1 **An approach to estimating flame radiation in combustion chambers**
2 **containing suspended-particles**

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Abstract

Thermal radiation often plays a key role in heat transfer between flames and the surrounding environment. Therefore, reliable measurement of flame radiation is indispensable in the study of combustion. While the measurement of radiative flux from flames is straightforward in an open environment, it is challenging to do so in a confined chamber, due to the influence of the walls. Radiation emitted by the walls and the reflection of flame radiation off the walls can interfere with measurements of flame radiation. For wall surfaces that are at high temperature or highly reflective, the error in the flame radiation measurement can be significant.

In this paper, a theoretical analysis was carried out to study the contribution of wall emission and wall reflection to the incident radiation measured by radiometers. The results reveal that for a typical lab-scale air-fired flame, the contribution of wall emission and wall reflection to the measured incident radiation can be as high as 55% for a refractory-lined wall and 40% for a water-cooled wall. A convenient approach is proposed to distinguish flame radiation from the measured incident radiation. The approach is based on measurements of surface incident radiation and net radiative heat flux through the wall. The approach can be applied to optically thin systems (most lab-scale and some pilot-scale atmospheric pressure combustion systems) and optically dense systems (typically pressurized large-scale combustion systems). CFD simulations were also performed for two coal-fired combustors, one which represents optically thin and the other optically thick systems, to validate the theoretical model and the approach to estimating flame radiation. Good agreement was observed between CFD results and the

theoretical model results, and the proposed approach to estimation of flame radiation was found to have reasonable accuracy.

Key words: thermal radiation, flame, combustion, wall emission, wall reflection

1. Introduction

Thermal radiation quite often plays an important role in heat transfer between flames and the surrounding environment. In fact, flame radiation caused by gas and particles is the dominant heat transfer mode in many combustion applications, such as industrial furnaces and coal-fired combustion boilers [1-5]. Butler and co-workers performed laboratory-scale experiments to understand heat transfer by thermal radiation in a pulverized coal-fired reactor, and they reported that the thermal radiation from coal flames can be as high as 90% of the total heat transfer [6, 7]. Moreover, radiation has a significant influence on flame ignition, flame structure, and flame spread [1]. Park and Tien showed that the absorption of incident radiation by the gas phase in the near-burner region is one of the critical mechanisms responsible for gas-phase ignition of solid fuels when exposed to strong external thermal radiation [8]. A reliable measurement of flame radiation is indispensable to the study of combustion. As new technologies are developed to address environmental challenges associated with flames and combustion, the need for flame radiation measurement will become increasingly important, especially for the design and optimization of advanced combustion systems [9, 10].

While the measurement of radiation coming from flames is straightforward in an open environment [11], it is challenging to do so in a confined chamber, due to the influence of the walls. Radiation emitted from the walls and flame radiation that is reflected off the walls can interfere with the measurement of flame radiation. The difficulty of distinguishing flame radiation from the measured incident radiation has been emphasized in several studies [12-15]. Also, determination of flame radiation through computational calculations remains a subject with high uncertainty in practical combustion systems [16-19]. This is principally due to the complexity associated with determining the key radiative properties, such as the local absorption and scattering coefficients of optically active gaseous components and, more importantly, those of various particulates (coal, char, ash, and soot) [20-24].

As shown in Fig. 1, the surface incident radiation received by a radiometer at the inner surface of a combustion chamber has three components:

- (1) radiation coming from the flame and combustion products (i.e., from the particles and optically active gases),
- (2) radiation emitted from the walls, and
- (3) radiation reflected by the walls.

In addition, in all three components the received radiation is partly absorbed and scattered by the gas/particle medium on its way to the wall. The first component above is referred to as “flame radiation” in this study.

The surface incident radiation, G_{total} , measured by a radiometer can be expressed as

$$G_{total} = E_{flame} + G_{wall} , \quad (1)$$

where E_{flame} is the flame radiation (i.e. radiation directly coming from the high temperature combustion products and partly absorbed and scattered by the medium on its way), and G_{wall} is the combined contribution of wall emission and wall reflection to the surface incident radiation. As will be shown later, in many combustion applications, G_{wall} is comparable to or even higher than E_{flame} , especially when the wall surfaces are at high temperature or are highly reflective. The strong impact of the wall on the radiative measurement can lead to difficulties in fundamental studies of combustion. For example, the difference in flame radiation caused by varying operating parameters, such as the stoichiometric ratio and swirl number, can be buried in the dominating effect of wall emission and/or wall reflection when the wall surfaces are hot and/or reflective.

In order to distinguish flame radiation from measured incident radiation, some researchers measured incident radiation with and without the flame (i.e., immediately after the flame is shut off) and used the difference between the two to approximate flame radiation [12]. While this approach addresses the influence of wall emission, it does not take into account of the influence of wall reflection, which, in some circumstances, can be significant, as will be shown later.

