Radiation Emission Characteristics of an Open-Cellular Porous Burner*

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Abstract

Radiation emission characteristics of an open-cellular porous burner, where methane-air premixed combustion occurs, were investigated experimentally and theoretically. In the analysis, we assumed that the chemical kinetics of gas-phase reactions are governed by a single-step Arrhenius rate expression. The energy liberation due to combustion and the effects of radiation were considered in the energy equations for the gas and solid phases. To evaluate the radiative transports in the solid-phase energy equation, the equation of transfer for the radiation field in a porous burner was solved using Barkstrom's finite difference method and the P_1 approximation. Three kinds of Ni-Cr open-cellular porous material with different porosities and pores per inch (PPI) were examined. Radiant output from the porous burner was measured based on a two-color radiometry. Calculated results of the forward radiative heat flux and the burner surface temperature were favorably compared with experimental data: satisfactory agreement between theory and experiment was obtained, and thereby the validity of the present theoretical model for predicting the radiation from a porous burner was confirmed. Moreover, it is found that there is only a little difference between predicted results of Barkstrom's method and these of the P_1 approximation.

Key words: Radiation Emission Characteristics, Porous Burner, Open-Cellular Porous Material, Premixed Combustion

1. Introduction

Porous radiant burners are widely used for many purposes including spacing heating, paper drying, paper finishing, baking and textile drying. There are two advantages in this type of a burner, i.e., higher thermal radiant output and minimal pollutant emission, which are the major reasons why a number of researchers and engineers have been paid attention to theoretical and experimental studies of porous burners and improvements of the burner performances⁽¹⁾. Echigo et al.⁽²⁾ first made a theoretical and experimental study of combustion in porous media: they revealed that significant combustion augmentation occurred in porous media due to energy recirculation by radiation and combustion from the combustion region to the unburned gas mixture one. Yoshizawa et al.⁽³⁾ presented more realistic models for combustion in porous media including multi-mode heat transfer, a single-step global mechanism of the chemical kinetics of gas phase reactions and radiative transfer in absorbing/emitting medium, and included that the reaction of combustion (flame location) occurred within a porous medium and depends strongly on the optical thickness. Hanamura et al.⁽⁴⁾ studied the dynamics of flame within a porous burner and showed that the flame stabilized in the region from the upstream end to about the middle of the porous

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burner and around or anchored at the surface one. They suggested that the flame stabilization mechanism related to three critical limits including of blow-off, extinction and flash back.

In according with the theoretical results of above-mentioned works may usually be classified porous burner into two categories on whether the flame can be anchored at the surface or be submerged in a porous medium. For a submerged flame burner, Sathe et al.⁽⁵⁾ performed a detailed numerical analysis by considering a single-step global mechanism. They suggested that the flame should be stabilized at the center of the porous medium to maximize the radiant output. An experimental study and a simplified model for radiant output from a surface porous burner were performed by Khanna et al.⁽⁶⁾. They presented that the radiant output and the pollutant emission increased with flame speed. Furthermore, an experimental study of radiation emission has been performed by Mital et al.⁽⁷⁾. They measured the radiation efficiency and pollutant emissions; the efficiency depends on an equivalence ratio, the rate of chemical energy input and *PPI* of the porous burner. For a surface flame type burner, Nakamura et al.⁽⁸⁾ performed an experiment and made a simplified theoretical analysis of the combustion on the surface of a porous ceramic plate. They reported that the surface temperatures increase with the optical thickness, but the radiant output was not quantified. Bouma et al.⁽⁹⁾ studied an experimental and numerical analysis of the radiation output by premixed methane-air combustion on a ceramic foam burner. They reported that the radiant efficiency decreased with increasing thermal load. Recently, Leonardi et al.⁽¹⁰⁾ developed a theoretical model to predict the thermal performance of a porous burner and showed that the calculated results for the burner surface temperature, the exit gas temperature and the radiation efficiency reasonably agreed with experimental data.

The purpose of the present study is to investigate the radiation characteristics and the surface temperature of an open-cellular porous burner. Theoretical analysis of one-dimensional, lean premixed methane-air combustion was performed without assuming a flame position. Combustion was modeled using a single-step Arrhenius rate expression. The energy equations for the gas and solid phases include a chemical reaction term and a radiation term. Barkstrom's finite difference method⁽¹¹⁾ and the P_1 approximation⁽¹²⁾ were used to solve the equation of transfer. Three kinds of Ni-Cr open-cellular porous material were examined and a two-color radiometry⁽¹³⁾ was used to measure the radiant energy from a porous burner. Comparison between the experimental results and the theoretical predictions were also addressed.

