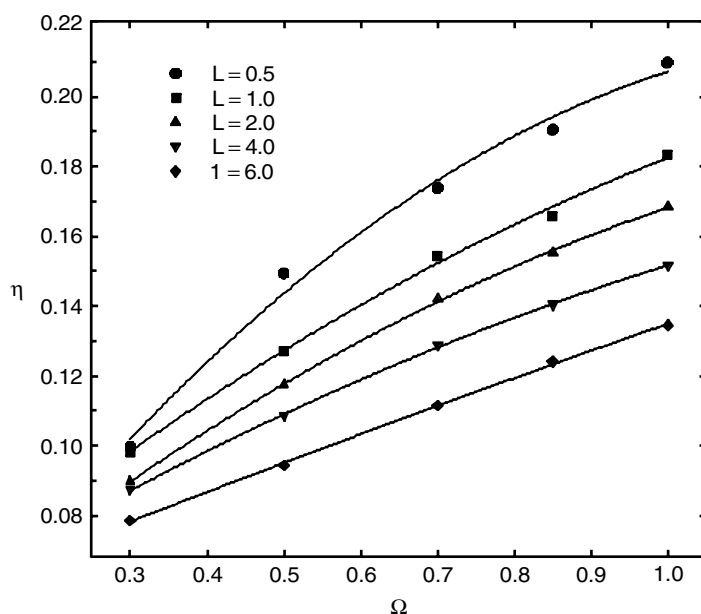


FIGURE 6.26 Thermal efficiency (η) for stoichiometric natural gas flames ($q_f = 15.0$ kW), with a variable oxidizer composition (Ω), impinging on the stagnation point ($R_{\text{eff}} = 0.16$) of an untreated stainless target.²²

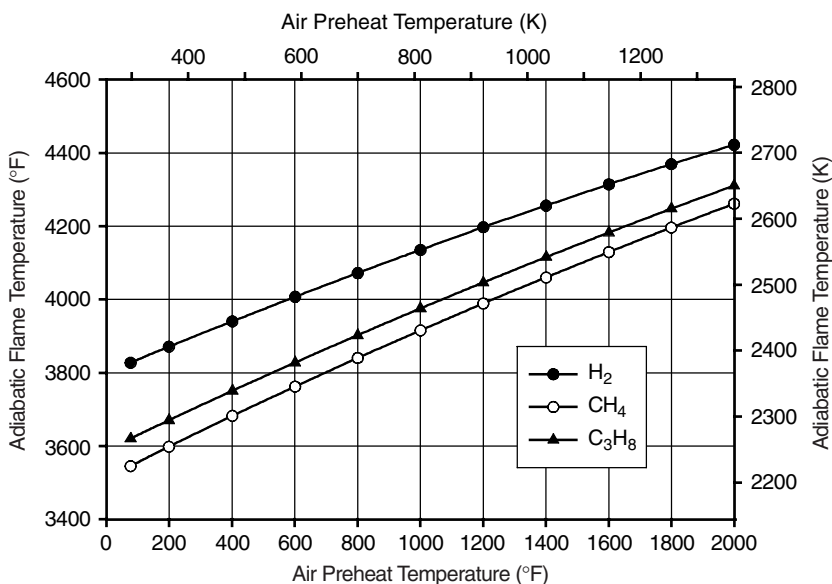


FIGURE 6.27 Adiabatic equilibrium flame temperature vs. air preheat temperature for stoichiometric air/H₂, air/CH₄, and air/C₃H₈ flames. (Courtesy of John Zink Co., LLC.³)

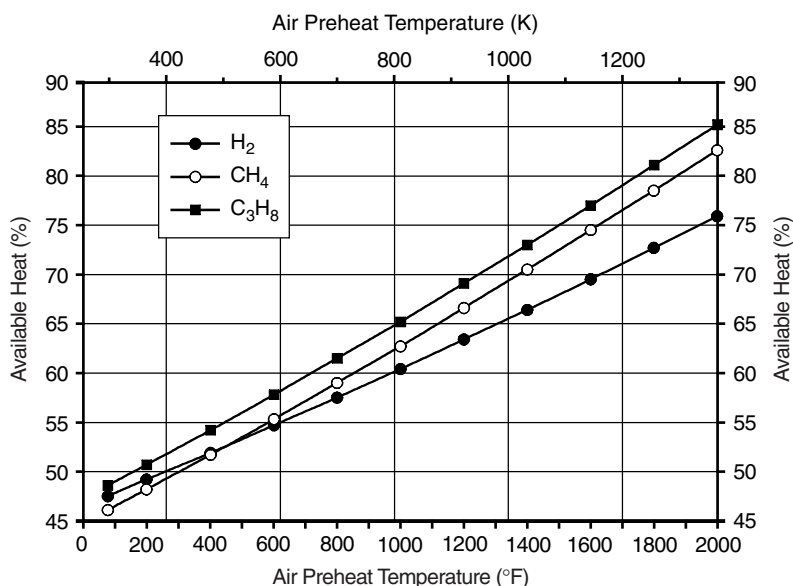


FIGURE 6.28 Available heat vs. air preheat temperature, for equilibrium stoichiometric air/H₂, air/CH₄, and air/C₃H₈ flames at an exhaust gas temperature of 2000°F (1366K). (Courtesy of John Zink Co., LLC.³)

oxidizer also improves the thermal efficiency of a process as shown in Figure 6.28. Higher flame temperatures can dramatically increase the radiant heat flux from the flame because of its dependence on the absolute temperature raised to the fourth power. Guénebaud and Gaydon (1957) studied the effect of air preheating on the shape and radiant output from methane, bunsen-burner-type flames.⁹⁹ As shown in Figure 6.29, preheating the air to 550°C (1020°F) caused the flame to become

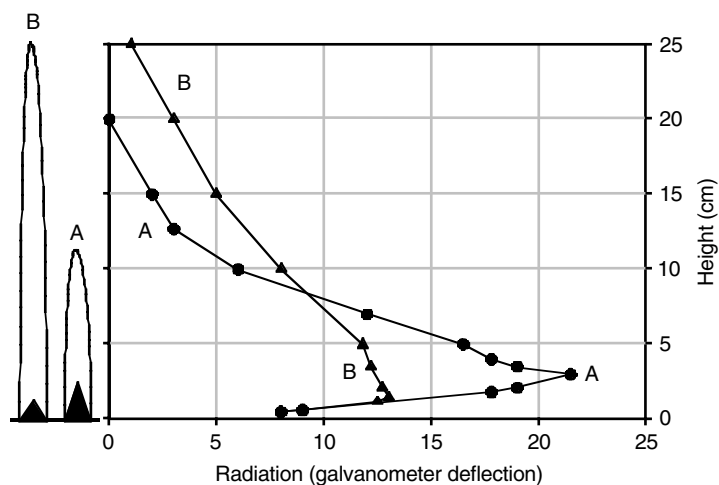


FIGURE 6.29 Flame shapes and radiant heat flux from a methane flame produced by a bunsen burner: (A) no air preheating and (B) air preheated to 550°C (1020°F). (Courtesy of The Combustion Institute.⁹⁹)

twice as long and the radiant heat flux profile to become more uniform compared to the burner with no air preheat.

