
14 Radiant Tube Burners

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14.1 INTRODUCTION

Indirect firing of the heat processing equipment is essential for many heat treatment processes where the work must not come into contact with the products of combustion or the flame. While separation can be achieved by such means as using an externally heated muffle or saggers to enclose the work, the radiant tube offers many advantages as a source of indirect heating and has become firmly established in the metallurgical heat treatment field. For processes requiring special atmospheres or scrupulously clean conditions in the furnace, it provides very even heat distribution at high intensity. Radiant tubes are normally gas-fired, although oil-fired burners are available.

A radiant tube furnace has one or more radiant tubes inside which the combustion process is completely contained. Heat is given up to the inside wall of the tube, conducted through the wall, and radiated to the work in the furnace. In some cases where a special atmosphere is recirculated within a furnace, heat is also dissipated by convection, but the typical main function of a radiant tube is that of a radiant heat source. There are many different types of radiant tubes for industrial furnaces, but they can generally be subdivided into “straight-through” and “single-ended” types.

14.2 TYPES OF RADIANT TUBES

The straight-through types principally comprise straight, “U”-shaped or “W”-shaped tubes, each of which can be fitted with internal or external recuperators or regenerative burners. The “U”- and “W”-types provide a greater heating path length with consequent increases in heat transfer area and potential efficiency. A specific process or equipment configuration need may dictate that radiant tubes have another geometry.

The single-ended tube type has the burner and flue combined at one end and utilizes an internal tube or structure that takes the hot combustion gases down the tube axis to the closed end of the outer tube, where they are diverted back along the external annulus in a counter-flow direction. A refinement of this principle incorporates a recuperative section closely coupled to the burner head — a single-ended self-recuperative radiant tube (commonly designated SERT).

Radiant tubes are designed to effectively transfer the heat from the combustion products to the radiant tube and subsequently to the load. The tube geometry is based on the load to be heated, the available space to locate the radiant tubes, and the tube material characteristics and limits. It is also important that the temperature of the radiant tube be as uniform as possible to optimize the heating capabilities of the radiant tube and to ensure that no part of the tube is significantly above the mean tube temperature, with that mean being comfortably within the design limits of the tube material selected for the application.

To meet these radiant tube design requirements, two firing system concepts have evolved — *nonrecirculating* and *recirculating* — as designated by Pritchard et al.,¹ each being used in various ways with both straight-through and single-ended designs.

Nonrecirculating radiant tubes are the most common. The distinguishing feature of these tubes is that the combustion products make a single pass through the tube. For a straight-through tube, a burner fires at one end and the products of combustion are exhausted at the other end. In the case of a nonrecirculating, single-ended radiant tube, the burner inlet and radiant tube outlet are located coaxially on the same end of the tube, typically with an annular exhaust surrounding the burner. Nonrecirculating radiant tubes rely on the burner design and adjustment to distribute the heat uniformity over the surface of the tube.

Recirculating radiant tubes use high-velocity burners to entrain combustion products from the exhaust leg of the radiant tube with the jet action of a high-velocity burner, and pump the combined mass of fresh and recirculated products of combustion through the tube. Adding the recirculated products reduces the peak temperature of the burner’s flame envelope and thus the potential for tube hot spots.

14.2.1 NONRECIRCULATING TUBE TYPES

Non-recirculating tubes are manufactured in a number of geometries, as shown in [Figure 14.1](#).

Straight radiant tubes are the simplest to manufacture. These tubes enter the chamber to be heated through one wall and exit on the opposite wall. Straight tubes are typically fixed at the inlet end, with the exit end floating through a gland to allow for differential expansion. Both “U”- style and “W”-style radiant tubes enter and exit the chamber on the same side, usually configured with both entrance and exit legs passing through a single insulated plate (a “bung”) that is either bolted or welded to the furnace casing. The expansion of the tube is allowed for in the method of mounting of the tube to the bung. The “U”- and “W”-styles have the advantage of increasing the exposed

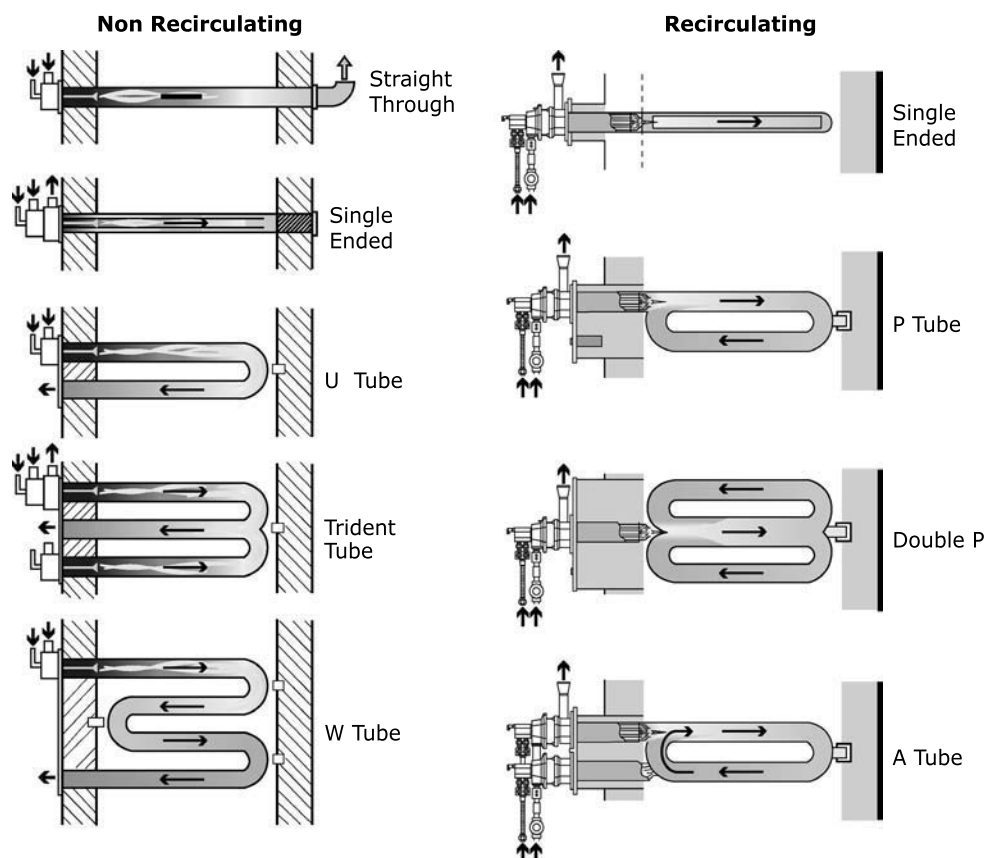


FIGURE 14.1 Nonrecirculating and recirculating radiant tube configurations. (Courtesy of Joachim G. Wunning, WS Thermal Process Technology, Inc., Elyria, OH. With permission.)

heating length and therefore tend to be more efficient than straight-through tubes, as well as reducing the number of penetrations in the furnace casing. The “trident-style” radiant tube was developed to increase the effective total heat transfer from the tube surface. In this type of radiant tube, two burners share a common radiant tube exhaust leg, thereby optimizing the heat transfer of the system. The challenge with the longer radiant tubes is to provide adequate temperature uniformity to avoid local overheating of the tube material. As the length of the radiant tube increases, such as in large “U”- and “W”-type radiant tubes, there is greater potential of deviation in the temperature along the length of the radiant tube. Regenerative burners can greatly improve the temperature uniformity of large nonrecirculating radiant tubes.

The single-ended radiant tube can be built as a nonrecirculating radiant tube. The single-ended tube type has burner and flue combined at one end and utilizes an internal tube or structure that takes the hot gases to the closed end of the outer tube where they are diverted back along the annulus in a counter-flow direction. A refinement of this principle incorporates a recuperative section closely coupled to the burner head — the single-ended self-recuperative radiant tube (commonly designated SERT) — to provide energy savings.