Nomenclatures

- A_d : surface area of radiation meter, m^2
- A_p : surface area of a porous burner, m^2
- C_f : isobaric specific heat capacity of a gas, $J/(kg \cdot K)$
- $C_{f\infty}$: isobaric specific heat capacity of a gas at an ambient temperature, $J/(kg \cdot K)$
- D_s : equivalent strut diameter, m
- Ea : dimensionless the activation energy ($=E/RT_\infty$)
- G : incident radiation, W/m^2
- H_i : dimensionless specific enthalpy of a i -th species ($=h_i/C_{f\infty}T_\infty$)
- $I(x, \mu)$: intensity of radiation, W/m^2
- I_{bu} : radiation intensity of the upstream region, W/m^2
- I_{bd} : intensity radiation from the downstream region, W/m^2
- k_f : thermal conductivity of a gas, $w/(m \cdot K)$
- $k_{f\infty}$: thermal conductivity of a gas at an ambient temperature, $w/(m \cdot K)$
- k_s : thermal conductivity of a porous, $w/(m \cdot K)$
- $p(\mu, \mu')$: scattering phase function
- PPI : pores per inch

- Pr_{∞} : Prandlt number ($=\mu_{f\infty}C_{f\infty}/k_{f\infty}$)
 q_{Rx} : the axial component of the radiative heat flux vector, W/m²
 q_R^+ : forward radiative heat flux, W/m²
 q_R^- : backward radiative heat flux, W/m²
 Re : Reynolds number ($=\rho_f u_f D_s/\mu_{f\infty}$)
 RR : dimensionless reaction rate ($=\sum H_i \nu_i RS_i \rho_f^* Pr_{\infty} \delta Y_{CH_4} Y_{O_2} \exp\{-Ea/\theta_f\}$)
 RS_i : dimensionless term in RR ($=M_i A \rho_{f\infty}^2 D_s^2 / M_{CH_4} M_{O_2} \mu_{f\infty}$)
 T_f : gas temperature, K
 T_{mix} : temperature of a gas mixture, K
 T_R : radiation temperature from upper surface of a porous burner, K
 T_s : temperature of a porous medium, K
 $T_{s,low}$: lower surface temperature, K
 $T_{s,up}$: upper surface temperature, K
 T_{∞} : ambient temperature, K
 u_f : gas velocity, m/s
 V_{mix} : mixture velocity, m/s
 $\dot{W}_i$: reaction rate of a i -th species
 x : coordinate in the flow direction
 x_0 : thickness of a porous plate, m
 Y_i : mass fraction of a i -th species

Greeks

- δ : dimensionless a ratio of the geometric length to equivalent strut diameter ($=x_0/D_s$)
 Φ : equivalent ratio
 ρ_f : density of a gas, kg/m³
 $\rho_{f\infty}$: density of a gas at an ambient temperature, kg/m³
 ρ_f^* : dimensionless density of a gas ($=\rho_f/\rho_{f\infty}$)
 $\mu_{f\infty}$: viscosity of a gas at an ambient temperature, Pa·s
 ν_i : stoichiometric coefficient of a i -th species
 π : a ratio of the circumference of a circle to its diameter
 σ : Stefan-Boltzmann's constant, W/(m²·K⁴)
 θ_f : dimensionless temperature of gas phase ($=T_f/T_{\infty}$)
 τ_0 : optical thickness ($=\beta^* x_0$)
 ξ : dimensionless coordinate in the flow direction ($=x/x_0$)
 Ψ^+ : dimensionless forward heat radiative heat flux

2. Theoretical analysis

The present model of a porous burner is shown in Figure 1. The following assumptions were introduced for the analysis as follows: 1) An open-cellular porous burner of thickness x_0 is placed horizontally and its both ends are uniformly irradiated by the blackbody radiation at an ambient temperature T_{∞} ($=298$ K); 2) The fed gases consist of methane and air: the burned gases consist of CO₂, H₂O, O₂ and N₂. These gases are assumed to be non-radiating and are ideal gases; 3) The porous medium is gray and is capable of emitting, absorbing and anisotropically scattering thermal radiation; 4) The porous medium is non-catalytic; 5) The physical properties depend on temperature; 6) The combustion reaction is modeled by using a single-step global reaction; 7) The combustion occurs steadily and the flame is one-dimensional.