6.4.3 STAGING EFFECTS

Staging the fuel¹⁰⁰ or the oxidizer^{101,102} are common techniques for reducing NO_x emissions. This staging also impacts the heat transfer of the system, which is briefly considered next. In either type of staging, the flame is usually longer than unstaged flames. It is assumed in the discussions for both types of staging that both the furnace geometry and the heating process permit longer flames without either flame impingement on the walls of the combustor or without damaging the product quality.

6.4.3.1 Fuel Staging

An example of fuel staging is shown in [Figure 6.30](#). Staging the fuel normally means that the inner flame region has excess oxidizer or is fuel lean. Fuel-lean flames tend to be very nonluminous, depending on the mixing and, therefore, generate only gaseous radiation, with little or no soot formation. The balance of the oxidizer is added downstream of the main flame region and normally brings the overall combustion process from very fuel-lean conditions to slightly lean conditions. Again, this does not favor luminous flame radiation even in the secondary flame region. Therefore, fuel staged flames tend to be nonluminous and the heat transfer from these flames is more dominated by forced convection compared to luminous flames. Depending on the application, this may not only be acceptable, but desirable. If putting too much heat near the beginning of the flame can cause overheating, then fuel staging may be an option for stretching out the heat flux over a longer length, while simultaneously reducing the heat flux near the base of the flame. An example would be a counterflow rotary kiln. The material enters the kiln at one end and the burner is located at the other end. This means that the processed material exits at the burner end. Too much heat applied at the material exit could cause slagging, which is partial melting and agglomeration. Pushing the flame more toward the material feed end puts more of the heat where there is much less chance of overheating.

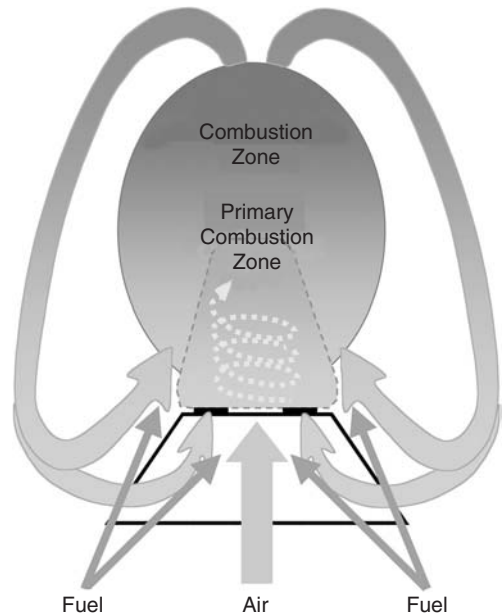


FIGURE 6.30 Schematic of fuel staging. (Courtesy of John Zink Co., LLC.³)

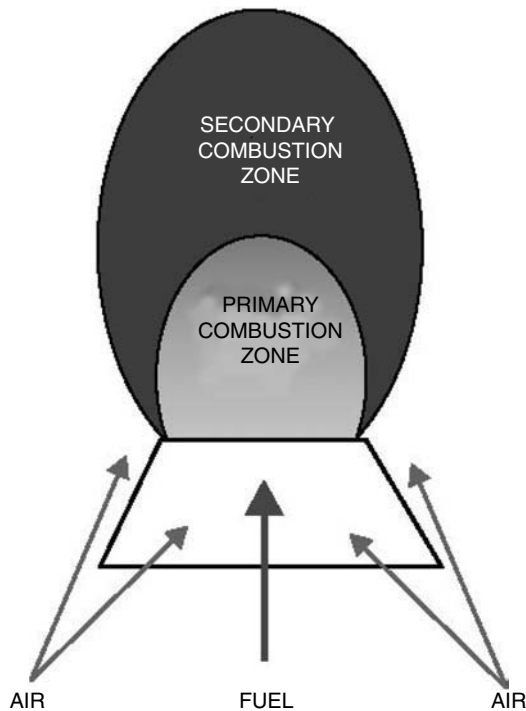


FIGURE 6.31 Schematic of air staging. (Courtesy of John Zink Co., LLC.³)

6.4.3.2 Oxidizer Staging

A schematic of air staging is shown in Figure 6.31. In this type of flame, the main flame region closest to the burner does not have enough oxygen and is fuel rich. This normally produces soot and therefore a luminous flame. This type of flame has a higher heat flux from the flame, although the overall efficiency of the process is still dependent on the system geometry and any heat recovery

devices that have been incorporated into the system. A staged oxygen flame commonly has higher heat flux near the burner outlet than the staged fuel flame. Again, depending on the application, this may be desirable or undesirable. In the counterflow rotary kiln process discussed above, this would be undesirable and could lead to slagging. However, for a co-flow rotary kiln, an initially fuel-rich and luminous flame is often desirable because the incoming cold/wet material can absorb higher heat fluxes without affecting product quality. In the staged oxidizer flame, the secondary flame zone combusts the unburned fuel coming out the primary, fuel-rich, flame region to avoid emitting particulates or CO from the exhaust products. This secondary flame zone helps lengthen the flame and the resulting heat flux profile, compared to an unstaged flame.

6.4.4 BURNER ORIENTATION

The choice of burner orientation can vary widely, depending on the industry. Some examples will suffice to illustrate. In the metals and minerals industries, the burners are usually mounted in the side wall and either fire parallel to or angled toward the materials being heated. In the heat treating industry, the burners may fire over or at a muffle that separates the combustion products from the materials being processed. In the petrochemical industry, a more common configuration is for the burners to be mounted in the floor of the furnaces and fire vertically upward. Several common configurations are briefly discussed in this section to show their impact on the heat transfer in the process.

6.4.4.1 Hearth-Fired Burners

A diagram of a hearth-fired burner is shown in Figure 6.32. Pictures of hearth-fired burner are shown in Figure 6.1 and Figure 6.2. These burners are commonly used in the petrochemical industry and are often natural draft (see Chapter 16). Depending on the heater design, the burners can be arranged in a variety of ways. A smaller cylindrical heater may have a single burner in the middle of the floor, while larger cylindrical heaters usually have multiple burners located at some radius from the centerline, arranged at equal angular distances apart (see Figure 6.33). Rectangular heaters generally have one or more rows of burners in the floor.

