LECUTRE 31: Refractory and gas radiation

Key words: Radiation, View factor, Gas radiation, Convection

Thermal role of refractory surface:

Refractory surfaces play an important role in keeping the furnace temperature constant by minimizing the heat loss to the surrounding. Consider a furnace which is at an average temperature $T_F$ and is enclosed by the refractory. The intensity of radiation striking a refractory surface is $5.67 \times \left(\frac{T_F}{100}\right)^4$.

Of this incident radiation, $5.67 \times \epsilon_R \times \left(\frac{T_F}{100}\right)^4$ is absorbed and $5.67(1 - \epsilon_R)\left(\frac{T_S}{100}\right)^4$ is reflected. Here $\epsilon_R$ is reflectivity. Due to incident flux the refractory surface is heated to a temperature $T_R$ and refractory surface will also emit energy at the rate of $5.67 \times \epsilon_R \left(\frac{T_R}{100}\right)^4$ W/m². At steady state, heat balance is:

Heat loss/unit area through the furnace wall to the surroundings = rate of absorption of heat from the furnace — Rate of emission of radiation back into the furnace

$Q \frac{A}{A} = 5.67 \times \epsilon_R \left[\left(\frac{T_F}{100}\right)^4 - \left(\frac{T_R}{100}\right)^4\right] W/m^2$ \hspace{1cm} (1)

According to equation 1 heat loss to the surrounding depends on the inside surface temperature of the refractory, furnace and surrounding temperatures and the thermal resistance offered by the refractory wall. Consider a 0.23m firebrick wall of thermal conductivity 1.731 Wm⁻¹ K⁻¹ at 1273K inner surface temperature of the refractory. Heat loss per unit area from this wall to the surrounding is calculated to be 7338W/m² for the surrounding temperature of 298K. Assuming reflectively of the refractory $\epsilon_R = 0.8$ and heat loss as 7338W/m², furnace temperature can be calculated to be 1292K which is only 19°C higher than the inner surface temperature of the refractory. The difference between the inner surface temperature of the refractory and the furnace temperature can be reduced further by increasing the thickness of the wall and using refractory of lower thermal conductivity. This calculation suggests that the inner surface temperature of the refractory wall is close to the average temperature of the furnace.

The role of refractory surface can be illustrated. For the firebrick wall the total incident flux is 126390 W/m². Of this incident flux, 25728 W/m² is reflected back into the furnace and 7338 W/m² is
lost to the surrounding through the wall of the refractory. The balance 93776 W/m² is absorbed and re-radiated by the refractory back into the furnace. Thus refractory surface is an efficient reflector (75%) but the mechanism of reflection is absorption and re-radiation.

**Effect of refractory surface on the heat flow between source and sink:**

Consider a system in which three surfaces \(A₁, A₂\) and \(A_R\) forms an enclosure as shown in the figure. Each surface is characterized by its own uniform temperature \(T₁, T₂\) and \(T_R\) respectively. Also \(A₁\) and \(A₂\) surfaces are shaped such that each cannot use itself. The following assumptions are made:

1. Enclosure is adiabatic: Refractory surface \(A_R\) does not allow any heat which means all incident flux is re-radiated or reflected.
2. Surfaces \(A₁\) and \(A₂\) are black.

Heat flows directly from \(A₁\) to \(A₂\) and to \(A_R\). As a result of incident radiation, \(A₂\) is heated and also begins to radiate. Of the total energy leaving from \(A₁\) to \(A₂\) fraction is absorbed by \(A₂\) and is controlled by \(F₁\). \(F₁\) is view factor and corresponds to faction of total energy leaving \(A₁\) which is absorbed by surface2. The remaining fraction \((1 - F₁)\) is intercepted by surface \(A_R\).