Under these assumptions, the flame can be analyzed theoretically using a standard laminar flame theory. The governing equations may be written as follows:

Fig.1 The theoretical model of a porous burner

The continuity equation is given by

$$\frac{\partial(\rho_f u_f)}{\partial x} = 0. \quad (1)$$

For the upstream region $(-\infty < x < 0)$ and downstream region $(x_0 < x < \infty)$, the species conservation equations and the gas energy equation, respectively, are written as

$$\rho_f u_f \frac{\partial Y_i}{\partial x} + \frac{\partial}{\partial x} (\rho_f Y_i V_i) = \dot{W}_i M_i, \quad i = 1, 2, \dots, N-1, \quad (2)$$

$$\rho_f u_f C_f \frac{\partial T_f}{\partial x} + \sum_{i=1}^N \rho_f Y_i V_i C_{f,i} \frac{\partial T_f}{\partial x} = \frac{\partial}{\partial x} \left(k_f \frac{\partial T_f}{\partial x} \right) - \sum_{i=1}^N h_i \dot{W}_i M_i, \quad (3)$$

where, V_i is the diffusion velocity, N is a total number of the species considered; M_i , $C_{f,i}$ and h_i are the molecular weight, the isobaric specific heat capacity and the specific enthalpy of a species i , respectively. The mass of the N_2 species (inert nitrogen) is estimated from $\sum Y_i = 1$.

For the porous medium region $(x < x < x_0)$, the species conservation equations and the energy equations for both gas and solid phases are given as follows:

$$\rho_f u_f \frac{\partial Y_i}{\partial x} + \phi \frac{\partial}{\partial x} (\rho_f Y_i V_i) = \phi \dot{W}_i M_i, \quad i = 1, 2, \dots, N-1, \quad (4)$$

$$\rho_f u_f C_f \frac{\partial T_f}{\partial x} + \phi \sum_{i=1}^N \rho_f Y_i V_i C_{f,i} \frac{\partial T_f}{\partial x} = \phi \frac{\partial}{\partial x} \left(k_f \frac{\partial T_f}{\partial x} \right) - h_v (T_f - T_s) - \phi \sum_{i=1}^N h_i \dot{W}_i M_i, \quad (5)$$

$$f(1-\phi) \frac{\partial}{\partial x} \left(k_s \frac{\partial T_s}{\partial x} \right) + h_v (T_f - T_s) - \frac{\partial q_{Rx}}{\partial x} = 0, \quad (6)$$

$$\frac{\partial q_{Rx}}{\partial x} = 4\beta^* (1-\omega^*) \left(\sigma T_s^4 - \frac{G}{4} \right). \quad (7)$$

Here ϕ is the porosity, h_v is the volumetric heat transfer coefficient between the gas and solid phases, β^* is the scaled extinction coefficient and ω^* is the scaled albedo. Modeling of these physical properties was summarized in Ref. (14).

The boundary conditions for Eqs.(2), (3) and (6) are given as follows:

$$\left. \begin{aligned} x = -\infty : & \quad T_f = T_{mix}, \quad Y_{CH_4} = Y_{CH_4,in}, \quad Y_{O_2} = Y_{O_2,in}, \\ x = 0 : & \quad \frac{\partial T_s}{\partial x} = 0, \\ x = x_0 : & \quad \frac{\partial T_s}{\partial x} = 0, \\ x = \infty : & \quad \frac{\partial T_f}{\partial x} = 0, \quad \frac{\partial Y_i}{\partial x} = 0. \end{aligned} \right\} \quad (8)$$