The flames from hearth-fired burners are normally directly vertical, in line with the natural buoyancy force. These burners are designed to release the majority of their heating value in the lower part of the furnace or heater in what is usually referred to as the radiant section. Many of these heaters have a convection section near the top, where further heat is released from the combustion products to the tubes in the convection section.

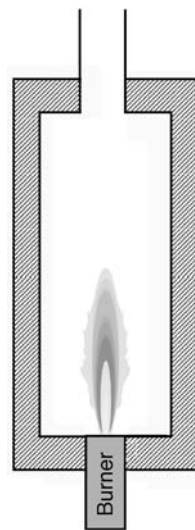


FIGURE 6.32 Elevation view of a hearth-fired burner configuration. (Courtesy of John Zink Co., LLC.³)

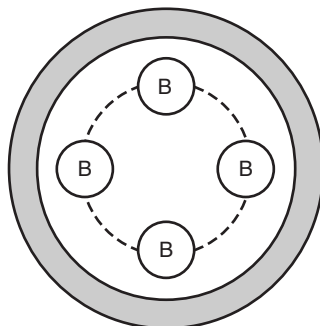


FIGURE 6.33 Cross section of a vertical cylindrical furnace with multiple burners. (Courtesy of John Zink Co., LLC.³)

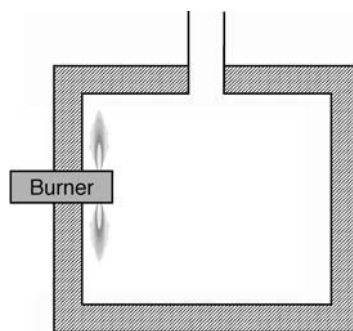


FIGURE 6.34 Elevation view of a perpendicular wall-fired burner configuration. (Courtesy of John Zink Co., LLC.³)

Often, the vertical heat flux profile from hearth-fired burners to the tubes in the radiant section is an important design consideration. An important heat transfer consideration for this type of burner is the distance from the burners to the tubes. If the spacing is too close, then the tubes can fail prematurely due to overheating. If the spacing is too far, then not enough heat will be transferred in the radiant section and too much can be transferred in the convection section. The former reduces productivity, while the latter can cause damage to the tubes in the convection section. The spacing between burners is also important in order to maximize the power density without causing adverse interactions between flames.

6.4.4.2 Wall-Fired Burners

There are a variety of wall-fired burner configurations. A diagram of a perpendicular wall-fired burner is shown in [Figure 6.34](#). A schematic is shown in [Figure 6.17](#). The flame comes out in all angular directions, flowing radially outward from the burner. A variation of this type of burner that is sometimes considered to be a wall burner is an infrared burner (see [Chapter 13](#)). In that case, the burner tile (sometimes referred to as a block or quarl) is heated by flames and radiates toward the load. A diagram of a parallel wall-fired burner is shown in [Figure 6.35](#). That is a variation of the hearth-fired burner where the burner is located next to and fires along a vertical wall. A schematic of a terrace-fired burner is shown in [Figure 6.36](#). The schematic shows a single terrace, but furnaces using this technology usually have multiple terraces along the vertical wall. There, the combustion products from lower terrace burners also flow against upper terraces.

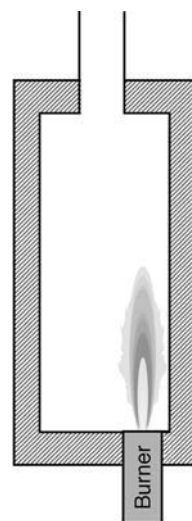


FIGURE 6.35 Elevation view of a parallel wall-fired burner configuration. (Courtesy of John Zink Co., LLC.³)

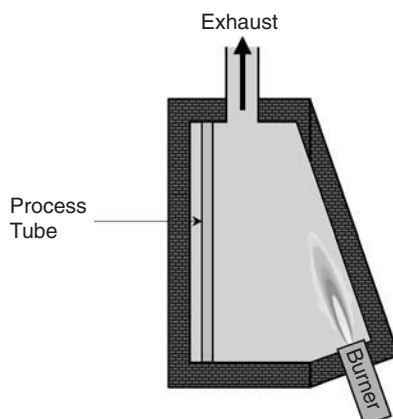


FIGURE 6.36 Elevation view of a terrace-fired burner configuration.



FIGURE 6.37 Example of a down-fired, forced-draft, preheated air burner. (Courtesy of John Zink Co., LLC.³)

The basic principle of wall-fired burners is to heat a refractory wall, which then radiates to the load. It is often important to have fairly uniform heat flux to the walls to prevent overheating the refractory, which could cause the walls to fail prematurely. Uniform heating of the wall is often desired to uniformly heat the load. In vertical wall-fired burners used in the petrochemical industry, there may be a desired heat flux profile along the wall to optimize the heat transfer to the tubes in the radiant section.

6.4.4.3 Roof-Fired Burners

A picture of a roof- or down-fired burner is shown in [Figure 6.37](#). These burners are used in a variety of applications. One application is to supplement electric arc furnaces. There, burners are often mounted in the roof or upper side wall of an electric arc furnace (EAF) and fire down onto the scrap metal (see [Figure 6.38](#)). The flames may impinge directly on the scrap. Another application is in reforming furnaces where the burners are often natural draft. In this type of configuration, the flow of the combustion gases is against the buoyancy force. This can affect the flame shape if there is not adequate draft in the furnace. This configuration is not used in many applications because of the difficulties of mounting burners on the roof of a furnace.

An important heat transfer consideration with this design is to ensure that the roof is not overheated, which could become a safety issue if the hot combustion products leaked through the roof where they could damage the burner or the fuel gas piping. Another challenge is to get the proper heat flux distribution with the desired flame length, especially in a furnace like a down-fired reformer.

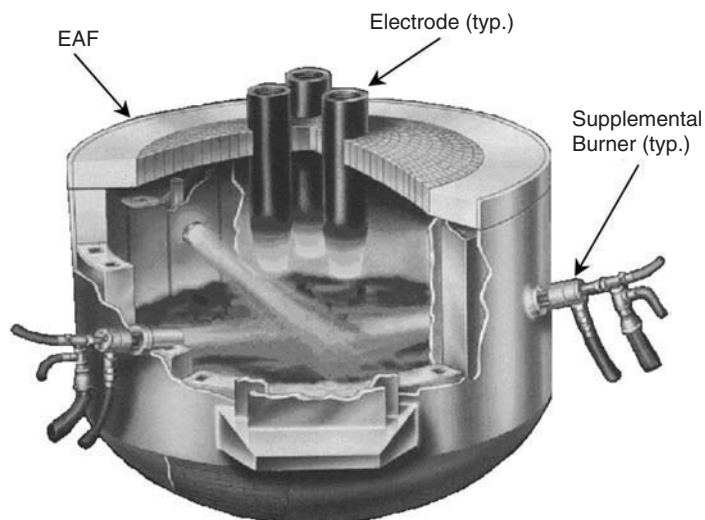


FIGURE 6.38 Supplemental burners firing from the roof of an electric arc furnace.