After a while surface \(A₂\) is heated up. Of the total energy leaving the surface \(A₂\), \(F₂\) is absorbed by surface \(A₁\) and \((1 - F₂)\) is intercepted by \(A_R\). \(F₁\) and \(F₂\) are related by

\[
F₁ \times A₁ = F₂ \times A₂ \quad (2)
\]

\[
F₂ = F₁ \times \frac{A₁}{A₂} \quad (3)
\]
Direct heat exchange between $A_1$ and $A_2$

$$Q_{1-2} = 5.67 \times A_1 \times F_1 \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_2}{100} \right)^4 \right]$$  \hspace{1cm} (4)

Heat exchange between $A_1$ to $A_R$ and $A_2$ to $A_R$ is

$$Q_{1-R} = 5.67 \times (1 - F_1) A_1 \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_R}{100} \right)^4 \right]$$ \hspace{1cm} (5)

$$Q_{2-R} = 5.67 \times \left(1 - \frac{F_1 \times A_1}{A_2} \right) A_2 \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_R}{100} \right)^4 \right]$$ \hspace{1cm} (6)

Since there is no heat flow to the refractory $q_{1-R} + q_{2-R} = 0$.  \hspace{1cm} (7)

Equations 4 to 7 can be solved to give

$$T_R = \left[ \frac{(A_1 - F_1 A_1) T_1^4 + (A_2 - F_1 A_1) T_2^4}{A_1 + A_2 - 2 F_1 A_1} \right]^{0.25}$$ \hspace{1cm} (8)

$$Q_{1-R} = -Q_{2-R} = 5.67 \frac{(A_1 - F_1 A_1) + (A_2 + F_1 A_1)}{A_1 + A_2 - 2 F_1 A_1} \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_2}{100} \right)^4 \right]$$ \hspace{1cm} (9)

Equation 9 is the additional indirect heat exchange between $A_1$ and $A_2$ as obtained by reflection and re-radiation from surface $A_R$.

Total heat flow $= Q = Q_{1-2} + Q_{1-R}$ \hspace{1cm} (10)

By equation 4, 9 and 10

$$Q = 5.67 F_C A_1 \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_2}{100} \right)^4 \right]$$ \hspace{1cm} (11)

Where $F_C$ is a composite geometrical factor for the closed system of two black surfaces $A_1$, $A_2$ and $A_R$.

$$F_C = \frac{A_2}{A_1} \frac{F_1}{1 + \frac{A_2}{A_1} - 2 F_1}$$ \hspace{1cm} (12)

It is to be noted that in the above derivation we have identified furnace to consist of source, sink and refractory as independent entities and each one is at uniform temperature. However, in practical situations, it is not very often easy to separate the source, sink and refractory and the temperatures may be far from uniform. Thus a careful engineering judgment is necessary for accurate calculations.

**Effect of emissivity of surfaces**

For most general calculations, it is necessary to take into account the effect of emissivities of the source and sink surfaces. If $A_1$ and $A_2$ are not black but gray surfaces with emissivities $\varepsilon_1$ and $\varepsilon_2$

$$Q = 5.67 \times F A_1 \left[ \left( \frac{T_1}{100} \right)^4 - \left( \frac{T_2}{100} \right)^4 \right]$$ \hspace{1cm} (13)
Where

\[ F = \frac{1}{\frac{1}{\epsilon_1} (\frac{1}{\epsilon_2} - 1) + \frac{A_1}{A_2} \frac{1}{\epsilon_2 - 1}} \]  

(14)

Equation 14 reduces to \( F = F_c \) when \( \epsilon_1 = \epsilon_2 = 1 \)

The following procedure may be adopted to calculate the rate of heat transfer between two surfaces within a furnace:

Determine or estimate the temperatures of surfaces \( A_1 \) and \( A_2 \) and emissivities \( \epsilon_1 \) and \( \epsilon_2 \)

Find \( F_1 \) which is the fraction of energy from \( A_1 \) intercepted by \( A_2 \)

Calculate \( F_c \) by equation 12 and then determine \( F \) by equation 14

Calculate \( Q \) by equation 11

**Gas radiation**

In gas radiation, one has to distinguish between the gases which are transparent to radiation and which emit and absorb radiations. All diatomic gases like nitrogen and oxygen are transparent to radiation that means they neither emit nor absorb radiation at low temperatures. Whereas, tri-atomic gases like \( \text{CO}_2, \text{H}_2\text{O}, \text{SO}_2 \) absorb and emit radiation to a considerable degree so that they are important in heat exchange.