Others tried to block most of the radiation caused by the walls [13, 14, 25, 26] by utilizing a narrow angle radiometer together with a non-reflecting, cold surface positioned on the wall opposite the radiometer. While this approach can minimize the

effects of wall emission and reflection, it provides only line-of-sight information about the radiative characteristics in the chamber. However, in most circumstances, hemispherical radiative heat flux is of more interest because it is directly related to the design and operation of combustors. For example, in the power industry, material selection for water-tube walls and the arrangement of heat exchangers are determined by the magnitude of hemispherical heat flux in the combustion boilers. In addition, hemispherical radiative flux measurements provide information that is not biased by the non-homogeneous distribution of particle number density in the combustion chamber, unlike the narrow angle measurement, which may vary wildly with directions [27]. While a number of studies have reported measurement of hemispherical radiation in various combustion chambers [3, 4, 12, 28, 29], and other studies have tried to design and improve probes for hemispherical radiative measurement [30-32], few studies have discussed how to distinguish flame radiation from the incident radiation that is received by these hemispherical radiative measurement probes.

The purpose of this work is to develop a theoretical approach to quantify the contribution of wall emission and wall reflection to measured surface incident radiation, and to propose a simple and convenient approach to estimate the magnitude of flame radiation. CFD simulations of two different kinds of combustors are performed to validate the theoretical analysis and the approach for flame radiation estimation.

2. Theoretical analysis

A key assumption made in order to derive a simplified model of radiative heat transfer in a combustion chamber is that the medium is gray. This assumption implies that the radiative properties of the medium are independent of wavelength, which is a reasonable assumption for flames involving suspended particles, like coal-fired flames, highly luminous oil flames or some sooty gas-fired flames [33]. It could be less accurate in the case of non-sooty gas-fired flames, where non-gray gaseous molecules such as CO₂ and H₂O are the main contributors to radiation.

A simple axisymmetric combustor geometry as shown in Fig. 1 is used here to illustrate the theoretical model. More general geometries will be discussed later. It is assumed that the gas/particle medium in the combustor can be interpreted as a diffuse, gray surface that is close to the wall such that the areas of this fictitious surface and the wall are approximately equal. The dotted line in Fig. 2 represents the fictitious surface, which is defined to emit the same radiation power as the flame radiation, and has the same emissivity as that of the medium. The emissivity of the fictitious surface, ε_s , is equivalent to the emissivity of the medium, $\varepsilon_{m, eff}$. The flame radiation received by the wall (on a unit area basis) can be expressed as

$$E_{flame} = \varepsilon_{m, eff} \sigma T_{m, eff}^4, \quad (2)$$

where $T_{m, eff}$ is the effective temperature of the medium, and σ is the Stefan-Boltzmann constant ($5.67 \times 10^{-8} \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-4}$). Since the fictitious surface emits the same radiation power as the flame radiation, the temperature of the fictitious surface, T_s , should be equal to the effective temperature of the medium.

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161 For this analysis, the radiative properties of the wall are assumed to be diffuse and gray.

162 If the temperature and emissivity of the wall are T_w and ε_w , respectively, the net radiative

163 heat flux (W/m^2) transferred through the wall can be expressed by [34]

$$164 \quad q_{net} = \frac{\sigma(T_{m,eff}^4 - T_w^4)}{\frac{1}{\varepsilon_{m,eff}} + \frac{1}{\varepsilon_w} - 1}. \quad (3)$$

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166 According to Kirchhoff's law for diffuse, gray systems, the absorptivity of the wall is

167 equal to the emissivity of the wall, so the net radiative heat flux, q_{net} , is related to the

168 surface incident radiation on the wall, G_{total} , by the following:

$$169 \quad q_{net} = \varepsilon_w G_{total} - \varepsilon_w \sigma T_w^4. \quad (4)$$

170 By combining Eq. (3) and (4) and rearranging, the surface incident radiation can be

171 expressed as

$$172 \quad G_{total} = \frac{\sigma(T_{m,eff}^4 - T_w^4)}{\frac{\varepsilon_w}{\varepsilon_{m,eff}} + 1 - \varepsilon_w} + \sigma T_w^4. \quad (5)$$

173 As mentioned above, G_{total} is the value obtained by a radiometer and is the combination

174 of flame radiation, E_{flame} , and the contribution of the walls, G_{wall} . Therefore, G_{wall} can be

175 written as

$$176 \quad G_{wall} = G_{total} - E_{flame} = \frac{\sigma(T_{m,eff}^4 - T_w^4)}{\varepsilon_w \left(\frac{1}{\varepsilon_{m,eff}} - 1 \right) + 1} + \sigma T_w^4 - \varepsilon_{m,eff} \sigma T_{m,eff}^4. \quad (6)$$

177 The contribution of the reflection of flame emission on G_{wall} can be obtained by setting

178 the wall temperature T_w to 0 K, such that:

$$G_{wall, reflect} = \varepsilon_{m,eff} \sigma T_{m,eff}^4 \frac{(1 - \varepsilon_w)(1 - \varepsilon_{m,eff})}{\varepsilon_w + (1 - \varepsilon_w)\varepsilon_{m,eff}}. \quad (7)$$