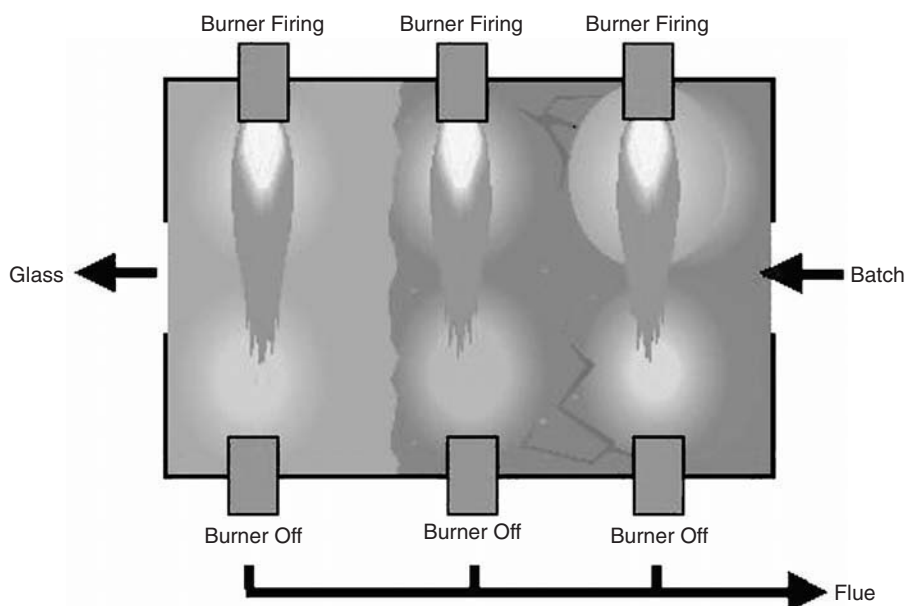


FIGURE 6.39 Example of a side-fired regenerative glass furnace.

6.4.4.4 Side-Fired Burners

This is a common configuration in a wide range of applications. An example is shown in Figure 6.39. The burners are mounted on the side walls and fired perpendicular to those walls, in contrast to wall-fired burners which fire along the wall. Side-fired burners often fire parallel to and over top of the load. The location of the burners on opposing side walls may be either directly opposed, or staggered, depending on the furnace design. Low-momentum flames are often more luminous and have lower NO_x emissions compared to high-momentum flames. However, there are potential problems with

lower momentum flames. Buoyancy effects can lift the flames toward the roof and away from the load. This may reduce the roof refractory life and reduce the heating efficiency to the load. Another problem is that lower momentum flames are also more easily disturbed by neighboring flames. Therefore, the burner design is a balance of many factors affected by flame momentum.

Traditional side-fired burners were and, in many cases, still are round flame designs. A relatively newer side-fired burner is a rectangular design with a wide, fan shape, often referred to as a flat flame burner (e.g., Figure 20.24). The wide part of the flame is parallel to the load below it, to maximize the flame radiation surface area, which maximizes the heat transfer to the load. It is usually desirable to have a fairly uniform heat flux output from the flame and to minimize hot spots in the flame. The flame length and surrounding burners also play important roles in determining the heat flux to the load.

6.4.5 HEAT RECUPERATION

There are two strategies that are commonly used to recuperate heat from the combustion exhaust products. The first strategy is to use those hot gases to preheat an incoming feed stream to the combustor, such as the incoming fuel, oxidizer, both the fuel and the oxidizer, or the raw materials being processed in the combustor. In this strategy, a heat exchanger is used so that the hot exhaust gases do not contact the material they are preheating. This is commonly done using either recuperative or regenerative techniques, which are discussed below. Freeman (1986) compared these two techniques for an innovative air preheating system for use in glass melting furnace, where, in that case, the regenerative system performed slightly better than the recuperative systems, as shown in Figure 6.40.¹⁰³ The second strategy for recuperating heat from the hot exhaust gases is to actually blend them into a feed stream coming into the furnace. This is commonly done by either furnace or flue gas recirculation.

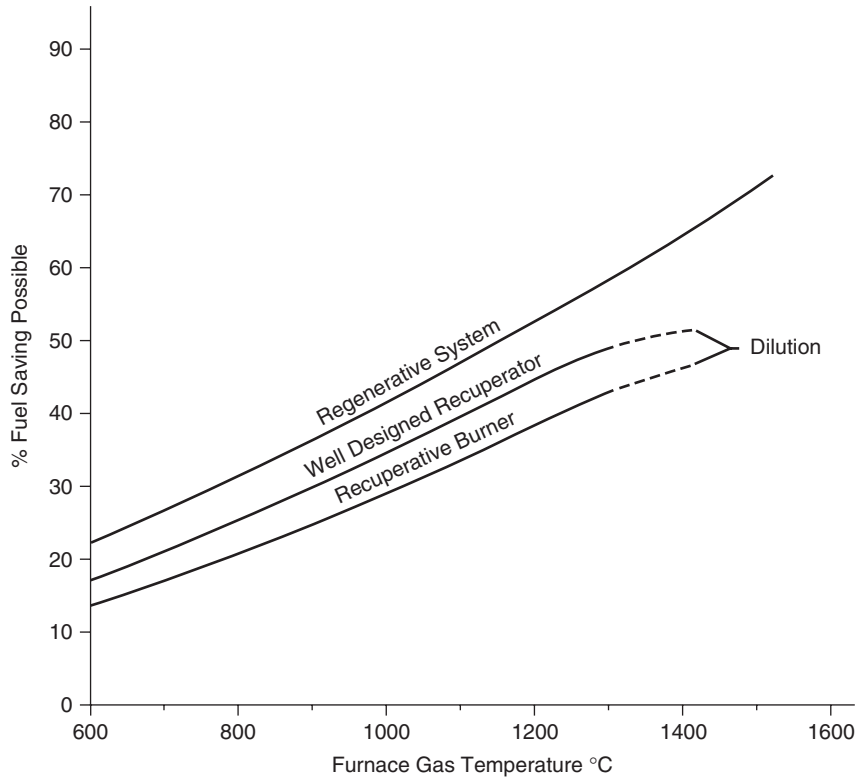


FIGURE 6.40 Calculated fuel savings from regenerative and recuperative systems in a glass-melting furnace. (Courtesy of American Society of Metals.¹⁰³)

There are two primary reasons for recuperating the sensible energy in those exhaust products. The first is simply to improve the thermal efficiency of the combustion system. The increase in the available heat for the stoichiometric combustion of methane by preheating the incoming combustion air is shown in Figure 6.28. The second reason that heat may be recuperated is to increase the flame temperature for processes that need higher temperatures, such as in the melting of raw materials like sand to produce glass. Figure 6.27 shows the adiabatic flame temperature for a stoichiometric air/methane flame increases as the combustion air temperature increases.