14.2.2 RECIRCULATING TUBE TYPES

Recirculating radiant tubes are produced in several configurations, also shown in [Figure 14.1](#). The most common current configuration is the recirculating single-ended radiant tube where the central tube structure conveying the burner products of combustion is constructed to allow a portion of

the exhausting products to pass from the annulus into the center. However, recent adaptations of 1950s U.K. designs, such as the Wellman Furnaces “Jetube” to include recuperative burners, have produced other configurations such as the “P” tube, the double “P” tube, and the “A” tube for use with regenerative burners.

The main advantage of the recirculating type of radiant tube is that the combustion process is highly dilute, leading to lower flame temperatures and the minimization of hot spots near the flame envelope. Additionally, high recirculation ratios within the tube result in improvement in gas temperature uniformity and correspondingly more uniform tube wall temperatures. A further advantage of recirculating radiant tubes with their inherent flame temperature reduction is a corresponding reduction in thermal NO_x emissions. The mixing of the flame and recirculated, cooled, exhausting products of combustion limits the peak flame temperature and exploits the well-known potential of the use of the recirculated products to reduce the NO_x emissions of the radiant tube — commonly known as the use of flue gas recirculation (FGR). With increasing environmental quality pressures, recirculating radiant tubes are enjoying growth in popularity.

14.2.3 A BRIEF HISTORY OF THE APPLICATION OF “INTEGRAL” EXHAUST HEAT RECOVERY TO RADIANT TUBES

Not a great deal has been written about the history of the radiant tube, but it has been reported² that the first straight, single-ended recuperative radiant tube was developed in Germany in the 1940s. A developmental form of recuperative tube is described in a 1955 paper by Anderson and Broome.³ In the early 1960s, two British gas industry groups pursued its further development. Licensing to manufacturers produced a flood of successful industrial applications in batch and continuous heat treatment furnaces in the U.K. Rapid popularity followed throughout high-fuel-cost industrial Europe and gave rise to further development utilizing the ever-growing base of metallic and ceramic high-temperature materials. Similar developments took place in Japan in the 1970s and 1980s. The use of the single-ended self-recuperative radiant tube is now considered an “industry standard” application.

In the late 1980s, the authors’ company developed and successfully applied compact ceramic regenerators to “U” and “W” tube designs in several large U.S. steel-industry strip line installations.⁴ These led to licensing to Tokyo Gas of Japan and similar applications there in the 1990s, and to local Japanese manufacturers also picking up the regenerative design challenge.

The performance and features of these designs are discussed in later sections of this chapter.

14.3 RADIANT TUBE BURNERS

There are a number of burner options for radiant tubes. Typically, these burners are classified as cold (ambient) air burners, preheated air burners, self-recuperative burners, and regenerative burners. Cold air burners are generally used for lower process temperatures and tube firing rates, where the fuel economies provided by heat recovery cannot be economically justified. At higher temperatures and/or high tube firing rates, recuperators, self-recuperative burners, or regenerative radiant tube burners can be justified. It is important to note that these “lower” and “higher” temperature references are to the temperatures of the exhaust gases of the radiant tube, *not* the heated-process temperature.

Burners for radiant tubes need to provide a uniform heat release profile along the length of the radiant tube to promote an even temperature distribution. A typical limiting factor for radiant tubes is the maximum temperature that the tube material can withstand, given the imposed mechanical load, thermal stresses arising from the heat flux through the tube wall, and the limiting creep stresses. Uneven heat release from the flame and/or combustion products along the length of the radiant tube results in local hot spots: these will either lead to premature tube failure or limit the output for a given radiant tube to that of the lower average temperature. Uniform heat flux from the radiant tube will also likely improve the load temperature uniformity and offer

quality improvement. In practice, few designs come close to providing a uniform heat flux — the exceptions are regenerative burners and highly recirculating tubes.

Before considering the details of the specific types of burners available for radiant tubes, it is appropriate to discuss some system design factors that significantly influence the choice of burner for an application.

Just as with any other fired system, the radiant tube's internal space that forms the combustion chamber can be chosen to be fired under suction, under pressure, or "neutral," compared with the atmospheric pressure or the pressure in the chamber that the tube is heating. A combination of factors can affect this choice. A major consideration for furnaces using a controlled atmosphere to produce specific qualities in the heated product can be the effect of a ruptured tube on the atmosphere: if a pressurized tube ruptures, combustion products will leak out and mix with the furnace atmosphere if the tube pressure is higher than the internal atmosphere pressure in the furnace. In such a case, it is common for the furnace designer to specify a suction or neutral pressure condition inside the tube, such that furnace atmosphere will be drawn into the tube if a crack develops in the tube, and it will not be necessary to shut down the furnace to find "a leaker." These systems can be provided as suction-only systems with no pressure air supplied, or as "push-pull systems" that use a combination of forced air supply to the burner and induced exhaust at the outlet such that the resulting tube internal pressure is slightly negative to the furnace internal pressure.

Another influence on the system choice can be the furnace layout or physical space constraints. Forced-draft systems, with burners supplied with pressure air and fuel, are conventionally preferred because of the fine degree of control that can easily be exerted over the fuel input and burner air/fuel ratio. Such a system, however, requires air, fuel, and exhaust piping to each radiant tube in the system. If space is tight, or for economic considerations, it might be prudent to use an all-suction system in which the exhaust is directly coupled to a draft-producing device and the combustion air is pulled in as ambient air from the furnace surroundings, thereby eliminating the need for a forced combustion air supply. This situation is also most applicable to vertical radiant tube-fired furnaces where tubes are installed at many elevations and the system designer is faced with the effects of buoyancy in considering the distribution of flows in vertical manifolds. For "basic" systems, although not popular today because of their inherent lack of ability to closely control air/fuel ratio, high-pressure fuel can be used to inspire ambient air into the burner inlet and eliminate the need for combustion air piping, when a low-resistance tube and exhaust system is in use.

The following discusses burners applicable to these various scenarios.

Most radiant tube burners are of nozzle-mixing design and have long flames resulting from delayed combustion. The common approach to achieve a delay in the combustion of the fuel is to have multiple air entry points that stage the mixing of the air into the fuel within the tube. This design approach allows greater control of the heat release within the radiant tube. Burners designed specifically for recirculating-style tubes will typically use a high-velocity burner rather than a delayed combustion design in order to have sufficient burner jet energy to promote recirculation. Thermal efficiency must also be considered in the design and selection of radiant tube burners. Incorporation of an exhaust heat recovery means is the standard approach to enhancement of thermal efficiency.

14.3.1 INSPIRATOR AND PULL-THROUGH BURNERS

Inspirator and pull-through burners are designed to be used without a pressure combustion air supply. In the inspirator style, all the combustion air is pulled in by the entrainment action of a high-pressure fuel jet, although sometimes with the assistance of the "pull" created by a vertical exhaust stack on the tube or by a close-coupled induced draft system. Pull-through burners use only an induced draft system to pull the combustion air into the burner. Adjustable shutters on the entrance of the burner can be used to adjust the flow of combustion air relative to the applied suction. [Figure 14.2](#) illustrates a typical pull-through radiant tube burner.



FIGURE 14.2 “Pull-through” radiant tube burner.

The main advantage of the inspirator and pull-through radiant tube burners is the simplicity of the combustion system. Requiring only that the fuel supply be connected to the burner, combustion air piping is eliminated and installation costs are low.

Because the combustion air is not metered in an inspirator or pull-through burner design, it is difficult to maintain the optimum fuel/air ratio for combustion. The fuel/air ratio is subject to the local ambient conditions, including temperature and humidity fluctuations as well as the inherent ratio characteristics of the air entraining feature. Many of the environments in which radiant tube burners are installed are dirty, any plugging of the air passages leading to a change in the fuel/air ratio.

The low-pressure drop style of fuel and air mixing inherent in a typical simple inspirator or pull-through burner limits the degree to which control over mixing can influence the production of a uniform temperature profile along the length of the radiant tube.

These limitations of the simple inspirator and pull-through burner designs are largely eliminated by forced-draft radiant tube burners.