The remaining part of G_{wall} is a combination of two factors: (1) wall emission that comes directly from the wall; and (2) wall emission that comes after being reflected off the wall at least once. Both factors are considered as the contribution of wall emission in this work, which can be expressed as

$$G_{wall, emission} = G_{wall} - G_{wall, reflect} = \varepsilon_w \sigma T_w^4 \frac{1 - \varepsilon_{m,eff}}{\varepsilon_w + (1 - \varepsilon_w)\varepsilon_{m,eff}}. \quad (8)$$

From Eq. (7) and Eq. (8), as the wall temperature, T_w , increases, $G_{wall, emission}$ increases, but $G_{wall, reflect}$ remains the same, indicating that G_{wall} increases with T_w . Equation (6) reveals

that G_{wall} decreases as the wall emissivity ε_w increases, because $\left(\frac{1}{\varepsilon_{m,eff}} - 1\right)$ is always

greater than 0. Based on these findings, it is evident that the influence of the wall (reflection and radiation) on the measurement of flame radiation can be minimized if the wall surface is cold and non-reflective (i.e., $\varepsilon_w = 1$). However, for most practical combustion applications, a low wall temperature and low wall reflectivity usually do not exist at the same time. Typically, there are two types of walls for combustion chambers: refractory-lined walls and water-cooled metal walls [35]. Although refractory-lined walls generally have high emissivity (i.e., low reflectivity), they also typically have large thermal resistance (i.e., low conductivity and large thickness), which means the fire-side surface of the wall becomes hot during operation. On the other hand, water-cooled walls usually have a relatively low surface temperature, but the water tube materials generally have lower emissivities than refractory materials. If there are ash/slag deposits on the

water tubes, the emissivity of surface of the wall can vary from 0.4 to 0.9 depending on the temperature and characteristics of the ash/slag deposits [36], but the surface temperature of the wall will also increase due to the increased thermal resistance caused by the deposits.

Table 1 quantifies the effect of wall conditions on the measurement of flame radiation. Nominal properties of a lab-scale air-fired flame combusted under atmospheric pressure are used, assuming $T_{m,eff} = 1500$ K and $\varepsilon_{m,eff} = 0.1$. Both a refractory-lined wall and a water-cooled wall are considered. The temperature and emissivity of these two types of walls are based on typical values reported in the literature [13].

Table 1 shows that, for the given flame and wall properties, the contribution of wall to the measured surface incident radiation can be up to 56% for the refractory-lined wall and 40% for the water-cooled wall. For the refractory-lined wall, the major contribution of the wall is from wall emission since the high wall temperature results in high radiation power. For the water-cooled wall, the contribution of wall emission is smaller, but the contribution of wall reflection is as high as 34%, equivalent to 56% of the flame radiation. Even if the emissivity of the water-cooled wall is increased to 0.8, the total contribution of the wall is still as high as 25%, equivalent to 33% of the flame radiation, and 67% of this contribution is from wall reflection. This example illustrates that the contribution of the wall to the measured surface incident radiation is significant for both refractory-lined walls and water-cooled walls. If the measured surface incident radiation is used as an

indicator of flame radiation, the radiative characteristics of the flame can be grossly underestimated.

3. An approach to estimate flame radiation

Since, when surface incident radiation is measured in practical combustion chambers the contribution of the wall can be substantial compared with the flame radiation, it is important to be able to deduce flame radiation from these measurements. Equation (5) indicates that G_{total} is a function of $T_{m,eff}$ and $\varepsilon_{m,eff}$ for given wall boundary conditions (i.e., T_w and ε_w). Theoretically, if the wall boundary conditions change slightly, G_{total} would change, but $T_{m,eff}$ and $\varepsilon_{m,eff}$ would remain the same. Then, with two sets of measurement data, the two unknowns, $T_{m,eff}$ and $\varepsilon_{m,eff}$, can be solved, and then the flame radiation can be calculated by Eq. (2). However, this approach requires the ability to manually vary the wall conditions and accurately determine the surface temperature and emissivity of the wall. While the former is possible for chambers with water-cooled walls, the latter is sometimes challenging because deposits can affect the surface properties of the wall.

In this section, an approach is proposed to estimate flame radiation without a prior knowledge of the surface conditions of the wall. This approach is based on measurements of the incident radiation and net radiative heat flux at the surface of the wall. The surface incident radiation can be measured directly with a radiometer, like the ellipsoidal radiometer used in [29]. The net radiative heat flux can be obtained by subtracting the convective heat flux from the total heat flux through the wall. Typically, the total heat

flux can be measured by a heat flux meter, like the disk-type heat flux meter used in [37]. The convective heat flux can be obtained by measuring or calculating the convective heat transfer coefficient [3]. In some circumstances where radiation dominates heat transfer, for example, in some oxygen-enriched solid-fuel combustors, convective heat flux may even be neglected, and the total heat flux through the wall can simply represent the net radiative heat flux.