Absorption and emission of gases differ from liquids and solids in the following ways.

i. Gases emit and absorb radiation within a narrow band of wavelength and

ii. Emission and absorption take place through the body of the gas. No. of modules of a gas is important for radiation. No. of molecules of gas at a given temperature is proportional to partial pressure of gas and size of the body (\( L \)) of the gas, where.

\[ L = 3.5 \frac{\text{volume}}{\text{surface area}}. \]  

(15)

Heat transfer between gas and enclosure can be given by

\[ Q = \frac{1 \times 5.67}{\epsilon_G} \frac{1}{\epsilon_S} A_G \left[ \left( \frac{T_G}{100} \right)^4 - \left( \frac{T_S}{100} \right)^4 \right] \text{W/m}^2 \]  

(16)

\( A_G \) is area of enclosure, \( T_G \) and \( T_S \) are temperatures of gas and surfaces respectively. \( \epsilon_S \) is emissivity of surface and \( \epsilon_G \) is emissivity of gas.
If the gas is a mixture of say CO₂ and H₂O then

$$\varepsilon_G^1 = \varepsilon_{CO₂}^1 + \varepsilon_{H₂O}^1.$$  

\(\varepsilon_{CO₂}^1\) or \(\varepsilon_{H₂O}^1\) can be determined as follows.

1. Determine the partial pressure of CO₂ or H₂O in the gaseous mixture
2. Determine the value of L by using equation 15.
3. Make the P × L either in atm × m or atm × ft.
4. Use the standard chart for emissivity of CO₂ and H₂O vs. temperature. In these chart \(\varepsilon_{CO₂}^1\) or \(\varepsilon_{H₂O}^1\) is plotted against temperature for different P × L value.
5. Determine \(\varepsilon_G^1\) by adding the emissivities of the individual gas.

**Flame radiation:**

In fuel fired furnaces and other furnaces like reverberatory furnace, the principle mechanism of heat transfer is from the flame to the charge. The flame consists of the products of combustion. By considering sink as the surface area of the charge \(A_C\) and source as the flame of area \(A_{FL}\) and assigning to the flame an equivalent body of gray emissivity \(\varepsilon_G^1\), the rate of heat flow from the flame to the charge can be written as:

$$Q = 5.67A_C \frac{1}{\tau_G^1} A_C \left( \frac{1}{\tau_G^1} \right) \times \left[ \left( \frac{T_G}{100} \right)^4 - \left( \frac{T_S}{100} \right)^4 \right]$$  

(17)

Note \(\varepsilon_G^1\) is gas emissivity without solid particles whereas \(\varepsilon_G^2\) is emissivity of gas with solid particles. Also \(\varepsilon_G^2 > \varepsilon_G^1\).

Heat flow from flame to charge for a given area of source and sink, and temperatures of the source and sink depends on emissivity of the flame. In a non-luminous flame such as pale blue or visually transparent flames obtained by burning hydrogen and other gaseous fuels, the radiation and convection are the principle mechanisms of heat transfer. In combustion of oil and pulverized coal, the flame contains clouds of fine particles of ash and carbon in addition to the products of combustion. These particles increases the emissivity of the flame and as a result heat transfer is enhanced. The furnace operator tries to make the flame luminous over as much length of the furnace as possible. For large pulverized coal flames, the emissivity could approach to unity which results in maximum rate of heat flow from the flame to the charge. By putting \(\varepsilon_G^2 = 1\) in the equation 17, we get the expression for maximum heat flow:

$$Q_{max} = 5.67 A_C \varepsilon_G \left[ \left( \frac{T_F}{100} \right)^4 - \left( \frac{T_S}{100} \right)^4 \right]$$  

(18)

According to equation 18, the maximum heat transfer from the flame requires a knowledge of actual flame temperature in the furnaces.
Reference:

2. R. Schumann: Metallurgical Engineering, volume 1 engineering Principles