14.3.2 FORCED-DRAFT RADIANT TUBE BURNERS FOR NONRECIRCULATING TUBES

Forced-draft burners receive combustion air from an external source, typically a low-pressure (1 psig or less) combustion air blower. Pressurizing the combustion air supplied to the burner allows for a variety of combustion controls to be applied to maintain the desired fuel/air ratio over a range of burner firing rates. With pressure combustion air and metering, the potential firing efficiency of this type of radiant tube burner is higher than its pull-through counterpart.

14.3.2.1 Forced-Draft Burners without Integral Heat Recovery

Forced-draft burner designs are available for both ambient air and preheated combustion air from an external recuperator. In many cases, the burner design is the same for both but uses different materials of construction to accommodate the higher burner body and flame stabilizer temperatures from the use of preheated air.

Forced-draft burners often incorporate a means to adjust the flame length to allow it to be matched to a particular radiant tube length or configuration. An old rule of thumb often used for “U” tubes is to adjust the flame length so that it can just be seen coming around the U bend when looking up the exhaust leg. Some burner designs require mechanical adjustment of the flame stabilizer location during installation of the radiant tube burner in order to change flame length. A more desirable method is to provide separate combustion air paths to a primary, secondary, and

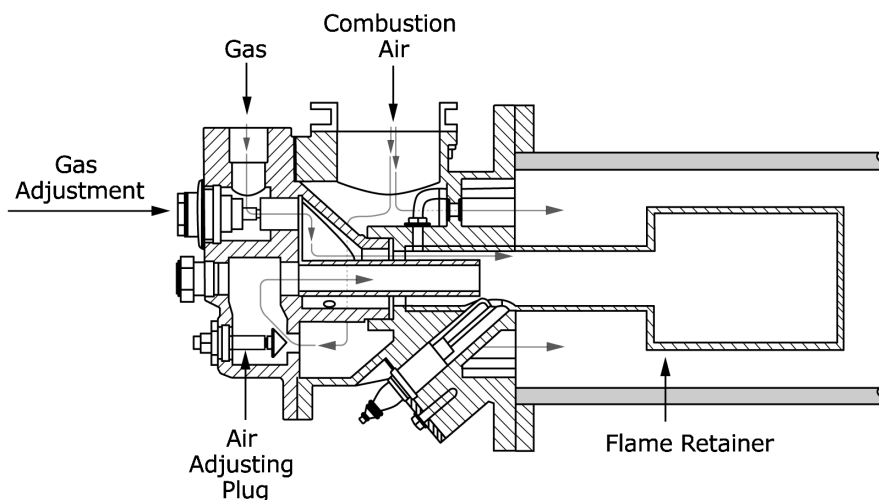


FIGURE 14.3 Forced-draft radiant tube burner.

sometimes tertiary combustion zone, with adjustment of the relative flow rates to each. This approach allows for the rate of mixing and combustion to be controlled in each zone with resulting variation of the flame characteristics. Burners of this design typically have an external adjustment to vary the portion of air introduced into the primary zone, having the advantage of adjustment capability with the burner in operation, allowing visual fine-tuning of the flame characteristics. [Figure 14.3](#) illustrates a typical forced-draft radiant tube burner.

14.3.2.2 Regenerative Burners

Regenerative burners combine both the burner and the heat recovery device into a single package and are capable of providing the highest thermal efficiencies of all the radiant tube burner styles. Thermal efficiencies on an HHV-available-heat basis range from 75 to 85% for regenerative burners.

Within the burner structure is a heat storage bed. One such burner is mounted on each end of the nonrecirculating radiant tube. One burner will fire while the companion burner will exhaust the products of combustion from the tube for a specific period, typically from 20 to 60 s. At the end of the period, the operation is reversed: the burner that was previously firing is placed in exhaust mode while the burner that was previously exhausting is placed in firing mode. The periodic cycling continues while there is a demand for heat by the process.

The combustion air to the firing burner is heated within the heat storage bed by recovering the heat stored in the bed material during the previous exhaust cycle. The resulting air temperature is typically within 200°F of the temperature of the radiant tube exhaust gases. The preheated combustion air is then delivered to the burner nozzle where fuel is introduced, mixing occurs, and combustion is initiated by a pilot burner. In the exhausting mode, the hot exhaust gases from the radiant tube pass the burner nozzle to enter the regenerative bed. These gases heat the bed material that was previously cooled during the firing cycle. The cooled exhaust gases are then either vented to atmosphere or taken away by a directly connected exhaust collection system, depending on the particular combustion system configuration. Some portion of the cooled exhaust gas may be directly recirculated to the firing burner for NO_x emission reduction purposes.

The regenerative bed typically takes the form of either loose packing such as balls, or a monolithic design of supported webs with through-gas passages, both normally of ceramic material. Loose packing is housed in a container that allows either vertical or horizontal flow through the bed, depending on the burner configuration. [Figure 14.4](#) illustrates a typical regenerative burner

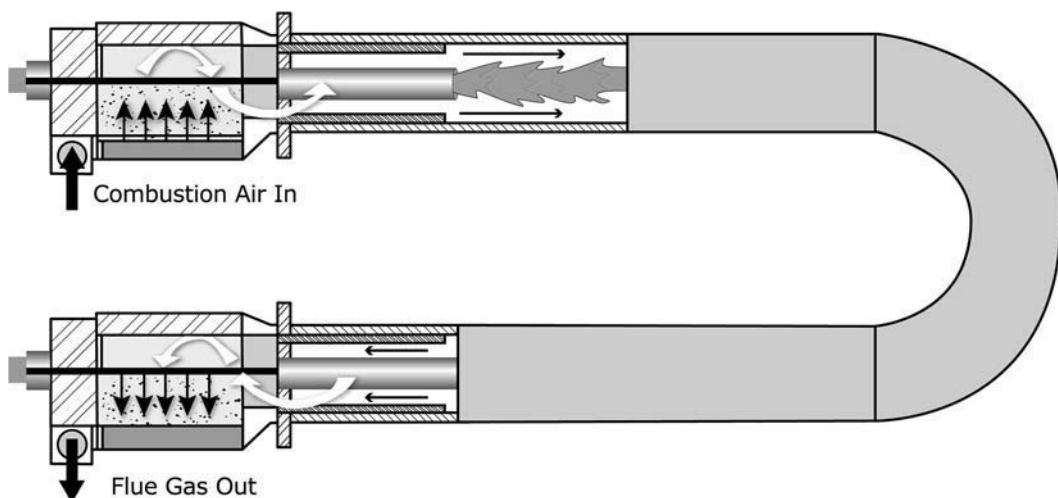


FIGURE 14.4 Regenerative radiant tube burner.

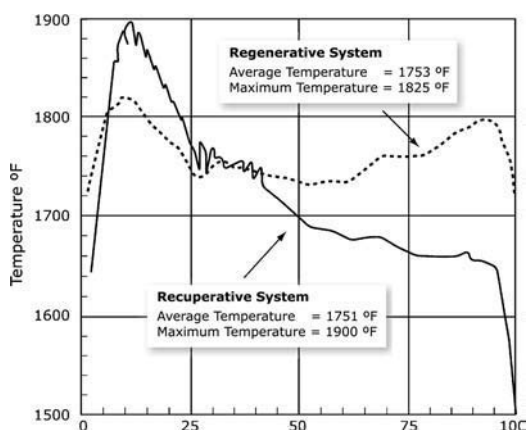


FIGURE 14.5 Comparison of actual tube temperature profiles for recuperative and regenerative tubes at an average temperature of 1750°F.

employing a loose-packed bed. The vertical flow path has the advantage that gravity maintains the bed material location, which is critical to maintaining thermal performance over time. For horizontal packing geometries, the bed housing and interior baffles are used to ensure that material settlement due to thermal cycling and vibrations typical around most industrial furnaces do not allow the products of combustion or the air to bypass the storage bed.

Regenerative burners offer several advantages in the firing of radiant tubes. The inherently high thermal efficiency leads to the highest possible fuel savings and thus to reduced operating costs. Potentially even more significant is the improvement in the temperature uniformity of the radiant tube provided by the alternate firing from each end of the tube, especially on large, long tubes. Figure 14.5 compares the temperature distribution of a regeneratively fired tube and a conventional forced-draft burner-fired recuperated tube.