Dividing Eq. (2) by Eq. (3) gives the ratio of the flame radiation to the net radiative heat flux:

$$\frac{E_{flame}}{q_{net}} = \frac{1 + \varepsilon_{m,eff} \left(\frac{1}{\varepsilon_w} - 1 \right)}{1 - \left(\frac{T_w}{T_{m,eff}} \right)^4}. \quad (9)$$

If $T_w^4 \ll T_{m,eff}^4$, Eq. (8) becomes

$$\frac{E_{flame}}{q_{net}} \approx 1 + \varepsilon_{m,eff} \left(\frac{1}{\varepsilon_w} - 1 \right) > 1. \quad (10)$$

Based on Eq. (10), E_{flame} is always larger than q_{net} , and the difference between E_{flame} and q_{net} decreases as $\varepsilon_{m,eff}$ decreases, given a constant wall emissivity. If $\varepsilon_{m,eff}$ is very small (e.g., the medium is optically thin), then

$$q_{net} \approx E_{flame}. \quad (11)$$

Hence, the net radiative heat flux can be used to approximate flame radiation if the wall surface temperature is relatively low compared with the medium temperature and the medium is optically thin. Optically thin systems include most lab-scale and some pilot-scale combustion facilities operating under atmospheric pressure. It follows from Eq. (10)

that the accuracy of this approximation decreases with increasing optical thickness of the medium and decreasing wall emissivity.

By dividing Eq. (2) by Eq. (5), the ratio of the flame radiation to the surface incident radiation can be obtained:

$$\frac{E_{flame}}{G_{total}} = \frac{\varepsilon_w + \varepsilon_{m,eff} (1 - \varepsilon_w)}{1 + \varepsilon_w \left(\frac{1}{\varepsilon_{m,eff}} - 1 \right) \left(\frac{T_w}{T_{m,eff}} \right)^4}. \quad (12)$$

If $T_w^4 \ll T_{m,eff}^4$, Eq. (12) becomes

$$\frac{E_{flame}}{G_{total}} \approx \varepsilon_w + \varepsilon_{m,eff} (1 - \varepsilon_w) < 1. \quad (13)$$

For those gas-fired systems where flame radiation mainly comes from soot emission, scattering can be neglected due to the small size of the gas molecules and soot particles. So if the medium is optically thick, the effective emissivity, $\varepsilon_{m,eff}$, would be close to 1.0. In coal-fired systems, char and ash particles can also be large contributors to radiation. Since the particles in pulverized coal combustion boilers are large compared with the wavelength of the radiation emitted at flame temperatures, the local particle absorption coefficient α_p and scattering coefficient σ_p can be respectively evaluated as [38, 39]:

$$\alpha_p = \sum_i \varepsilon_{p,i} n_i A_{p,i}, \quad (14)$$

$$\sigma_p = \sum_i (1 - f_{p,i}) (1 - \varepsilon_{p,i}) n_i A_{p,i}. \quad (15)$$

Here, i is the index number of a group of particles with the same size. Further, $\varepsilon_{p,i}$, n_i , $A_{p,i}$ and $f_{p,i}$ are particle emissivity, number density, projected area, and scattering factor of the i th group of particles, respectively. Particle emissivity, $\varepsilon_{p,i}$, in pulverized coal combustion is generally believed to vary between the emissivity of unburned coal (1.0) and that of fly

ash (0.6) [5]. The particle scattering factor, $f_{p,i}$, indicates the scattering properties of a particle; a value of 1.0 indicates total forward scattering (effectively treated as transmission). Generally, values from 0.6 to 0.9 were used for the particle scattering factor in coal combustion processes [5, 40]. Thus, the value of $(1-f_{p,i})(1-\varepsilon_{p,i})$ lies in the range of 0~0.016, which indicates that σ_p is much smaller than α_p , in coal combustion chambers. Therefore, the effective emissivity of the medium, $\varepsilon_{m,eff}$, can also be expected to be close to 1.0 when the medium is optically dense.

Based on Eq. (13), E_{flame} is always smaller than G_{total} , and the difference between them decreases as $\varepsilon_{m,eff}$ increases, given a constant wall emissivity. If $\varepsilon_{m,eff}$ is close to 1.0, then

$$G_{total} \approx E_{flame} , \quad (16)$$

which indicates that the contribution of the wall to the measurement of flame radiation becomes negligible, and the measured surface incident radiation can be directly used to approximate flame radiation. Optically dense systems include pilot- and large-scale coal combustion systems operating under elevated pressure [10, 23, 41]. Since the particle number density almost increases proportionally with pressure for a given thermal input and stoichiometric ratio, the optical thickness is also approximately proportional to pressure.