The quantity G represents the incident radiation and q_{Rx} denotes as the net radiative heat flux in the flow direction:

$$G = 2\pi \int_{-1}^1 I(x, \mu) d\mu, \quad (9)$$

$$q_{Rx} = 2\pi \int_0^1 I(x, \mu) \mu d\mu. \quad (10)$$

These quantities can be determined from the equation of transfer. Once the radiation field is specified, the quantities G and q_{Rx} can be readily evaluated. The equation of transfer and the associated boundary conditions are given by

$$\mu \frac{\partial I(x, \mu)}{\partial x} + \beta^* I(x, \mu) = \beta^* (1 - \omega^*) \frac{\sigma T_s^4(x)}{\pi} + \frac{\beta^* \omega^*}{2} \int_{-1}^1 I(x, \mu) p(\mu, \mu') d\mu', \quad (11)$$

$$\left. \begin{aligned} x = 0 : \quad I(0, \mu > 0) &= I_{bu} = \frac{\sigma T_\infty^4}{\pi}, \\ x = x_0 : \quad I(x_0, -\mu) &= I_{bd} = \frac{\sigma T_\infty^4}{\pi}. \end{aligned} \right\} \quad (12)$$

Eq.(11) was solved using Barkstrom's method and the P_I equations.

For Barkstorm's method, the radiation intensity $I(x, \mu)$ should be separated into two components including of a forward $I^+(x, \mu > 0)$ and backward radiation intensity $I^-(x, \mu < 0)$, and discrete points of the cosine of scattering angle μ is 10 points, the components were rearranged following as:

$$\mu \frac{d\tilde{H}(x, \mu)}{dx} + \tau_0 \tilde{J}(x, \mu) = \beta^* (1 - \omega^*) \frac{\sigma T_s^4(x)}{\pi} + \frac{\beta^* \omega^*}{2} \int_0^1 P(\mu, \mu') \tilde{J}(x, \mu') d\mu', \quad (13)$$

$$\mu \frac{d\tilde{J}(x, \mu)}{d\xi} + \tau_0 \tilde{H}(x, \mu) = \frac{\beta^* \omega^*}{2} \int_0^1 Q(\mu, \mu') \tilde{H}(x, \mu') d\mu', \quad (14)$$

$$\tilde{J}(x, \mu) = \frac{1}{2} [I^+(x, \mu) + I^-(x, -\mu)], \quad (15)$$

$$\tilde{H}(x, \mu) = \frac{1}{2} [I^+(x, \mu) - I^-(x, -\mu)], \quad (16)$$

where, $P(\mu, \mu')$ and $Q(\mu, \mu')$ denote the metrics of the scattering phase function as can be expressed by

$$P(\mu, \mu') = p(\mu, \mu') + p(\mu, -\mu'), \quad (17)$$

$$Q(\mu, \mu') = p(\mu, \mu') - p(\mu, -\mu'). \quad (18)$$

Furthermore, boundary conditions for solving Barkstrom's method are given by

$$\left. \begin{aligned} x = 0 : \quad \tilde{J}(0, \mu) + \tilde{H}(0, \mu) &= \frac{\sigma T_\infty^4}{\pi}, \\ x = x_0 : \quad \tilde{J}(x_0, \mu) - \tilde{H}(x_0, \mu) &= \frac{\sigma T_\infty^4}{\pi}. \end{aligned} \right\} \quad (19)$$

Thus, both quantities G and q_{Rx} can be evaluated from the following expression:

$$G = 4 \int_0^1 \tilde{J}(x, \mu') d\mu', \quad (20)$$

$$q_{Rx} = 4 \int_0^1 \tilde{H}(x, \mu') \mu' d\mu'. \quad (21)$$

On the other hand, the P_I approximation is the first order spherical harmonic method for solving the equation of transfer, and in this approximation only G and q_{Rx} appear. The P_I equations are written as:

$$\frac{dq_{Rx}}{dx} + (1 - \omega^*)\beta^*(G - 4\sigma T_s^4) = 0, \quad (22)$$

$$\frac{dG}{dx} + 3(1 - \omega^*\tilde{g}^*)\beta^*q_{Rx} = 0. \quad (23)$$

The boundary conditions for Eqs.(20) and (21) are given by Marshak's ones:

$$\left. \begin{aligned} x = 0: \quad G + 2q_{Rx} &= 4\sigma T_\infty^4, \\ x = x_0: \quad G - 2q_{Rx} &= 4\sigma T_\infty^4. \end{aligned} \right\} \quad (24)$$