The improved tube temperature uniformity inherent with regenerative burners leads directly to improved radiant tube life. Small reductions in peak tube temperature result in operating the tube material at the higher creep strength of the average operating temperature of the radiant tube. A second benefit of an improvement of the temperature uniformity for a given radiant tube is that

more heat can be transferred for a given average temperature, and thus more transferred while not exceeding the maximum tube temperature service limit. The advantage of improving the temperature uniformity can be realized through an increase in the heating capability of an existing furnace without negative impact on the tube life, or in a reduction in the size of the heating zone at the design stage of a new furnace. Finally, improvement in the thermal profile can be expected to have a positive impact on the heating quality of the load.

14.3.3 FORCED-DRAFT BURNERS FOR RECIRCULATING TUBES

Although ambient air burners have been used with recirculating tubes, self-recuperative burners have become the accepted means of firing these tubes.

14.3.3.1 Single-Ended Radiant Tube Burners

Early single-ended radiant tubes used ambient air burners. However, self-recuperative burners are now more commonly used, given the proximity of burner and exhaust. Single-ended radiant tube burners fire into small-diameter tubes or tubular structures mounted on the axis of a larger-diameter, outer tube capped at one end. The products of combustion from the combustion process in the inner tube return in an annulus around the inner tube and exhaust around the burner. The relative location of the outer flue gases makes for an ideal match with the self-recuperative burner geometry (Figure 14.1).

Single-ended radiant tube burners must provide combustion of a relatively low-intensity within the inner tube, especially when used with nonrecirculating style tubes. The operating temperature limit of the center tube material is a limiting factor.

14.3.3.2 Self-Recuperative Burners

Self-recuperative burners are commonly used on single-ended style radiant tubes and on some of the newer styles of recirculating radiant tubes. These burners integrate the burner components and a heat exchanger into a single package. Because the burner nozzle is mounted directly on the end of the air side of the recuperator, there are no air-side heat losses. Accordingly, self-recuperative burners often have higher efficiencies than their remote-recuperator counterparts. Typical thermal efficiencies of 65 to 72% on an HHV-available-heat basis are achieved. Figure 14.6 illustrates a typical self-recuperative radiant tube burner.

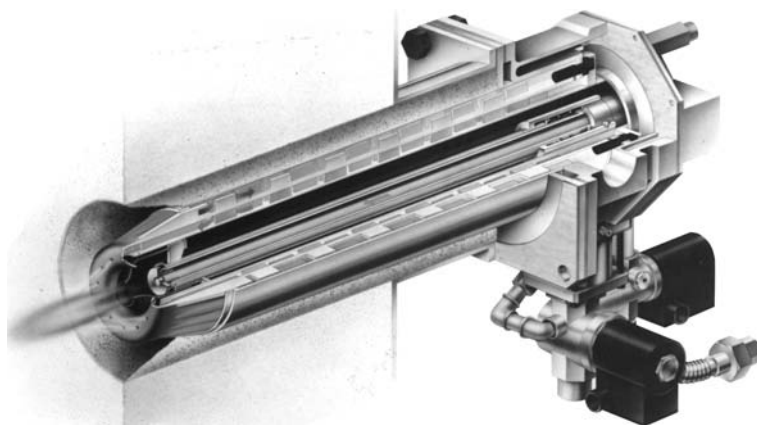


FIGURE 14.6 Self-recuperative radiant tube burner. (Courtesy of Joachim G. Wunning, WS Thermal Process Technology Inc., Elyria, OH. With permission.)

14.4 ENERGY RECOVERY METHODS FOR RADIANT TUBES

Regenerators and recuperators integrated with burners were described in Chapter 14.3. Non-integrated heat recovery devices are described below.

14.4.1 RECUPERATORS

Recuperators using radiant heat transfer from the exhaust gas to the heat transfer partition, in conjunction with preheated air burners, are the typical choice for radiant tube applications. These recuperators can be an insertion type (also known as “plug-in”) in the exhaust end of the radiant tube itself, or externally mounted and connected to the tube by a short section of insulated ductwork.

14.4.1.1 Insertion-Type Recuperators

The insertion-type recuperator (Figure 14.7) is placed into the exhaust leg of the tube. Typical thermal efficiencies on an HHV-available-heat basis for this class of recuperator are 50 to 55%.

A typical design has the exhaust gases passing through an outer annulus formed by the outside wall of the recuperator partition and the inside wall of the radiant tube exhaust leg. It will include fins, sometimes staggered, to increase the heat transfer on one or both sides of the partition. The air side of the recuperator will consist of two passes: one to deliver the combustion air to the recuperator and the other to return the air to the exit of the recuperator, although only one of them truly constitutes a heat transfer pass with the exhausting products of combustion. The preheated air is then connected to the companion burner, usually on the other end of the same radiant tube, although it may be on an adjacent tube in the same firing group if more convenient in the particular physical layout.

Located within the exhaust leg of the radiant tube as it passes through the furnace wall, the insertion-type recuperator minimizes external space requirements. Additionally, ambient losses are minimized by containment within the furnace wall. However, care must be taken so that the end of the insertion-type recuperator does not extend within the heated interior of the furnace chamber. Should the recuperator be allowed to extend past the wall and into the part of the radiant tube within the furnace chamber, then useful energy that would otherwise be available for heating the load will be extracted into the recuperator, both limiting the heat available to the furnace and reducing the effectiveness of the recuperator.

14.4.1.2 External Recuperators

An alternative to the internal recuperator is an externally mounted recuperator. For best efficiency, external recuperators are normally of the counter-flow type, with the cold combustion air entering near the flue gas outlet and the preheated combustion air exiting near the flue gas inlet. The preheated air is then connected to the companion burner. Figure 14.8 illustrates a typical externally mounted recuperator configuration. Typical thermal efficiencies on an HHV-available-heat basis for this class of recuperator are 55 to 65%.

The external recuperator can take advantage of greater space availability for heat exchange surface area than the internal designs, leading to the potential for greater thermal efficiency. Flow passages can be made larger on external recuperators for much lower fluid pressure losses in the system. Locating the recuperator externally minimizes the potential of extraction of useful heat from the radiant tube. The advantages of the external recuperator can be offset by the recuperator size, which can be an issue in some installations. Further, external recuperators and their associated hot piping will require external insulation to minimize ambient losses.

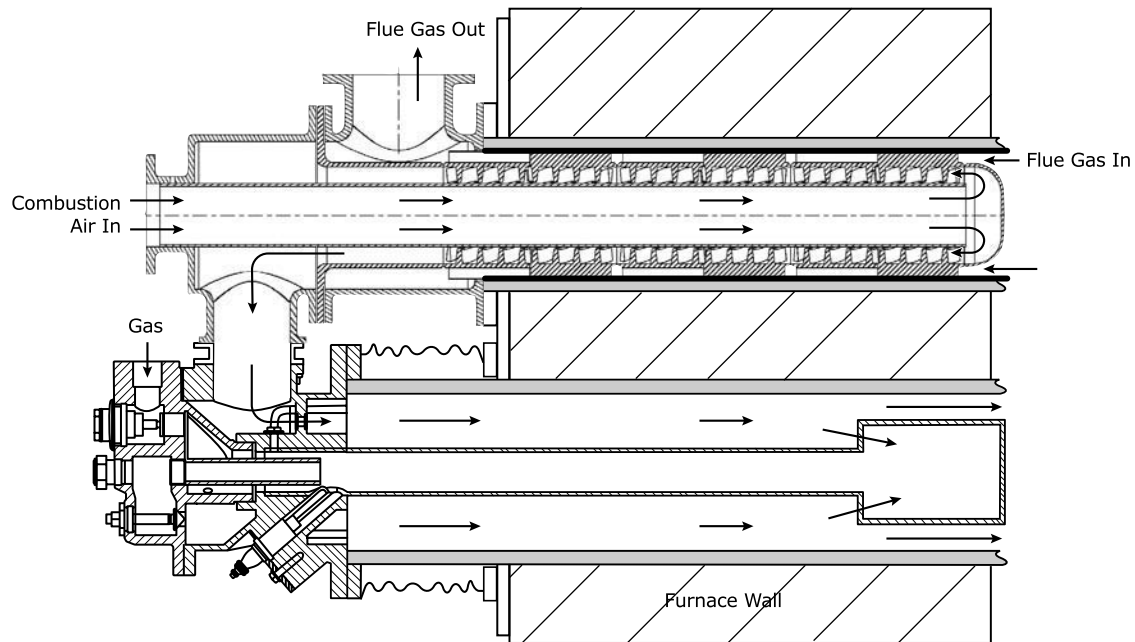


FIGURE 14.7 Insertion type recuperator.