Eq. (10) and (13) shows that E_{flame} lies between q_{net} and G_{total} . As $\varepsilon_{m,eff}$ increases from 0 to 1, E_{flame} shifts from q_{net} to G_{total} . It should be noted that, for a combustor geometry other than the one shown in Fig. 1 (e.g. rectangular), the measured q_{net} and G_{total} could be non-uniform along the perimeter of the chamber. In this case, the average value of q_{net} and

G_{total} along the perimeter of the chamber should be used in the above approximation. Also, for chambers with hot walls at a steady state (i.e. refractory-lined walls), measurements of flame radiation can be conducted before a steady-state condition is achieved (i.e., during start-up stage) so that the “cold wall” condition can still be satisfied. Due to the heat capacity of the refractory walls, the change of the net radiative heat flux at the surface of the wall should be small in the short period during start-up. In reality, q_{net} should be measured as close to the inner surface of the wall as possible to minimize measurement error.

The above analysis provides a very convenient approach to estimate the flame radiation for optically thin and optically dense systems without prior knowledge of the surface properties of the wall. For systems with an optically thickness between optically thin and optically dense, more complicated approaches have to be used and the surface properties of the wall have to be known.

4. Validation by computational fluid dynamics (CFD) simulation

To validate the theoretical analysis and the approach to estimating flame radiation, CFD simulations were performed on two coal-fired combustors: (1) a 19.2 kWth, atmospheric pressure combustor, chosen to represent optically thin conditions; and (2) a 120 MWth, 50-atm pressurized combustor, chosen to represent optically dense conditions. The lab-scale atmospheric pressure combustor is a simple 2-meter cylinder with a diameter of 0.2 m, as depicted in Figure 3. The inlet of the combustor is split into two streams. A central high-speed air stream at 1500 K enters at 10 m/s. The surrounding air stream at 300 K

enters at 0.6 m/s. Coal particles enter the furnace near the center of the combustor with a mass flow rate of 0.0008 kg/s.

The large-scale pressurized combustor is a scale-up version of the lab-scale combustor, with a 10-meter length and a 1-meter diameter, and is operated at 50 atm. The residence time of this large-scale combustor is the same as that of the lab-scale combustor, meaning that the velocities of the air streams are five times larger than those of the lab-scale combustor (i.e. 50 m/s for the central stream and 3 m/s for the surrounding stream). To keep the stoichiometric ratio the same, the mass flow rate of the coal particles in the large-scale combustor becomes 5 kg/s, which is 6250 times as large as that in the lab-scale combustor. Since the gas volume flow rate is increased only by a factor of 125, the average particle number concentration of the large-scale combustor increases by a factor of 50 compared to that of the lab-scale combustor. Hence the average particle extinction coefficient also increases by a factor of 50. The optical thickness from the centerline to the wall, which equals the average particle extinction coefficient multiplied by the distance between the centerline and the wall, will increase by a factor 250 compared to that of the lab-scale combustor, leading to an optically dense system. The idea of utilizing pressurization to create optically dense media has also been found in [9, 10].

ANSYS FLUENT 13.0 was used for simulating the coal combustion processes in the two combustors. The flow field was modeled applying the Reynolds Averaged Navier Stokes (RANS) equations with the Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm to address the pressure-velocity coupling. The shear stress transport

(SST) $k - \omega$ model [42] and the Eddy Dissipation Model (EDM) [43, 44] were utilized for modeling of turbulence and turbulence-chemistry interactions. A simple single-rate devolatilization model and the kinetics/diffusion-limited char combustion model were employed since they are not the focus of this work. The radiative transport equation was solved by the Discrete Ordinates (DO) [45, 46] model, which can be used for both optically thin and optically dense media. Each octant of the entire angular space 4π at any spatial location was discretized into 5×5 solid angles (i.e. control angles), leading to a total of 200 directions. Further refined discretization did not show any noticeable improvement in terms of the distributions of temperature and incident radiation. Gaseous radiation and absorption was considered with the domain-based Weighted Sum of Gray Gases model (WSGGM). The combustor wall was assumed to be gray and diffuse. The particle emissivity and particle scattering factor were both set to be 0.9 [5, 47].

4.1. Lab-scale atmospheric pressure combustor

To demonstrate the effects of wall emission and wall reflection on the radiative heat transfer in the lab-scale combustor, four types of wall boundary conditions were employed in the simulation, as shown in Table 2. In Case 1, the wall temperature was set to room temperature 298 K and the wall emissivity was set to 1.0. Under these conditions, the incident radiation on the wall is equivalent to the flame radiation measured from a radiometer in an open atmospheric environment. This incident radiation should be very close to the actual flame radiation (i.e. $E_{flame} = \varepsilon_{m, eff} \sigma T_{m, eff}^4$).

Figure 4 shows the predicted temperature contours in the lab-scale combustor for different wall conditions. Even though the heat transfer rate to the wall is quite different in these cases, in the combustion zone (i.e., $x \approx 0-0.4$ m, where x is the distance from the inlet), the flame shapes and flame temperatures among the different cases are very close. Varied wall boundary conditions have little impact on the flame in this combustor.