Heat recuperation is frequently done using heat exchangers and does not involve the burners at all. However, there are also systems that incorporate heat recuperation in the burner itself. This heat recuperation can have a significant impact on the heat transfer from the flame to the load, which is briefly discussed next for three types of recuperative burners. Brooks and Winters (1990) discussed the use of cellular ceramic materials (CCMs) to recuperate heat in a furnace.¹⁰⁴ Combustion products from burners flow through the CCM, which is separate from the burner and which then radiates energy to the furnace. This is not discussed further here as it is not commercially popular at this time, but it may have merit for future applications.

6.4.5.1 Regenerative Burners

Regenerative burners (see Chapter 12) are designed to fire intermittently, wherein half of the burners are on at any given time while half are off. The exhaust products from the burners that are firing flow through the burners that are off. The burners contain some type of heat storage material, usually some type of porous ceramic or ceramic beads, that remove much of the heat from the hot exhaust products flowing through them. After a certain amount of time, usually on the order of 0.5 to 15 minutes, the firing pattern is reversed. The combustion air flows through the heat storage material and is preheated prior to combustion. As previously shown, the preheated air can significantly raise the flame temperature. Although the level of dissociation may increase in the flame because of the higher temperatures, any dissociated species normally exist in trace quantities so that the bulk composition of the flame gases is essentially the same with or without preheating (see Table 6.3). The main difference is the flame temperature, which affects the radiation from the flame.

TABLE 6.3
Adiabatic Equilibrium Combustion Products
(Mole Fraction) for the Stoichiometric
Combustion of Methane With and Without Air
Preheating

Species	Combustion Air Temperature	
	77°F	2000°F
CO	0.012	0.041
CO ₂	0.103	0.071
H	0.000	0.005
H ₂	0.003	0.011
H ₂ O	0.148	0.129
NO	0.002	0.008
N ₂	0.721	0.699
O	0.000	0.004
O ₂	0.006	0.018
OH	0.003	0.014
Adiabatic Flame Temp. (°F)	3620	4311

Source: Courtesy of CRC Press.¹

Davies (1986) described the use of a regenerative burner for radiant tubes for heating galvanized steel coils.¹⁰⁵ The cycle frequency for switching the firing between burners was 20 seconds. Fuel savings of more than 50% were reported. Newby (1986) reported on an all-ceramic, high-temperature regenerative burner for use in applications requiring temperatures up to 3000°F (1650°C).¹⁰⁶ The burner has been proven to operate in dirty environments without plugging the regenerators or causing corrosion damage in the burner. Examples were given of its use in an open pot glass melting furnace with a reported fuel savings of 20%, an argon-oxygen decarburization vessel preheater, and in (5) small aluminum melting furnaces. Morita (1996) described Japanese research into high-temperature regenerative air preheaters capable of air temperatures exceeding 1000°C (2100°F).¹⁰⁷ Katsuki and Hasegawa (1998) discussed a regenerative preheater for industrial burners capable of air preheats above 1300K (1900°F).¹⁰⁸ Takamichi (1998) described the use of regenerative burners for improving the performance of forging furnaces.¹⁰⁹ Energy savings of up to 40% were reported.

6.4.5.2 Recuperative Burners

In a recuperative burner, a heat exchanger is built into the burner so that the hot combustion products are exhausted through the burner where they preheat either the fuel, the oxidizer, or both. This is a challenging design due to the large temperature and pressure differences in the various streams. Using heat recuperation in the burner has some advantages over using an external heat exchanger for recovering energy from the furnace, which requires a significant amount of large, insulated ductwork. However, the heat exchanger built into the burner may be less efficient than an external exchanger and therefore tends to have lower preheat temperatures. As discussed above, preheating the incoming fuel or oxidizer raises the flame temperature, which enhances flame radiation.

Singh et al. (1986) reported on the use of a recuperator for single-ended radiant tube burners with a mullite outer tube and silicon carbide inner recuperator.¹¹⁰ The heat flux rates from the tube ranged from 55 to 69 Btu/hr-in.² (25 to 31 kW/m²) and had exhaust gas temperatures up to 2000°F (1100°C). Flamme et al. (1998) compared several different types of ceramic heat exchangers for use in self-recuperative burners.¹¹¹ They defined the combustion efficiency (η) as:

$$\eta = \frac{h_{\text{fuel (LHV)}} + h_{\text{air}} - h_{\text{flue gas (inlet)}}}{h_{\text{fuel (LHV)}}} \quad (6.30)$$

where $h_{\text{fuel (LHV)}}$ is the lower heating value of the fuel, h_{air} is the enthalpy of the preheated combustion air, and $h_{\text{flue gas (inlet)}}$ is the enthalpy of the flue gas at the inlet of the recuperator. The relative air preheat rate (ϵ) was expressed as:

$$\epsilon = \frac{T_{\text{air (outlet)}} - T_{\text{air (inlet)}}}{T_{\text{flue gas (inlet)}} - T_{\text{air (inlet)}}} \quad (6.31)$$

where $T_{\text{air (outlet)}}$ is the combustion air temperature at the outlet of the recuperator, $T_{\text{air (inlet)}}$ is the combustion air at the inlet of the recuperator, and $T_{\text{flue gas (inlet)}}$ is the temperature of the flue gas at the inlet of the recuperator. Figure 6.41 shows the relationship between the combustion efficiency and the relative air preheating rate for various flue gas inlet temperatures. Figure 6.42 shows how three different ceramic recuperators were calculated to perform as a function of the flue gas inlet temperature. Louis et al. (1998) described a new self-recuperative single-ended radiant tube burner named CERAJET, which is made of silicon carbide and capable of combustion air preheat temperatures up to 900°C (1650°F) and a thermal efficiency up to 75%.⁷⁵

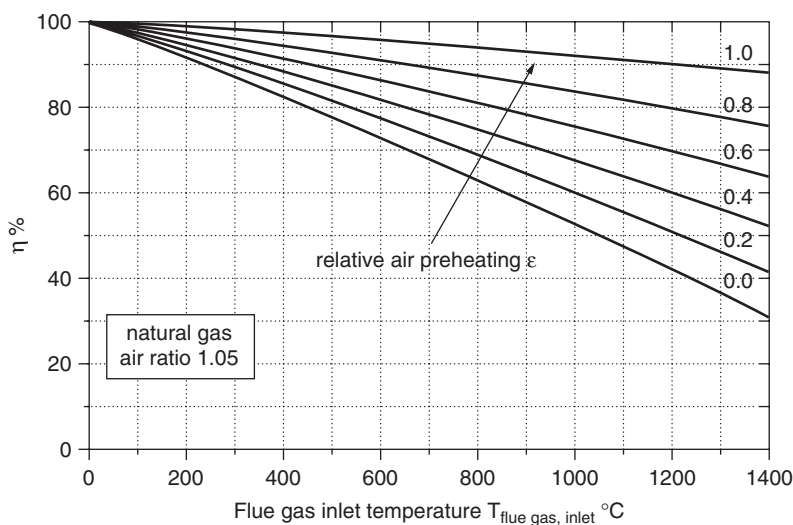


FIGURE 6.41 Combustion efficiency as a function of flue gas temperature at the inlet of the recuperator. (Courtesy of The Gas Research Institute.¹¹¹)