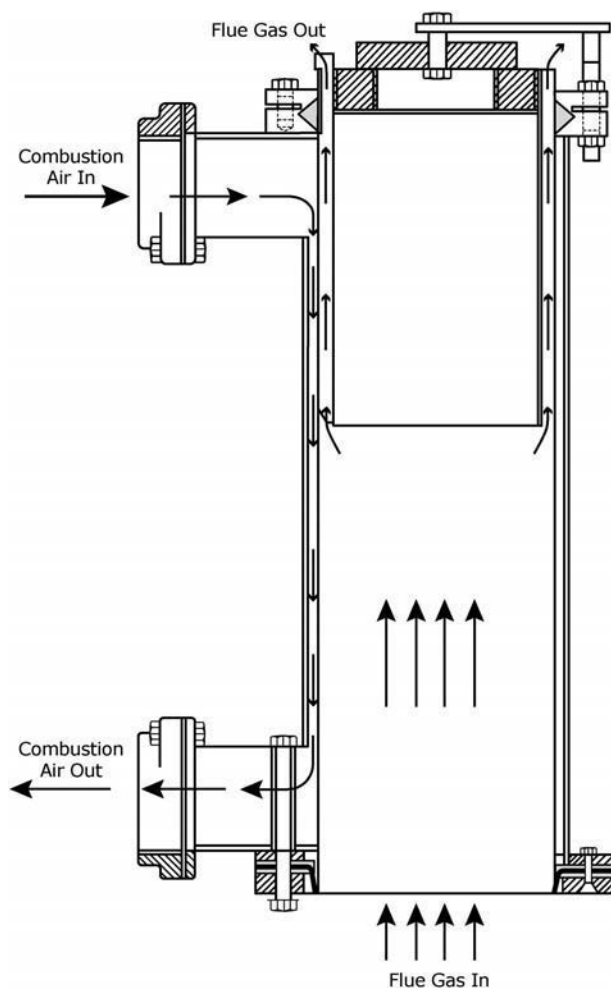


FIGURE 14.8 External recuperator.

14.5 EMISSION REDUCTION TECHNIQUES

Radiant tube burners fire into a confined hot space, limiting the range of techniques available for direct emissions control. External centralized control can be applied but requires that the exhaust collection system be designed and maintained to eliminate in-leakage of ambient “tramp” air if the size and operating cost of the emission post-treatment device is to be minimized. Accordingly, the burners and associated local equipment configurations generally directly incorporate emissions-reduction measures such as the previously mentioned use of recirculation and FGR.

For typical gaseous fuels used in radiant tube burners, NO_x formation is dominated by the thermal mechanism and is therefore dominated by flame temperature. This final flame temperature is a function of the fuel composition, the temperature of the combustion air, and the relative thermal loading on the radiant tube. NO_x reduction techniques for radiant tubes focus on minimizing this flame temperature while fulfilling the thermal requirements of the heating process.

One technique is to break the combustion process up into multiple stages. In this process, an initial portion of the combustion reactants are mixed and combusted. The portion that is combusted is allowed to transfer a portion of the heat of reaction to the process. The resulting combustion products are then delivered to a second combustion zone where either additional air or fuel is

injected, depending on whether the first stage was fuel-rich or fuel-lean. As a portion of the heat of the first reaction has been released, the subsequent flame temperature will be lower than that of a single combustion stage. Multiple stages can be employed by various designs.

A second common technique is to dilute the combustion process by the addition of a nonreactive medium to reduce the ultimate flame temperature and hence NO_x formation. The most common and effective diluent is flue gas from the combustion process and is referred to as flue gas recirculation (FGR). In this technique, combustion products are recirculated into the combustion zone.

There are several variations of the use of FGR. A common approach is to provide the FGR to the burner from a central external blower. In this application, the exhaust from a zone of radiant tubes is connected to a central blower that feeds into the combustion air. This system can be easily applied to a push-pull design with both exhaust and combustion air headers in place. When using FGR, which requires a close coupled exhaust to ensure that the FGR is of “good quality” (low O₂ content), serious consideration must be given to the safety implications of an “explosion-proof system” with no pressure relief — see the comments later in this chapter on safety requirements.

A simpler method is to use an in-burner fuel- or air-driven jet pump to directly inspirate and recirculate the flue gases from the exit of the radiant tube to the burner, relying on the momentum of the jet to provide the motive force to overcome the pressure losses in the recirculating FGR loop.

By design, a recirculating radiant tube provides significant dilution of the combustion zone by hot FGR. Significant quantities of recirculated combustion products (FGR) can be introduced into the combustion zone in such a tube design. However, as the root of the combustion zone is upstream of the point of FGR introduction, substantial thermal NO_x may already have been generated, depending on the specific burner design.

14.6 HEAT TRANSFER CONSIDERATIONS

Optimizing the performance of a radiant tube involves balancing the heat transfer from the flame and products of combustion to the inner surface of the tube, its conduction through the tube wall, and the emission from the outer surface of the tube, while respecting the tube material limitations. Neglecting this balance can result in premature tube failure from thermally induced stresses and material creep. In the authors’ experience, the opportunity costs of lost production generally far outweigh any costs associated with more conservative design of the radiant tube heat transfer area for increased tube life.

Heat is transferred to the inside of a radiant tube by a combination of gas radiation, flame radiation, and convection. Heat is then conducted through the radiant tube material to the outer surface where it is transferred to the load by radiation. Some furnace designs use recirculating fans to improve temperature uniformity and provide convection heating of the load. For these furnaces, there may also be a significant convection component in the transfer of heat from the exterior surface of the radiant tube.

In the case of the more common nonrecirculating tube designs, it is desirable for the burner flame to be long to reduce the potential for local hot spots and to distribute the heat along the length of the tube. The requisite slow heat release flame is created by diffusion mixing of the fuel and air. For the purposes of analysis, it is beneficial to be able to predict the heat release as a function of distance along the axis of the radiant tube. The composition of the combustion products within the radiant tube at any point is actually a mixture of unreacted fuel, combustion air, and reacted combustion products. Ramamurthy et al.⁵ investigated a “fuel burn-up rate” — that proportion of the total input fuel reacted related to the axial distance from the burner nozzle. They expressed this correlation by Equation 14.1.

$$\kappa(z) = 1 - \exp \left[-8.24 \times 10^{-4} \frac{AR^{0.54}}{\Phi^{1.53}} P_{\tau}^{0.41} \left(\frac{z}{D} \right)^{1.6} Re_{\tau}^{-0.08} \right] \quad (14.1)$$

where κ is the fuel burn-up coefficient, z is the distance from the burner outlet, AR is the area ratio (ratio of the annular air inlet port area to the inner circular fuel inlet port area), Φ is the fuel

equivalence ratio, $P_{\dot{v}}$ is the momentum ratio of air to fuel taken at $z = 0$, D is the inside diameter of the radiant tube, and Re_{FT} is the Reynolds number of the combustion products evaluated at adiabatic flame temperature conditions at the stoichiometric fuel/air ratio conditions. This fuel burn-up coefficient can be used to estimate the temperature and composition of the combustion products along the length of a radiant tube.

When the temperature and composition of the combustion products are known within the radiant tube, the rate of heat transfer to the inner surface can be calculated. The heat transfer to the inner surface of the radiant tube is both by convection and by radiation. The local one-dimensional heat convection heat transfer can be expressed by Equation 14.2.

$$q_{\text{conv}} = h(T_{cg} - T_w) \quad (14.2)$$

where h is the local convective heat transfer coefficient, T_{cg} is the local combustion gas temperature, and T_w is the local tube wall temperature.