Figure 5 shows the radial profiles of temperature and total extinction coefficient (i.e., gas extinction coefficient plus particle extinction coefficient) for all cases at $x = 0.4$ m. By integrating the extinction coefficient profiles along the radial distance, the optical thickness of the medium from the centerline to the wall (i.e., the area below the extinction coefficient profile) can be obtained. The calculated optical thickness is around 0.06. Based on Beer's law, this optical thickness corresponds to a transmissivity of 94%, which means that 94% of the radiation originating from the centerline can pass through the medium. Obviously, the medium in the combustor is optically thin.

The radiation from an optically thin medium mainly originates from local regions where the temperature and absorption coefficient are both high [10]. For the lab-scale combustor, the absorption coefficient is close to the extinction coefficient, as mentioned above. As shown in Figure 5, all cases have identical extinction coefficient profiles and nearly identical temperature profiles, except that the near-wall temperatures (between 0.04 and 0.05 m) are different due to different wall boundary conditions. While the temperature in this region varies between the cases, the emission in this region is negligible for all cases due to the low absorption coefficient. Thus, it can be reasonably

considered that the four cases have the same magnitudes of flame radiation. This assumption allows us to investigate the effects of wall emission and wall reflection with the same flame characteristics.

Table 3 shows the predicted surface incident radiation, G_{total} , and the net radiative wall heat flux, q_{net} , at $x = 0.4$ m on the wall from the CFD model. As mentioned above, for optically thin systems, if the wall temperature is relatively low, E_{flame} can be approximated by q_{net} . In Table 3, the measured G_{total} of Case 1 (i.e. cold and black wall) is expected to be close to the flame radiation. It can be found that the q_{net} of Case 3, where the wall temperature is relatively low, agrees reasonably well (around 13% error) with the G_{total} of Case 1, while the G_{total} of Case 3, and both the q_{net} and G_{total} of Case 2 and Case 4 differ significantly with the G_{total} of Case 1. Later we will show that, compared with the actual flame radiation, the net radiative heat flux in Case 3 differs even less.

It was mentioned earlier that, if the flame properties are not affected by the wall conditions, then with two sets of wall conditions and associated G_{total} or q_{net} , it is possible to solve for $T_{m,eff}$ and $\varepsilon_{m,eff}$. Substituting $T_w = 298$ K, $\varepsilon_w = 1.0$, $G_{total} = 6.196$ kW/m² and $T_w = 900$ K, $\varepsilon_w = 0.4$, $G_{total} = 42.363$ kW/m² into Eq. (5), and the effective temperature and emissivity of the medium can be solved for: $T_{m,eff} = 1025.3$ K, $\varepsilon_{m,eff} = 0.092$. It should be noted that using any two of the above cases gives very similar results. The flame radiation can be obtained based on these two parameters: $E_{flame} = \varepsilon_{m,eff} \sigma T_{m,eff}^4 = 5.791$ kW/m². Because this flame radiation is very small, even the wall emission at room

temperature can have a noticeable effect on the incident radiation. The G_{total} measured in Case 1 differs by 7% compared with this E_{flame} . However, the q_{net} in Case 1 only differs by 0.7%. In Case 3 ($T_w = 500$ K, $\varepsilon_w = 0.8$), the G_{total} differs by 77% compared with the E_{flame} and the q_{net} only differs by 7%.

More cases were simulated, in which the wall temperature was set to 500K and the wall emissivity was varied stepwise from 0.8 to 0.2. Table 4 shows the G_{total} , q_{net} , and the relative differences between the q_{net} and E_{flame} at $x = 0.4$ m. As wall emissivity decreases, G_{total} changes substantially, but the change of q_{net} is relatively small. The relative difference between q_{net} and E_{flame} increases as the wall emissivity decreases, in accordance with Eq. (10). As long as the wall emissivity is larger than 0.6, which is true for most combustion chambers, the relative difference remains lower than 10%. For comparison, Table 4 also provides the relative errors for the traditional approach of obtaining the flame radiation by measuring the difference of incident radiation with and without the flame [12]. In this approach, only the impact of wall emission has been considered, and the flame radiation is given by $(G_{total} - \sigma T_w^4)$. Clearly, for optically thin systems, the proposed approach in this study has a much better accuracy than the previous approach.

With the determined $T_{m, eff}$ and $\varepsilon_{m, eff}$, G_{total} and q_{net} for any wall conditions can also be calculated based on the theoretical model in Section 2. Figure 6 shows the G_{total} and q_{net} predicted by the theoretical model in Section 2 using $T_{m, eff} = 1025.3$ K and $\varepsilon_{m, eff} = 0.092$, together with the CFD simulation results. The results of the two models agree very well.