The forward radiative heat flux on the upper surface of a porous burner can be evaluated from the following expression:

$$q_R^+(x_0) = \frac{I}{4}[G(x_0) + 2q_{Rx}(x_0)]. \quad (25)$$

Regarding to the species conservation equations and the gas energy equation, the reaction rate of a i -th species $\dot{W}_i$ is governed by a single-step Arrhenius rate expression:

$$\dot{W}_i = \nu_i \dot{W} = \nu_i A [CH_4] [O_2] \exp(-E/RT), \quad (26)$$

where ν_i is the stoichiometric coefficient of species i . $[CH_4]$ and $[O_2]$ are, respectively, the concentrations of fuel and oxygen (kmol/m^3). A single-step kinetic reaction rate adopted in the present study has been proved to be adequate for predicting in Ref. (15). The frequency factor A is $1.75 \times 10^{11} \text{ m}^3/\text{kmol-s}$ while the activation energy E is $1.4 \times 10^6 \text{ kJ/kmol}$: these were taken from Ref. (16) after slight modification. The original values of A and E are, respectively, $6.25 \times 10^{10} \text{ m}^3/\text{kmol-s}$ and $1.4 \times 10^6 \text{ kJ/kmol}$.

Figure 2 shows, to investigate burning velocities in the porous media, the four predictions based on the present kinetic parameters, a single-step global kinetic model ⁽¹⁶⁾, full kinetics mechanism ⁽¹⁶⁾ and a multi-step kinetics mechanism ⁽¹⁵⁾, and compared to available experimental data ⁽¹⁷⁾. It is obvious that none of these model satisfactory predict the experimentally determined the burning velocity. This discrepancy may be due to the limited knowledge of the thermal, radiative and fluid mechanism within porous materials as well as the uncertainties of solid properties. For some important parameters such as volumetric heat transfer coefficient, the effective thermal conductivity of the solid k_s and the radiative properties (e.g., extinction coefficient, albedo and scattering phase function etc.), it is difficult to obtain true values. Therefore, experimental property data are necessary, but the methods to probe the realistic properties are also difficult. In the present calculations, the above-mentioned important parameters have used from the experiment as discussed thoroughly in Ref. (14). In fact, when using data of experimental properties in the predicted model of the porous burners, it can be expected that significant errors may be introduced as summarized in Ref. (18). These uncertainty data lead to the discrepancy in the prediction of burning velocity, but the tendencies of predictions

Fig.2 Comparison of the predictions and the experimental data of the burning velocity

are qualitatively consistent with the experimental data. Agreement between the predictions based on the present single-step global reaction and the experimental data was better than the predictions obtained by Ref. (16), particularly, in the range of an equivalence ratio Φ less than about 0.8. In addition, Fig.2 also shows the good comparison of the laminar flame speed between the present kinetic model and the experimental data⁽¹⁹⁾: it is indicated that the present single-step global kinetic model is adequate for predicting the combustion phenomena in porous burner, at least for predicting the burning velocity.

Moreover, the mass burning velocity of fuel m_f is evaluated by integrating the net mass reaction rate.

$$m_f = \rho_f u_f = \rho_{f\infty} S_L = \int_{-\infty}^{\infty} \dot{W}_i M_i dx. \quad (27)$$

The equation of state is given by

$$\rho_f = \frac{\bar{M}P}{RT}, \quad (28)$$

where P is the absolute pressure, $\bar{M}$ is the mean molecular weight and R is the universal gas constant.

For convenience of calculations, the governing equations and associated boundary equations are transformed into dimensionless forms, and then the dimensionless equations are solved numerically using an implicit difference method. Both the upstream and downstream regions and the porous region are, respectively, divided into 100 and 200 equally spaced increments for calculations of T_f and Y_i . For calculations of T_s , the porous region is divided into 200 equally spaced increments, whereas the optical thickness is divided into 400 equally spaced increments for solving the equation of transfer and the P_i equations. To obtain the solutions of T_f , T_s , Y_i , G or q_{Rx} are first determined based on an assumed temperature profile, which is appropriate for occurring combustion, and then the quantities of G and q_R are obtained by solving equation of transfer⁽¹¹⁾ or the P_i equations at staggered lattice points⁽¹²⁾. The m_f is also calculated from Y_i . Once G and Y_i are obtained, the finite difference equations for T_f and T_s can be solved readily by Gaussian elimination. Thereafter, the derived solutions of T_f and T_s are, respectively, substituted into the species conservation equations and the equation of transfer or the P_i equations to get new solutions of these quantities; similar calculations are repeated until the following convergence criterion is satisfied: $|(Q^{(n)} - Q^{(n-1)})/Q^{(n)}| < 10^{-5}$. Here, Q represents T_f , T_s , Y_i , G or q_{Rx} .