Ramamurthy et al.⁵ found reasonable agreement with the Gnielinski correlation for the local Nusselt number Nu_D proposed for fully developed noncombusting flows in the range of $2300 < Re_D < 5 \times 10^6$ represented by Equation 14.3.

$$Nu_D = \frac{hD}{k} = \frac{(f_D/8)(Re_D - 1000)Pr}{1 + 12.7(f_D/8)^{1/2}(Pr^{2/3} - 1)} \quad (14.3)$$

where the Darcy friction factor f_D is calculated from Equation 14.4:

$$f_D = (0.79 \ln Re_D - 1.64)^{-2} \quad (11.4)$$

Ramamurthy et al.⁵ also investigated the correlation of the radiant heat transfer to the inner surface of the radiant tube for the one-dimensional case. The Hottel⁶ expression for an isothermal gas in a tube is given as:

$$q_{\text{rad}} = \frac{\epsilon_w \sigma (\epsilon_{cg} T_{cg}^4 - \alpha_{cg} T_w^4)}{1 - (1 - \epsilon_w)(1 - \alpha_{cg})} \quad (11.5)$$

where ϵ_{cg} and α_{cg} are the combustion gas emissivity and absorptivity, respectively, for the combustion gases as described by Hottel, and ϵ_w is the surface emissivity of the interior wall. Application of Equations 12.1 through 12.5 will allow one to estimate the total heat exchange to the inner surface of the tube for a given a tube temperature profile.

Radiant tubes are generally used in chambers where very low or static flow conditions result in low convective heat transfer coefficients. Convective heat transfer is typically ignored for the case of a furnace chamber without forced circulation in view of the high radiation component. Radiant tubes are commonly located adjacent to a refractory furnace wall or roof. It is common to locate several tubes adjacent to a surface, thus forming a tube bank. To reduce the complexity of the evaluation of the radiant heat exchange, Hottel⁷ suggests a gray plane method where the refractory backed bank of tubes can be replaced by an equivalent “gray plane” having an effective emissivity of ϵ_p where the subscript P refers to an effective plane area. The total radiation exchange presented by Hottel for the effective plane is as given in Equation 14.6:

$$\dot{Q}_{\text{net}} = A_p (E_p - E_T) \left[\frac{1}{\frac{1}{F_{PT}} + \frac{C}{\pi D \epsilon_T}} \right] \quad (11.6)$$

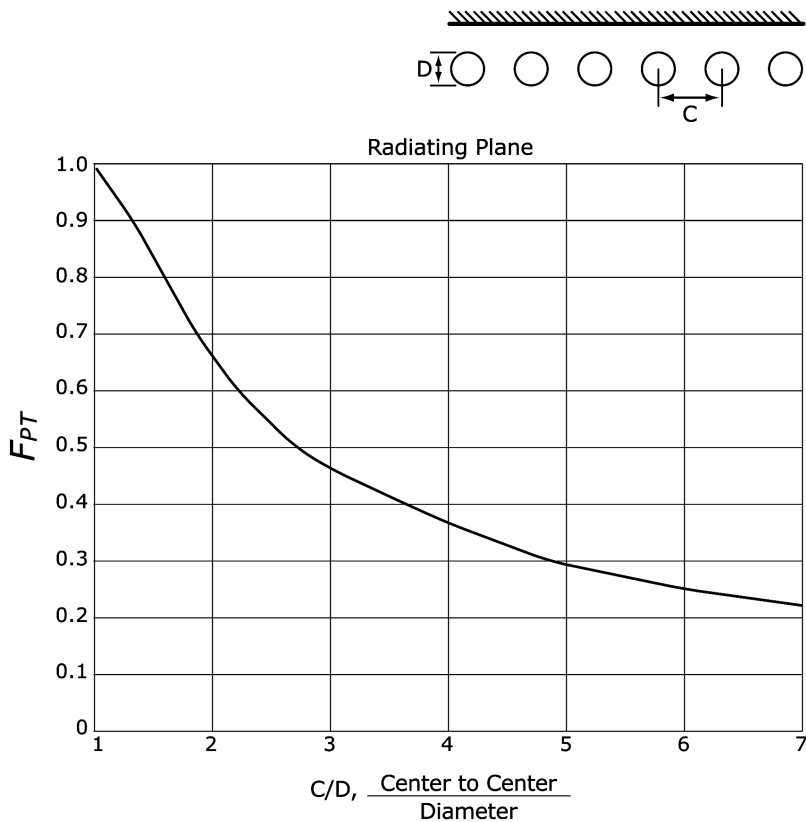


FIGURE 14.9 Total view factor for refractory backed single row of tubes. (Adapted from Hottel, H.C. and Sarofim, A.F., *Radiative Transfer*, McGraw-Hill, New York, 1967. With permission.)

where A_p is an effective black plane parallel to the tube surface and refractory covered wall, and E_p and E_T represent the emissive power for the black plane and the tube surface, respectively. The term in square brackets represents an interchange factor between the refractory-backed tube plane and the effective black plane A_p . Within the square brackets, F_{PT} is the total view factor, C is the center-to-center spacing of the tube banks, ρ_T is the tube reflectance, and ϵ_T is the tube surface emissivity.

F_{PT} depends on the ratio of tube center-to-center spacing to the tube diameter. For a single row of tubes in front of a refractory wall, Hottel presents F_{PT} as shown in [Figure 14.9](#).

14.7 RADIANT TUBE MATERIAL CONSIDERATIONS

There are two major classes of materials that are typically applied to radiant tubes. Metallic radiant tubes of Fe-Cr-Ni alloys in various grades are commonly used on process temperatures up to 1900°F, although some proprietary grades are capable of slightly higher temperatures. Ceramic radiant tubes capable of surface temperatures up to 2450°F are utilized to process temperatures up to 2250°F. Most commercially available ceramic radiant tubes are produced from the SiC (silicon carbide) material family.

Interested readers are encouraged to consult specialist texts for detailed information on the materials aspects of radiant tubes. Some general information follows.

14.7.1 METALLIC RADIANT TUBES

Selection of metallic radiant tube material for optimal performance requires knowledge of the operating temperature of the tube material and the atmosphere of the heating process. These influence creep rate, the rate of corrosion, and the general long-term stability of the material.

Creep is the slow deformation of the tube material resulting from applied stress. Creep failure can result in reduction of the tube cross section and burner flame impingement leading to tube rupture and is a major element in the failure of radiant tube supports. A material's creep strength typically limits the ultimate performance of a radiant tube design because it is much less than the yield strength of the same material. Because creep strength is logarithmic with temperature, small increases in tube temperature at normal operating conditions result in significant reduction of creep strength.

Radiant tubes are subject to the stresses of the applied mechanical loads and from thermally induced stress. Classical mechanical design methods can be used to estimate mechanical loads from the weight of the radiant tube itself and the concentrated loads from the radiant tube's supports. Thermally induced stresses result from hot spots with high local heat flux rates, poor overall temperature distribution along the length of the radiant tube, and periodic loads due to thermal cycling.

High temperature gradients from the inside to the outside of the tube result in thermally generated hoop stresses in the tube wall. In areas of high heat flux loading, consideration must be given to this stress and attempts made to minimize the wall thickness, while at the same time considering the stress due to applied mechanical loads as the mechanical and thermally generated loads must be superimposed upon each other.

Additional thermal stresses result from uneven axial and circumferential temperature distributions. However, Irfan and Chapman⁸ have shown that uneven temperatures along the length of the tube have to be significant in order to result in significant thermally induced stresses. Likewise, unevenness in circumferential temperatures generally does not create significant stress unless the local temperature differences exceed 75°F. This is not to say that there is no loss in strength resulting from these temperature maldistributions, and every attempt should be made to avoid them. Moreover, uneven temperature distributions lead to high tube surface temperatures (creep failure) and less effective use of the available tube surface area.

Corrosion results from interaction of the tube material with the furnace atmospheres and combustion products, starting on the outer and inner surfaces, respectively, and then migrating through the grain structure deeper within the tube wall. Corrosion will result in the loss of mechanical strength and lead to ultimate failure of the radiant tube. Therefore, it is very important that the atmospheres to which it will be exposed be considered in the selection of the radiant tube material.