The flame radiation, E_{flame} , is the incident radiation (or net radiative heat flux) at $T_w = 0$ K, which is also showed in Fig. 6 together with $\pm 25\%$ boundary lines (dashed lines). Note that Fig. 6a and Fig. 6b have different vertical scales, the solid lines for q_{net} are much tighter to each other compared with those for G_{total} . It is found that the G_{total} always deviates from E_{flame} by more than 25% when the wall emissivity is lower than 0.7 or the wall temperature is higher than 420 K, while the q_{net} always deviates from E_{flame} by more than 25% when the wall emissivity is lower than 0.2 or the wall temperature is higher than 750 K.

4.2. Large-scale pressurized combustor

The same simulations as shown in Table 2 were also carried out for the pressurized combustor. Figure 7 shows the temperature contours in the pressurized combustor for all these cases. Similar to the lab-scale combustor, the overall flame shapes and temperature distributions are insensitive to the wall boundary conditions.

Figure 8 shows the radial profiles of temperature and total extinction coefficient at $x = 2$ m. Again, the extinction coefficients and temperatures for these cases are almost identical. There are differences only in the temperatures in the near-wall region. The calculated total optical thicknesses from the centerline to the wall based on the extinction coefficient profiles are as large as 6.2, which corresponds to a transmissivity of 0.002, indicating that the medium is optically dense. In an optically dense medium, radiation on the wall should be determined by local conditions [10]. However, in this case shown in Fig.8, even though the optical thickness from the centerline to the wall is large, the optical thickness

in the near-wall region is still very small, due to the low extinction coefficient near the wall. From $y = 0.2$ (y is the radial distance from the centerline) to the wall, the optical thickness is only 0.88, corresponding to a transmissivity of 0.41. Hence 41% of the incident radiation at $y = 0.2$ can still reach the wall. Therefore, it can be assumed that the temperature difference in the very near-wall region will not cause substantial change in the flame radiation, meaning the four cases still have the same magnitudes of flame radiation.

Table 5 shows the G_{total} and q_{net} for all cases at $x = 2$ m. Substituting $T_w = 298$ K, $\varepsilon_w = 1.0$, $G_{total} = 125.52$ kW/m² and $T_w = 900$ K, $\varepsilon_w = 0.4$, $G_{total} = 138.26$ kW/m² into Eq. (5), and the effective temperature and emissivity of the medium can be solved for: $T_{m,eff} = 1263.2$ K, $\varepsilon_{m,eff} = 0.869$. The E_{flame} obtained from these two parameters is 125.46 kW/m². The G_{total} in Case 3 ($T_w = 500$ K, $\varepsilon_w = 0.8$) differs only by 1.2% compared with the E_{flame} but the q_{net} differs by 21%, demonstrating that for an optically dense system, G_{total} becomes a better approximation for the flame radiation.

Cases with various wall emissivities but a constant wall temperature at 500 K are shown in Table 6. As opposed to the cases for the lab-scale combustor, this time q_{net} changes substantially with the wall emissivity but the changes in G_{total} is small. The relative difference between the G_{total} and E_{flame} increases with decreasing wall emissivity, following Eq. (12). The relative differences remain lower than 10% even when the wall emissivity is as low as 0.4. Unlike the cases for the lab-scale combustor, here the present estimation approach has a very similar accuracy compared with the traditional approach

which accounts for the impact of only wall emission. For relatively high wall emissivities, the present approach has a slightly better accuracy, while for relatively low wall emissivities, the traditional approach has a slightly better accuracy.

Figure 9 shows a comparison between the results of the theoretical radiation model using flame properties of $T_{m,eff} = 1263.2$ K and $\varepsilon_{m,eff} = 0.869$, and the results of the CFD model. As can be seen, they agree very well. In contrast with the lab-scale combustor, the solid lines for G_{total} are much tighter to each other compared with those for q_{net} . It is found that in the entire range of the wall temperature investigated, the G_{total} never deviates from the E_{flame} by more than 25%. The q_{net} deviates from E_{flame} by more than 25% when the wall emissivity is lower than 0.7 or the wall temperature is higher than 900 K.

5. Conclusions

In the present study, a theoretical analysis was conducted to evaluate the contribution of the wall to the radiation values measured by a radiometer in a closed combustion chamber. A simplified radiation model was developed by considering the non-homogeneous gas/particle medium inside the chamber as a grey surface with uniform properties which radiates with the same emissive power as the medium. Analyses show that surface incident radiation measurements can be used to represent flame radiation only when the wall is cold and black (i.e., emissivity equals 1). For a lab-scale atmospheric pressure combustor the contribution of the wall to the measured incident radiation can be as high as 56% for a typical refractory-lined wall, and as high as 40% for a typical water-cooled wall.