3. Experiment

A schematic diagram of the present experimental apparatus is shown in Fig. 3. The combustion system consists of four parts: a conical flow distribution section, a plenum chamber, a porous burner and an exhaust duct. The conical section is made of stainless steel and is 120 mm long. The plenum chamber consists of a circular stainless steel pipe of 109 mm inner diameter and 100 mm long, and the porous burner is horizontally installed on the top end of this chamber. The exhaust duct is made of a tin pipe of 130 mm in diameter and 150 mm long. The methane-air premixed gas was fed from the bottom, and then was flowed upward through a porous plug

Fig.3 Schematic diagram of the experimental apparatus

and packed spheres, which were equipped to prevent the backward flame propagation under flash back conditions. The porous plate was horizontally settled between a supported pipe and a covering ring of 104 mm inner diameter. To measure temperatures of the upper and lower surfaces ($T_{s,up}$ and $T_{s,low}$) of the porous plate, three type-R thermocouples of 0.3 mm in diameter, which were made of Platinum/Rhodium 13%, were adhered to each surface and coated with silica to prevent catalytic reaction on them. Mass flow rates of air and methane were measured using calibrated volumetric flow meters. For a fixed mixture velocity, the equivalence ratio Φ was varied by simultaneously changing both air and fuel flow rates. However, the experiments were stopped for equivalence ration of over 0.8 although the combustion is still occurred. The mixture velocity V_{mix} was varied from 5.89×10^{-2} to 19.62×10^{-2} m/s. The range of stable combustion was governed by two combustion limits as explained later. Three kinds of Ni-Cr open-cellular porous material were examined and their physical characteristics were summarized in Table 1.

Table1 Physical characteristics of the examined open-cellular porous materials

Materials	Porosity, ϕ	PPI	x_0 (m)	τ_0	β^* (m^{-1})
Ni-Cr #1	0.920	8.5	0.0103	1.213	117.8
Ni-Cr #3	0.930	21.5	0.0103	2.828	274.6
Ni-Cr #5	0.899	39.5	0.0103	6.523	633.3

Fig.4 An arrangement of a radiation meter inside the experiment apparatus

Fig.5 A detail of radiation meter

Figure 4 shows an arrangement of radiation meter ⁽¹³⁾ within the experimental. Two kinds of radiation meters with different surface hemispherical emissivities were used: one surface is made of black-painted SUS304 foil of ϵ_H , while another surface was composed of $600^\circ\text{C} \times 8\text{ hr}$ oxidized Inconel 601 thin foil of ϵ_L . The ϵ_H and ϵ_L were, respectively, determined as a function of experimental surface temperature $T_{surface}$:

$$\epsilon_H = 0.8663 + 0.4505\theta^2 - 0.493\theta^3 + 0.175\theta^4, \quad (29)$$

$$\epsilon_L = 0.4697 + 0.1351\theta^2 - 0.1235\theta^3 - 0.1501\theta^4, \quad (30)$$

where θ is $T_{surface}/1000\text{ K}$.

Detail of the radiation meter is shown in Fig. 5. Outer tube of the radiation meter was made of a SUS304 pipe of 13.5 mm outer diameter, 1.5 mm thickness and 25 mm height. Ceramic tube of $\text{Al}_2\text{O}_3\text{-SiO}_2$ (HB) of 10 mm outer diameter and 18 mm height was inserted into the SUS304 pipe and fixed using sauerisen cement. A black-painted SUS304 foil of 10 mm diameter and 0.6 mm thickness or $600^\circ\text{C} \times 8\text{ hr}$ oxidized Inconel 601 thin foil of 10 mm diameter and 0.2 mm thickness was placed on the top end of the ceramic tube. A type K thermocouple of 0.5 mm diameter was welded on the back surface. Moreover, a MgO cement was pound to form a 6 mm height layer to reduce heat loss. On the back surface of a MgO layer a type-K thermocouple of 0.5 mm diameter was also anchored. By considering the energy balance of radiation and convection on the two kinds of radiation meter and eliminating the effect of composition from the two energy balance equation, we can obtain the incident radiation on the surface radiation meter G_{Exp} :

$$G_{Exp} = \sigma T_R^4 = \frac{\varepsilon_H \sigma T_{WH}^4 (T_g - T_{WL}) - \varepsilon_L \sigma T_{WL}^4 (T_g - T_{WH})}{\varepsilon_H (T_g - T_{WL}) - \varepsilon_L (T_g - T_{WH})} \quad (31)$$