Metallic radiant tubes are alloyed with high levels of chromium, typically on the order of 25%(wt.), to provide resistance to oxidation. The chromium reacts with local oxygen to form a protective layer of chromium oxide (Cr_2O_3) on the surface of the tube. This protective layer reduces and eventually effectively prevents further direct combination of the metal with the atmosphere. The protective oxide film is, however, subject to breakdown from the expansion and contraction of the base metal, causing exposure of the underlying material to the atmosphere. Minor alloying with silicon or aluminum improves the oxidation resistance of nickel-based alloys, especially when added in combination with rare earth metals.⁹

Radiant tubes frequently operate in carburizing atmospheres. In these processes, carbon is adsorbed and diffused into the metal, where, at sufficiently high temperatures, chromium, iron, and nickel carbides are formed. The precipitation of chromium carbides removes chromium from the alloy matrix, potentially increasing oxidation. Nickel carbides increase the hot strength of the material but decrease ductility. Silicon additions delay the carburization by preventing carbon absorption, and stable carbide formers such as tungsten can be added to increase resistance to the

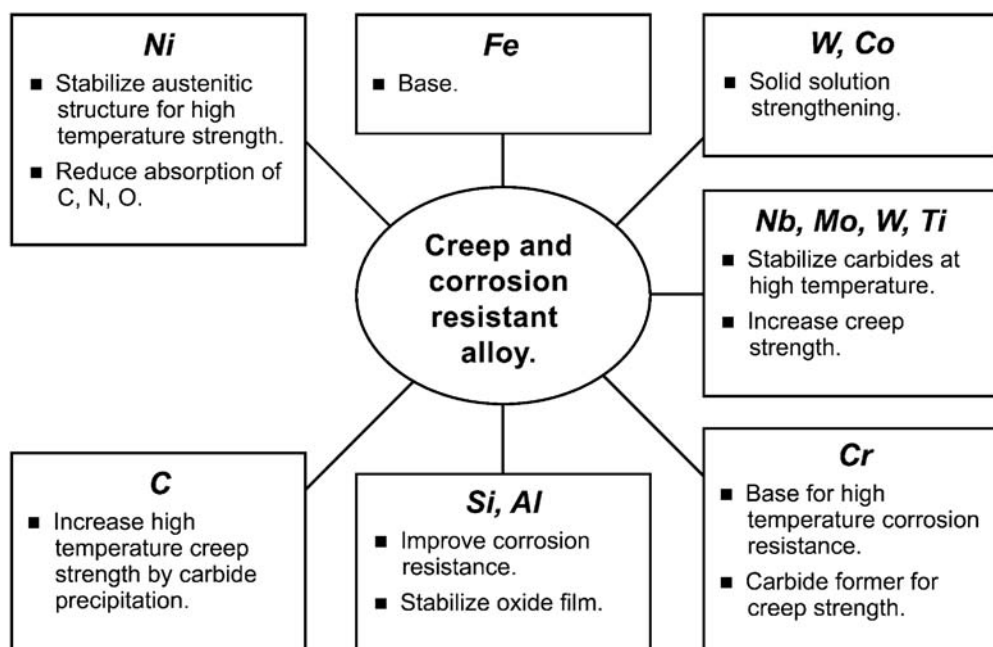


FIGURE 14.10 Major alloying elements in cast heat resistant alloys. (Courtesy of Kubota Metal Corporation, Fahramet division.⁹ With permission.)

effects of carburization. Larger concentrations of chromium reduce the diffusion rates of both carbon and chromium in nickel-based alloys.

Nitridation can also occur at temperatures in the range of 1800°F with nitrogen-containing or ammonia-based atmospheres. High nickel content alloys with low iron content perform well under nitriding conditions.

Figure 14.10 provides an overview of common alloying elements and the result of the use of each. It is recommended that the tube material manufacturer be consulted for the particular application.

14.7.2 CERAMIC RADIANT TUBES

The inherent heat flux limitations of metallic radiant tubes have led to increased popularity of ceramic radiant tubes over the past 20 years, and spurred development of materials and structures for this application. Typically constructed from the SiC family of ceramics, such radiant tubes exhibit great resistance to creep, improved resistance to carburization, and higher temperature capabilities. Current commercial ceramic radiant tube suppliers limit their tube material to 2450°F and process temperatures to 2250°F.

Several forms of SiC are commercially used in the manufacture of radiant tubes. The common forms are Nitride Bonded, Reaction Bonded, and Composites, each term referring to the method of bonding of the larger SiC particles within the matrix. Nitride-bonded SiC uses silicon nitride to bond SiC particles together. It has good resistance to oxidation but has lower thermal conductivity than its reaction bonded counterpart. The reaction bonded SiC is formed by covalently bonding a range of sizes of SiC particles together. This material is very durable, providing excellent high-temperature strength, excellent creep resistance, and good resistance to oxidation. The composite SiC materials integrate SiC particles in a matrix of another material such as silicon. Some examples of this class of material use reinforcement by SiC whiskers or other, similar high-temperature “fiber” materials.

14.8 APPLICATION CONSIDERATIONS

The application of radiant tubes to a specific furnace involves consideration of many factors. The required heat input to the furnace, available space for location of the tubes, required tube temperature to provide the heat transfer to the load, support locations for the radiant tube, material of the radiant tube, and overall costs must all be considered. Future operating and maintenance costs must be factored in to optimize the combined radiant tube and burner selection.

The common starting point is to establish the required heat input to the load and to balance losses from the furnace walls, to a flowing atmosphere, to water cooling, etc. Further, if the application is a batch-type process, the required heat-up time must also be considered. All these factors contribute to what is referred to as the *required available heat* for the process.

Once the required available heat is determined, attention must turn to the space in which the radiant tubes are to be located, giving consideration to the operating temperature of the furnace and the selected tube material. The common approach to determining the required radiant tube surface area is to apply heat flux guidelines. If the furnace temperature is high, heat flux from the tube must be limited to keep tube temperature down and provide acceptable life. In lower temperature furnaces, or if heat transfer from the surface of the tube is enhanced by convection, it may be possible to use high heat flux rates without significant compromise in tube life.

Typically, the heat flux is evaluated as an average over the length of the radiant tube. This average is determined by dividing the required available heat by the number of radiant tubes, and then by the exposed (inside the heating chamber) surface area of the radiant tubes. The heat flux rate is then compared to common values based on past experience with the particular process temperature and tube material. If the heat flux is higher than the experience-based numbers, either the exposed tube area must be increased or higher-grade tube material, allowing higher tube operating temperatures, must be considered. In some cases, the furnace chamber size may need to be increased to allow more space to install the radiant tubes. Table 14.1 shows common heat flux values for metallic radiant tubes. The use of ceramic radiant tubes can increase these values to 100 to 150 Btu/in², depending on the process temperature.

Although the average heat flux approach to specifying radiant tubes is commonly used throughout industry, this approach does not factor in the influence that radiant tube temperature distribution has on the heat flux.

Considerable research and development has obtained more even temperature distribution along the length of the tube by eliminating hot spots or reducing them to a minimum. The resulting consistent heat flux with respect to length allows maximization of tube life by operating with the lowest possible tube temperature for a given total heat flux. For a cold-air or recuperative tube, the challenge is to extend the combustion reaction such that it releases heat evenly along the tube length. The reversing action of the burners mounted at each end of a regeneratively fired tube achieves this objective as a natural outcome of the design, as previously mentioned.

Highly rated single-ended tubes are frequently rotated 180° as part of normal maintenance to allow gravity to correct the effects of sagging. This is generally not possible or practical with “U” or “W” tubes, enhancing the need for even temperature distribution to optimize the life of the tubes.