515

516 To distinguish the flame radiation from the measured incident radiation, an approach was
517 proposed to estimate the flame radiation for optically thin (most lab-scale and some pilot-
518 scale combustion facilities operating under atmospheric pressure) and optically dense
519 systems (pilot- and large-scale coal combustion systems operating under elevated
520 pressures). The main advantage of the approach is that it does not require prior
521 knowledge of the wall surface properties. The approach was based on measurements of
522 the net wall radiative heat flux and surface incident radiation on the wall. If the wall
523 temperature is much lower than the flame temperature, for optically thin systems, the net
524 radiative wall heat flux can be used to approximate the flame radiation; for optically
525 dense systems, the surface incident radiation can be used to approximate the flame
526 radiation. The analyses showed that the accuracy of the approximation approach
527 presented herein increases as the wall emissivity increases. CFD simulations were
528 performed for two combustors, one optically thin and one optically dense, to validate the
529 simplified radiation model and the flame radiation approximation approach. The CFD
530 model results agree well with the simplified flame radiation model results. For the
531 optically thin combustor, the error of the flame radiation approximation approach is less
532 than 10% when the wall emissivity value is greater than 0.6, which is true for most
533 practical combustion chambers. For the optically dense combustor, the error of the
534 approximation approach is less than 10% when the wall emissivity value is greater than
535 0.4.

536

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Figure captions

Figure 1. Illustration of the influence of walls on total incident radiation received by a radiometer. Flame is depicted by red zone.

Figure 2. Simplified model to describe radiative heat transfer inside a chamber.

Figure 3. Atmospheric pressure furnace with pulverized coal combustion.

Figure 4. Temperature contours (K) in the lab-scale atmospheric pressure combustor.

Figure 5. Predicted radial profiles of temperature (left) and total extinction coefficient (right) at $x = 0.4$ m in the lab-scale atmospheric pressure combustor.

Figure 6. (a) The incident radiation and (b) the net radiative heat flux predicted by the theoretical model (solid lines) and the CFD model (symbols) for the lab-scale atmospheric pressure coal combustor. The values in the parentheses indicate the wall temperature (the first value) and wall emissivity (the second value) in the CFD simulations.

Figure 7. Temperature contours (K) in the large-scale pressurized combustor for different cases.

Figure 8. Predicted radial profiles of temperature (left) and total extinction coefficient (right) at $x = 2$ m in the large-scale pressurized combustor.

Figure 9. Wall incident radiation (left) and net radiative heat flux (right) predicted by the simplified radiation model (lines) and the CFD model (symbols) for the pressurized combustor. The values in the parentheses indicate the wall temperature (the first value) and wall emissivity (the second value) in the CFD simulations.

Table 1. The effects of walls on measurements of flame radiation.

	Refractory-lined wall	Water-cooled wall
Wall conditions	$T_w=900\text{K}$ and $\varepsilon_w=0.9$	$T_w=500\text{K}$ and $\varepsilon_w=0.6$
E_{flame} (kW/m ²)	28.70	28.70
G_{total} (kW/m ²)	64.66	47.84
$G_{\text{wall}} / G_{\text{total}}$	55.6%	40%
$G_{\text{wall, emission}} / G_{\text{total}}$	51.2%	6.3%
$G_{\text{wall, reflect}} / G_{\text{total}}$	4.4%	33.7%
$(G_{\text{total}} - E_{\text{flame}}) / E_{\text{flame}}$	125%	67%

Table 2. Wall boundary conditions for different simulation cases

	Wall temperature (K)	Wall emissivity
Case 1	298	1.0
Case 2	900	0.8
Case 3	500	0.8
Case 4	900	0.4

Table 3. Surface incident radiation and net radiative heat flux at $x = 0.4$ m on the wall in the lab-scale combustor

	Case 1	Case 2	Case 3	Case 4
G_{total} (kW/m ²)	6.196	40.108	10.285	42.363
q_{net} (kW/m ²)	5.749	2.328	5.393	2.066

Table 4. Effects of the wall emissivity on the simulation results of the lab-scale combustor when the wall is at 500 K

Wall emissivity	0.8	0.6	0.4	0.2
G_{total} (kW/m ²)	10.285	12.235	15.674	23.232
q_{net} (kW/m ²)	5.393	5.215	4.852	3.938
$(q_{net} - E_{flame})/E_{flame}$ (%)	-6.9	-9.9	-16.2	-32
$(G_{total} - \sigma T_w^4 - E_{flame})/E_{flame}$ (%)	16.4	50.1	109.5	240.0

Table 5. Simulation results of the incident radiation and the net radiative heat flux at $x = 2$ m on the wall in the large-scale pressurized combustor

	Case 1	Case 2	Case 3	Case 4
G_{total} (kW/m ²)	125.52	131.53	126.99	138.26
q_{net} (kW/m ²)	125.07	75.67	98.76	40.42

Table 6. Effect of wall emissivity on the simulation results of the large-scale pressurized combustor when the wall temperature is 500 K.

Emissivity	0.8	0.6	0.4	0.2
q_{net} (kW/m ²)	98.76	77.33	52.71	27.51
G_{total} (kW/m ²)	126.99	132.42	135.32	141.10
$(G_{total} - E_{flame})/E_{flame}$ (%)	1.2	5.5	7.8	12.5
$(G_{total} - \sigma T_w^4 - E_{flame})/E_{flame}$ (%)	-1.6	2.7	5.0	9.6