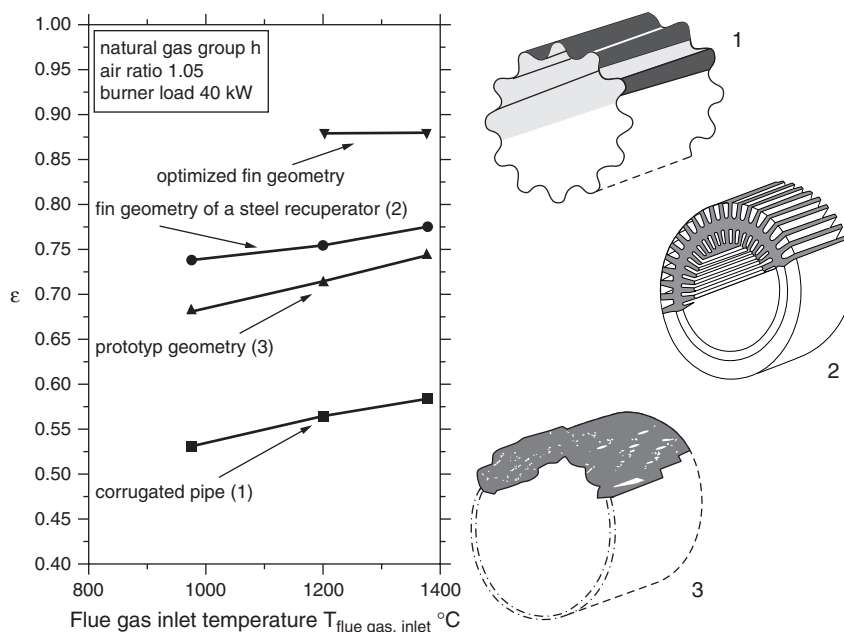


FIGURE 6.42 Calculated relative air preheating rates for different recuperator geometries. (Courtesy of The Gas Research Institute.¹¹¹)

6.4.5.3 Furnace or Flue Gas Recirculation

In flue gas recirculation (see Figure 1.20), some of the hot combustion products from the exhaust stack are recirculated back through the burner. This requires an external fan and insulated ductwork that is often fairly large because of the hot expanded gases and low gas pressures. Although there are added costs for the fan and ductwork, this method has better control as the amount of flue gas recirculation can be controlled independently of the burner.

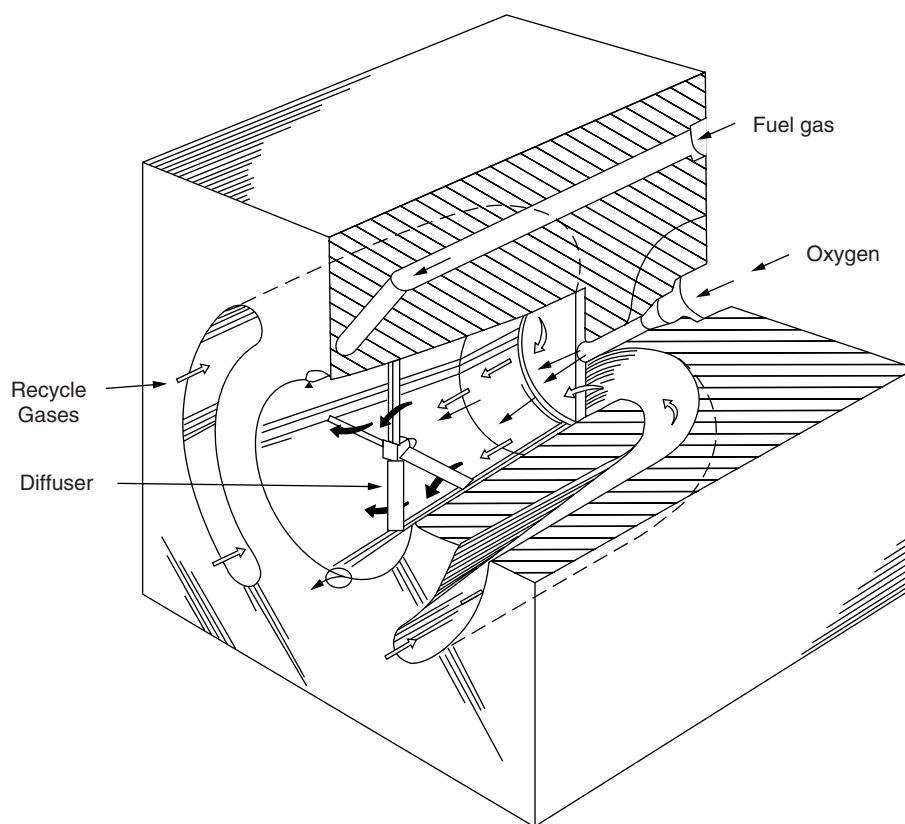


FIGURE 6.43 Oxygen/fuel burner incorporating furnace gas recirculation.¹¹⁴

In furnace gas recirculation, the hot combustion products inside the furnace are recirculated back into either the flame or inside the burner to mix with either the fuel or the oxidizer. An example is given in Wüning and Wüning (1996).¹¹² No external ductwork or fans are required, which makes it much less expensive than furnace gas recirculation. The fluid pumping is accomplished by the design of the burner, where the fuel or the oxidizer flows through a venturi to create a vacuum to induce the furnace gases to flow back toward the burner. This method is less controllable than flue gas recirculation because the amount of recirculated gas is dependent on the burner conditions. Also, at least one of the incoming gases must have a high enough supply pressure to create suction in the venturi. An example of an oxygen/fuel burner¹¹³ utilizing furnace gas recirculation is shown in Figure 6.43.¹¹⁴ The measured heat flux rates from that burner using a total heat flux probe inserted into a pilot scale furnace are shown in Figure 6.44. Plessing et al. (1998) described a combustion system that uses furnace gas recirculation and is known as flameless oxidation (FLOX) where the combustor temperature must be a minimum of 800°C (1500°F) for safe operation.¹¹⁵ The primary advantage of FLOX is low NO_x emissions. Although no specific measurements are given, the radiation from the flame was said to remain the same even though the flame is completely nonluminous. Abbasi et al. (1998) described an advanced U-tube radiant tube burner that uses internal furnace gas recirculation to improve temperature uniformity and lower pollutant emissions.⁷⁶