TABLE 14.1
Commonly Used Heat Flux Rates for Metallic Radiant Tubes

U-Tube Leg Length (ft)	Furnace Temperature (°F)		
	1100–1300	1300–1600	1600–1750
< 3	100	60	30
3 to 6	80	50	20
6 to 10	60	40	10

14.9 COMBUSTION CONTROLS FOR RADIANT TUBE BURNERS

Combustion controls play an important role in the successful application of radiant tubes. Stated elsewhere in this chapter is the importance of maintaining uniform radiant tube surface temperature. Because the mixing energy of a burner is a function of the burner firing rate, it is extremely important that the turndown requirements and low firing rate conditions are reviewed. Moreover, on preheated air burner systems, the temperature of the preheated air most likely increases as a function of turndown — as the flows go down, the heat recovery device efficiency increases. If consideration is not given to this aspect, local overheating of the radiant tube at low firing rates may result in early failure of the tube.

14.9.1 INPUT AND AIR/FUEL RATIO CONTROL

The most prevalent air/fuel ratio control system for radiant tubes uses the pressure-balanced (“cross-loaded”) regulator. The most common for ambient air burners is the single-diaphragm cross-connected regulator shown in [Figure 14.11](#). In this system, the air pressure to the burner is applied to one side of a diaphragm. The regulator’s outlet (downstream) gas pressure is applied to the other side of the diaphragm. As the pressure of the air varies, the regulator opens until the gas pressure “balances” the force of the air pressure plus whatever spring forces that are applied to the regulator diaphragm. Because the flows of both the fuel and the air follow a square-root relationship with their pressures, the two flows will modulate and maintain the desired ratio proportions.

In preheated air systems, the combustion air flow is both a function of pressure and temperature. For this application, a double-diaphragm regulator [Figure 14.12](#), uses differential pressures produced from the air and gas flows to provide its valve positioning forces. A restricting orifice is installed in the combustion air line and the pressure of upstream and downstream taps connected to each side of the regulator air diaphragm. An opposing force is created by connecting a differential pressure from a restriction in the fuel line to a second diaphragm. This differential force is typically created by installing an adjustable orifice downstream of the regulator.

The double-diaphragm regulator’s air flow differential device is installed upstream of the air preheating device where the air density is constant. The signal to the regulator will thus always be proportional to the air flow, and not affected by the change of the temperature of the air as it passes through the air preheating device.

There are several system heat input control philosophies applied to radiant tubes that allow good fuel/air ratio control and aid in maximizing the potential life of the tubes. Systems are typically classified as modulating, high/low, on/off, and combined mode.

In a modulating system, the fuel and air inputs are controlled (“modulated”), as required, to maintain the required process temperature — to match the radiant tube’s heat input to the load requirements. This modulation occurs down to a minimum firing rate, typically between 3:1 and 5:1

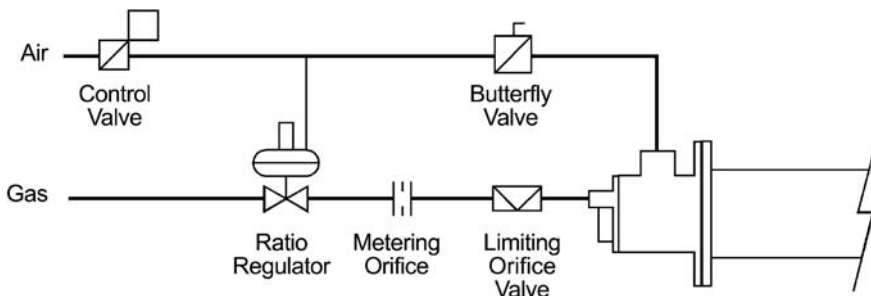


FIGURE 14.11 Pressure-balanced regulator system.

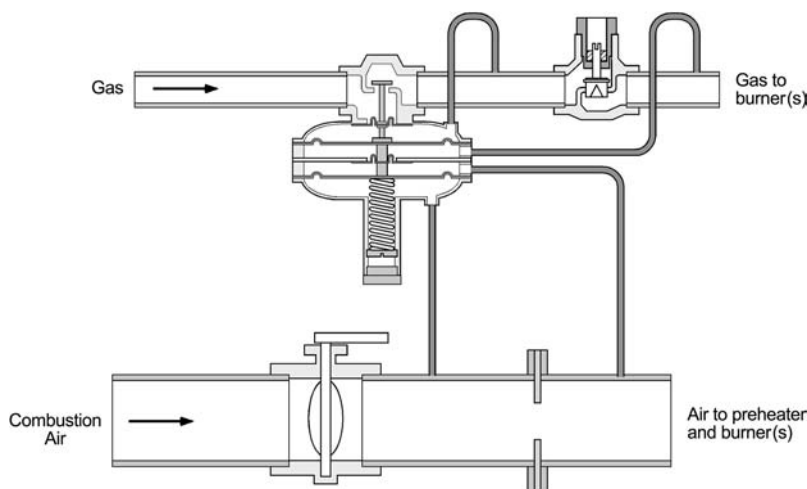


FIGURE 14.12 Double diaphragm regulator.

turndown from the maximum firing rate. Modulating systems have the inherent disadvantage of poorly distributing the heat release at the minimum firing rate where the mixing energy is minimal.

With high/low systems, the firing rate is either at the maximum firing condition (high) or the minimum firing condition (low), depending on the demand of the process temperature controller, with minimum transition time between these two states. If the process temperature is not met, the control places the burner in the high fire mode. Once the temperature is satisfied, the control drives the system to the low fire condition. The high/low control system minimizes the variation in thermal profiles between the two extreme conditions by allowing the two extreme conditions to be set in a predictable manner. However, for many processes, especially in batch operations, the heat input at an acceptable low firing rate may exceed the minimum process requirements.

The on/off control philosophy is similar to high/low as described above, with low rate replaced by turning the burner off. Using the off mode in place of the low fire mode allows a theoretically infinite turndown to be achieved. However, this control mode requires that the burner be re-ignited by a pilot or direct spark igniter each time the input controller calls for heat. A variation is a combination of high/low and on/off known as high/low/off. This allows for the infinite turndown capabilities of the on/off method to be achieved but requires less frequent re-ignition of the burner.

“Pulse firing,” a form of on/off control, can be used on cold-air or recuperative systems to control input with simultaneous enhancement of tube temperature uniformity by firing only at the burner’s maximum rate. In this control mode, the relative durations of the high fire and off pulses will be varied to achieve the desired heat input rate.

Combined control modes can mix any of the modes discussed above.

Regenerative radiant tube burner systems use a variation of pulse firing for input control. Both the combustion air and the fuel must cycle on and off frequently to perform the reversing action required of the burner operation. Pulse width modulated input control can be superimposed on the reversing action. In these systems, fast-acting pneumatic on/off valves are used to instantly turn on both the air and fuel to their maximum flow or off conditions. To compensate for natural system dynamics, it is common for the fuel opening to lag the combustion air opening, and the fuel closing to precede the combustion air closing, by a short interval, ensuring that the burner does not fire in a rich condition. Pulse firing, when applied to regenerative burners, ensures the best possible radiant tube temperature distribution as the burners fire only at the maximum design rate.

14.9.2 FLAME SUPERVISION

Note: This section is not intended to act as a specific guide to the use of flame supervision systems for radiant tube or any other burners. The reader is referred to the applicable international, national, local, and insurance industry codes, standards, requirements, etc. for appropriate information on this subject.

There are many radiant tube combustion systems installed that do not incorporate provisions for flame supervision. At the time of this writing, the U.S. National Fire Protection Agency (NFPA) covers the installation of metallic radiant tubes without flame safety when the system is “explosion proof.” For the explosion-proof determination, the entire combustion and radiant tube system should be reviewed, including any exhaust collection system that might be connected to the radiant tube. If any part of the system is determined not to be “explosion proof,” then an appropriate flame proving system should be installed. System designers should recognize that many areas in the world require flame proving on all systems and that some users require flame proving as a matter of practice.

Two alternatives are commonly employed when flame supervision is required with radiant tube systems. The most common is the use of a standard ultraviolet sensor or a flame rod to monitor the presence of a flame. In cases where direct flame detection may not be feasible, an acceptable alternative may be to monitor the exhaust system for the presence of combustible gases with an appropriately selected exhaust gas analyzer, usually limiting the operation to a conservative percentage of the lower explosive limit (LEL) of carbon monoxide before safety shutdown.

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