T_R and T_g denote a radiation temperature and a combustion of gas temperature. T_{WH} and T_{WL} are, respectively, the surface temperatures of the black-painted and the Inconel 601 thin foil.

In evaluating the forward radiative heat flux on the upper surface of a porous burner, the measured radiation temperature must be corrected by a configuration relation between the radiation meter and the porous plate as follows:

$$q_{Rx}^+ = G_{Exp} \left/ \left[1 - \frac{I}{1 + (r_p/H)^2} \right] \right. \quad (32)$$

Here, r_p is the radius of a porous burner (m) and H is the distance between the surface of a radiation meter and the porous burner (m).

4. Results and discussion

The stabilized combustion region of the present porous burner is shown in Fig.6. The blow-off limits, where the flame lift out from the upper surface of a porous plate and becomes extinguished, are in the range of Φ from 0.48 to 0.51 in all examined porous plates, though the lean limit of methane-air premixed flame in the free space is 0.53. The flash back limit (lifts down from the lower surface and extinguishment of the flame) is approximately 0.54 for the Ni-Cr #1 porous burner. The flash back limits of other porous plates are not presented here because their limits are greater than 0.8, in which it is the maximal condition for operating in the present experiment. From the flash back conditions, it can be discussed that the prominent physical properties of the open-cellular porous materials⁽¹⁴⁾, particularly optical thickness τ_0 (extinction coefficient β^*) for the Ni-Cr #3 and #5 porous materials are higher than the Ni-Cr #1 by three and six times respectively, sufficiently contributed the radiative transfer within porous burner that is the reason why the combustion for those burner still stabilized in spite of the equivalence ratios are over than 0.8. Agreement between the predicted results and the experimental ones is satisfactory.

Figure 7 illustrates the theoretical results of the temperature profiles of the gas and the solid phases and the dimensionless reaction rate RR for different kinds of porous plate. Here, RR is defined by $\sum H_i v_i R S_i \rho_f^{*2} PR_{\infty} \delta Y_{CH_4} Y_{O_2} \exp(-Ea/\theta_f)$ and represented a reaction zone of the combustion. In Fig.7, we assumed that the mixture velocity is 9.75×10^{-2} m/s and $\Phi = 0.5$. The peak temperatures of gases and RR move toward inside a porous medium and increase with increasing τ_0 : this suggests that the optical thickness of a porous burner increases, the radiative heat propagation from the combustion region into the regions upstream and downstream may be reduced, and thus the flame is stabilized inside a porous burner. Moreover, it is found that the temperature of solid phases rise in accord with the temperature of gas phases, but, in the case of a Ni-Cr#1 porous burner, the peak solid temperature is realized on the back surface of a burner. The predictions based on Barkstrom's method agree satisfactory with the P_I approximation

Figure 8 shows typical temperature profiles of the gas phases and the solid phases and the dimensionless reaction rate RR within a Ni-Cr#5 porous burner of $\Phi = 0.5$. Here, only theoretical results based on exact, Barkstorm's method, are shown because the prediction based on the P_I approximation agree sufficiently with the exact results and this case evens for other porous burner. The measured mean surface temperatures are also indicated by the symbols. The peak locations of the gas phase temperatures and RR (flame location) occur within a porous medium and shift toward the downstream region and RR increases as Re increases as long as Re is less than 1.24. However, when Re is greater than 1.42, the peak locations of T_f and RR occur outside a porous burner. This result suggests that the flame