6.4.6 PULSE COMBUSTION

Pulse combustion refers to the periodic change in the outlet flow of the exhaust products from a combustion process to produce a pulsating flow. Putnam (1971) has written an entire book on the subject.¹¹⁶ The main purpose of that book was how to suppress unwanted, combustion-driven

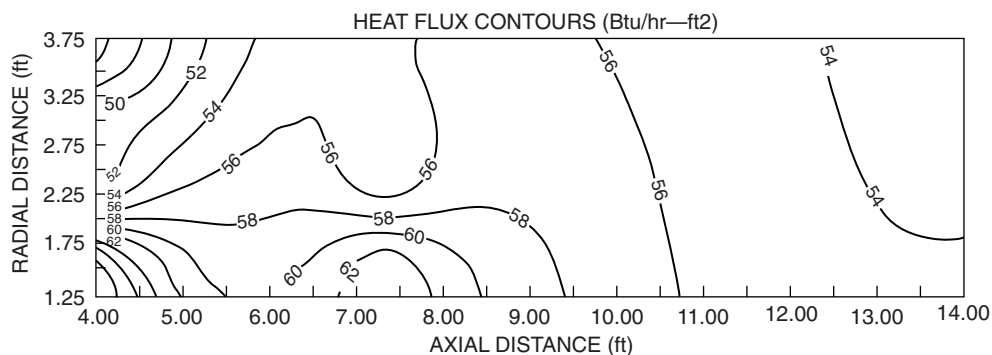


FIGURE 6.44 Measured heat flux rates from an oxygen/fuel burner incorporating furnace gas recirculation.¹¹⁴

oscillations that cause noise and vibration that are detrimental to processes which are not designed to pulse. He discussed how oscillations can be generated in a wide range of industrial combustors. Some of the early papers focused more on the acoustics and fluid dynamics than on the heat transfer from pulse combustion.¹¹⁷

Some of the benefits of combustion specifically designed to pulse include:

- Ability to burn various fuels
- High combustion intensities
- Low NO_x formation
- Low excess air requirements
- Self-aspiration, which eliminates the need for combustion air fans
- Improved heat transfer¹¹⁸

Note that for some pulse combustors, the burner design is integrally linked to the combustion chamber. Therefore, both the burner and, where appropriate, the combustor are considered together here for the sake of consistency and to avoid redundancy. Zinn et al. (1993) described a tunable pulse combustor capable of incinerating waste materials with high efficiencies.¹¹⁹

There are two common types of “pulse” combustion: natural and mechanical. In natural pulsed combustion, the combustion system is designed to deliberately create a resonance that causes the flow to pulse. The heat released in the combustion process excites the fundamental acoustic mode of the combustor. The resulting oscillations interact with the combustion process in such a way as to cause periodic reaction and heat release rates. When the periodicity of the heat release is in phase with the acoustic pressure oscillations, the pulsing process is self-sustaining. There are three common types of natural resonant combustor designs: the quarter wave or Schmidt combustor, the Helmholtz combustor, and the Rijke combustor.¹²⁰ The resonance in the chamber causes the exhaust gases to pulse in a regular and periodic fashion. The pressure initially rises during the first part of the cycle, forcing the exhaust gases out of the tailpipe. After the initial pressure wave, a partial vacuum is created, which pulls in fresh combustion air. The fuel valve (sometimes referred to as a flapper valve) is simultaneously opened by the vacuum pressure and fuel is injected into the incoming combustion air, creating a flammable mixture. The cycle frequency rates may be over 100 Hz and often appear as steady flow to the naked eye and to the ear. Neumeier et al. (1991) discussed modeling of the flapper valve in pulse combustors.¹²¹

The second type of pulse combustor is where a mechanical system causes the flow pulsations. This is often accomplished by opening and closing a valve at fairly high frequencies. One common way is to open and close the fuel valve while keeping a constant flow of oxidizer to a burner. This causes successive regions of fuel-rich and fuel-lean combustion, which can lead to significant reductions in NO_x emissions. This type of pulse combustion has a much lower cycle frequency

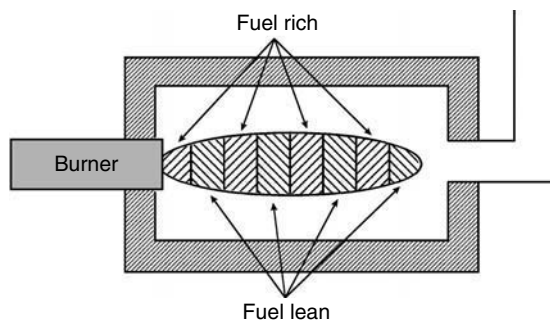


FIGURE 6.45 Alternating fuel-rich and fuel-lean zones in a pulse combustor.¹

and is often detectable by the human eye and ear. Two concerns with this type of pulse combustion system are the life of the mechanical valve because of the number of times it opens and closes, as well as the increase in noise that is often associated with the lower frequency pulsing. George and Putnam (1991) studied the development of a rotary valve for pulse combustors and looked at pulse frequencies from 25 to 90 Hz.¹²² They noted that significant improvements in heat transfer can be expected using a rotary valve pulse combustor. The Institute of Gas Technology (Chicago, IL) has developed an oscillating combustion process that uses a fast cycling valve to pulse the incoming fuel to a burner, while having a steady flow of oxidant.¹²³ By pulsing the fuel, alternating zones of fuel-rich and fuel-lean combustion are produced (see Figure 6.45). The preferred pulse rate is about 5 to 30 cycles per second. The primary reason to do this is to reduce NO_x emissions because NO_x formation is less favorable under either fuel-rich or very fuel-lean conditions. However, a secondary benefit of the process is often an increase in heat transfer from the flame to the load. This can occur for two reasons. The first is that the fuel-rich regions often produce a luminous flame that radiates heat more efficiently to the load compared to less luminous or nonluminous flames. A second reason is that the oscillating causes the flame gases to pulse, which tends to minimize boundary layer formation and increase convective heat transfer.

In either type of pulse combustion, an important benefit of the process is increased convective heat transfer rates. This is explained by analyzing the flow of the combustion products over a surface (the load). In continuous forced convective flow, a boundary layer builds up as the gases flow over a surface. This boundary layer causes thermal resistance to heat transfer from the gases to the solid surface. In pulsed flow, the boundary layer is continually broken up by the constant changes in flow over the surface. In the higher pressure part of the cycle, the flow goes in one direction, while in the low pressure (vacuum) portion of the cycle, the flow may actually reverse. Breaking up or reducing the thickness of the boundary layer lowers the resistance to heat transfer between the gases and the solid, which enhances the heat transfer rate for a given system compared to nonpulse flow. Hanby (1969) reported heat transfer improvements of up to 100% for an air/propane flame in a straight tube for pulse combustion with a frequency of 100 Hz.¹²⁴ The convective heat transfer coefficient was 15 Btu/hr-ft²-°F (85 W/m²-°C) for steady flow and 35 Btu/hr-ft²-°F (200 W/m²-°C) for pulse flow. Blomquist et al. (1982) measured an improvement in heat transfer ranging from 25 to 40% for a Helmholtz type pulse combustor operating at about 70 Hz and firing on natural gas.¹²⁵ Corliss and Putnam (1986) noted an increase in the convective heat transfer coefficient by as much as 100% using pulse combustion.¹²⁶ The results also showed that when considering both heat and mass transfer, a process can be improved by up to a factor of 5. Brinckman and Miller (1989) reported experimental measurements on methane combustion in a Rijke pulse combustor.¹²⁷ Results showed that pulse combustion could be maintained over a range of fuel flow rates, equivalence ratios, nozzle exit positions, and combustor lengths. Xu et al. (1991) discussed pulse combustion of heavy liquid oil fuels in a Rijke-type pulse combustor.¹²⁸ Arpaci et al. (1993) developed an empirical equation for the convective heat transfer in a pulse combustor tailpipe¹²⁹ to correlate the experimental

data of Dec and Keller (1989).¹³⁰ The Nusselt number increased with combustion chamber pressure (0.5 to 9 kPa) and oscillation frequency (54 to 101 Hz). Barr et al. (1996) described a computational tool for designing pulse combustors.¹³¹ The predicted heat transfer efficiency of a pulse combustion system can be improved by increasing the number of tailpipes in the system. However, there is a diminishing return as the number of tailpipes increases above about 10. Marsano et al. (1998) reported experimental measurements on a 250-kW Helmholtz-type pulse combustor.¹³² Lundgren et al. (1998) analytically showed that a Helmholtz-type combustor can increase the heat transfer by a factor of approximately 2 to 6.¹³³ Grosman et al. (1998) successfully demonstrated the use of an oscillating combustion system using oxygen-enhanced combustion (see [Chapter 21](#)) in a rotary iron melter, ladle preheater, and batch annealing furnace in the steel industry.¹³⁴ In laboratory tests, increases in heat transfer up to 59% were demonstrated.

6.5 IN-FLAME TREATMENT

In some applications, particularly waste incineration, it is possible to inject the material to be heated directly into the flame. The material can be a solid, a liquid, a gas, or a combination of the three. For example, Steward and Guruz (1974) describe the injection of solid particles (alumina or magnesia) into the fuel stream of a burner to study the effects on radiation.¹³⁵ The injected particles had relatively little effect on the measured radiation in the furnace. In many cases, the objective is to fully combust the injected material. For example, Santoleri (1986) describes the incineration of waste liquid fuels injected through a swirl burner, as shown in Figure 6.46.¹³⁶ Although the waste materials in these systems have some heating value, it may be fairly low because of the high water content. This can make the combustion process more difficult due to the high heat extraction from the water. The reason for injecting the waste materials into the flame is to improve the heat and mass transfer processes and more efficiently destruct the waste.

Another type of in-flame treatment involves injecting a solid material into the flame for some type of heat treatment. For example, an inorganic solid can be injected through the flame to be heated and possibly even melted. That type of material is strictly a heat load and does not contribute any heating through combustion reactions. Although the heat transfer to the load can be significantly

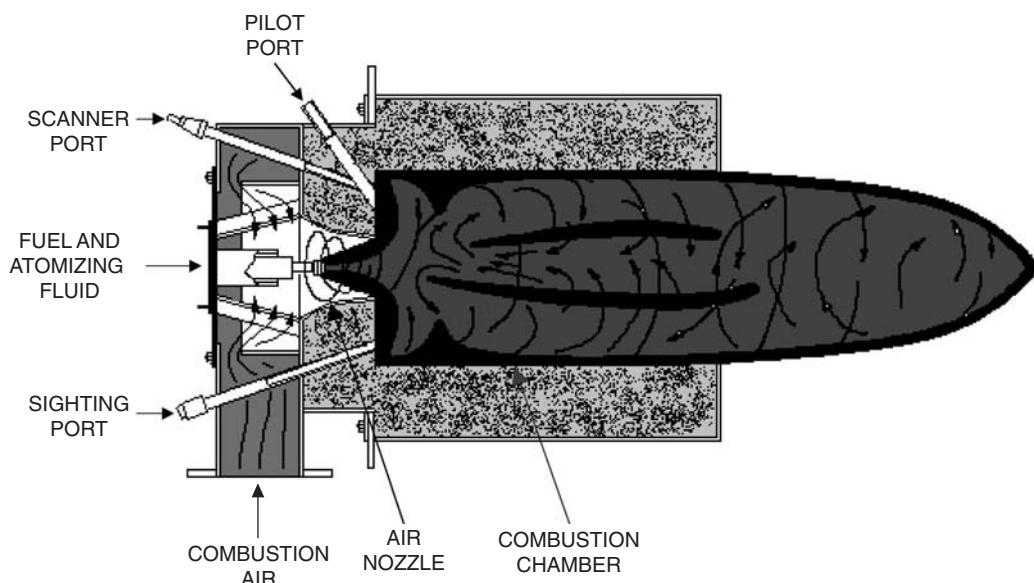


FIGURE 6.46 Swirl burner with waste liquid injection. (Courtesy of American Society of Metals.¹³⁶)

increased using in-flame treatment, because of the intimate contact with the hot exhaust gases, the main challenge is transporting and injecting the solids into the flame. Wagner et al. (1996) describe a process for injecting spent aluminum potliner (SPL) into a cyclonic combustor for destruction of the SPL, which is a hazardous waste.¹³⁷ SPL is a refractory-like substance containing roughly 30% carbon, 15% highly leachable fluoride, and 0.2% cyanide, and is used as the liner in furnaces used to reduce alumina into aluminum in the Hall reduction cell process. The SPL was fed in the top of the cyclonic combustor and traveled through the flame zone, which destroyed the hazardous waste with high efficiency.

Zhang et al. (1996) presented a novel method for incinerating solid wastes by feeding them through a counter-rotating air-oxy/fuel (see [Chapter 20](#)) burner.¹³⁸ High temperatures are important in destructing waste materials. The process was tested in a 50-kW (170,000-Btu/hr) pilot-scale furnace using phenolic-coated foundry sand as the waste material to simulate sand reclamation. The furnace had an internal diameter of 0.2 m (0.7 ft) and a height of 2.5 m (8.2 ft). Complete destruction of the resin was demonstrated and clean sand was produced.

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