location can be fire freely within or on the surface of a porous burner by appropriate adjustment of Re . Regarding to Ref. (4), the flame stability in the present prediction can be located in the half downstream region of the porous burners. To explain this contrast, Hanamura et al. (4) assumed the high volumetric heat transfer coefficient between the gas and solid phase in the model conducted to a large number of the energy liberation due to combustion can be converted as the radiant energy emitted toward preheat zone, and then the enormous energy loss simultaneously release to post-flame zone. To balance the total energy, the flame is only stabilized in about first half region of the porous burners. For the present model, the volumetric heat transfer coefficient obtained by experiment of Ref. (14), i.e., $Nu = h_v D_s^2 / k_f = 0.124 (Re Pr)^{0.791}$, were used, which it is much less than from Ref. (4). Therefore, some combustion gas enthalpy is converted as the emitted radiation: thus no high heat loss is occurred in almost all position of the burners, and then the total energy balance is proper for the flame stability in the region from about the middle to around upper surface of the porous burners. In addition, the porosity ϕ of open-cellular porous materials, which is free space within the porous burner, is higher than that of the solid particles or packed bed spheres in Ref.(4), leading to that our model is easy to achieve the combustion reaction. Agreement between the theoretical results of the solid phase temperature and the measured upper and lower surface temperatures are satisfactory.

Figures 9 (a) and (b) illustrate variations in the surface temperatures and Ψ^* against Re at $\Phi = 0.5$. As seen from Fig.9 (a), $T_{s,low}$ slowly decreases with an increase in Re , whereas $T_{s,up}$ increases first and then decreases as Re increases. Figure 9 (b) depicts that Ψ^* increases rapidly and then levels off with Re . Moreover, the maximum values of Ψ^* are reached at approximately 80 irrespective of kinds of examined porous medium. Agreement between the predictions and experiments is satisfactory, particularly, as for the porous plates with larger PPI . Agreement between the predictions based on Barkstrom's method and the P_i equations is good.

Figures 10 (a) and (b) illustrate effect of Φ on relations between the surface temperatures and Ψ^* of the N-Cr#5 burner and Re . In accord with the results observed in Fig.10 (a), $T_{s,up}$ increases slightly with Re , whereas $T_{s,low}$ decreases slightly with increasing Re . For a fixed value of Re , $T_{s,up}$ increases with Φ . Figure 10 (b) depicts that Ψ^* increases

Fig.6 Comparison of the predictions with the experimental results of the stability limits

Fig.7 Predicted profiles of the gas and solid phase temperatures, and dimensionless reaction rate at $\Phi = 0.5$ and $V_{mix} = 9.75 \times 10^{-2}$

Fig.8 Profiles of the gas and solid phase temperatures and the dimensionless reaction rate RR (The symbols in the figure indicate the present experimental data)

rapidly with Re and levels off around $Re = 1$. Agreement between the two predictions and the experiments is again satisfactory. Moreover, it is found that discrepancy between numerical results based on Barkstrom's method and those based on P_l method is indiscernible.

Figs.9 Comparisons of the predicted surface temperatures (a) and the dimensionless forward radiative heat flux Ψ^+ (b) with the corresponding experimental data (case of $\Phi = 0.5$)

Figs.10 Comparisons of the predicted surface temperatures (a) and the dimensionless forward radiative heat flux Ψ^+ (b) with the corresponding experimental data (case of the Ni-Cr#5 burner)

5. Conclusions

The major conclusions that can be drawn from the present study are summarized as follows:

- 1) The blow-off limits are in the range of Φ from 0.48 to 0.51 irrespective of kinds of porous burner, whereas the flash back limit is 0.54 as for the Ni-Cr#1 burner and is greater than 0.8 for other burners. Thus, the flammability range of the Ni-Cr#1 burner is the narrowest among the porous burners examined.
- 2) The flame shifts from the inside toward the upper surface of a porous burner as Re increases.
- 3) The upper and lower surface temperatures ($T_{s,up}$ and $T_{s,low}$) and the dimensionless forward radiative heat flux Ψ^+ depend on the equivalence ratio Φ , Reynolds number Re and kinds of porous material (PPI , ϕ and τ_0), but both Re and Φ are the major factors.
- 4) The present theoretical model yields satisfactory results in predicting the lean premixed methane-air combustion phenomena and the radiant output from the porous burner.
- 5) Agreement between the predicted results based on Barkstrom's finite difference method and theory based on the P_l equations is acceptable, particularly, for the porous plates with a thicker τ_0 .

From the present results, the radiant out put for maximizing should be operated at higher Re and the value of τ_0 should be large enough as long as the combustion occurs under blow off flammability limit; the higher equivalence ratio Φ should also be operated.

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