Heat Transfer Characteristics of Porous Radiant Burners Using Discrete-Ordinate Method (S_2-Approximation)

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ABSTRACT

This paper describes a theoretical study to investigate the heat transfer characteristics of porous radiant burners. A one dimensional model is used to solve the governing equations for porous medium and gas flow before the premixed flame to the exhaust gas. Combustion in the porous medium is modeled as a spatially dependent heat generation zone. The homogeneous porous media, in addition to its convective heat exchange with the gas, may absorb, emit and scatter thermal radiation. The radiation effect in the gas flow is neglected but the conductive heat transfer is considered. In order to analyze the thermal characteristics of porous burners, the coupled energy equations for the gas and porous medium based on the discrete ordinate method are solved numerically and the effect of various parameters on the performance of porous radiant burners are examined. Comparison between the present results with those obtained by other investigators shows a good agreement.

KEYWORDS

porous radiant burners, discrete ordinate method, radiative transfer, thermal characteristics

1. INTRODUCTION

In recent years porous radiant burners (PRBs) are widely used specially in high temperature systems. These types of burners have various advantages over the conventional burners. They have higher burning rates, increased flame stability and lower combustion zone temperature which led to a reduction in NO_x formation and low emissions of CO. Porous burners can provide high radiant flux with low pollutant emissions for many industrial processes such as drying and curing.

Flames in porous media have higher burning velocities and leaner flammability limits than open flames. These effects are the consequence of phenomena named "excess enthalpy combustion "[1], where the heat that is generated in the combustion zone is transferred by radiation and conduction through the solid matrix to the unburned gases. As the radiative heat transfer process has a main role in RPBs, it is very important to have a good radiation model for analyzing the characteristics of these burners. Although the PRBs technology is rather new, many literatures exist on the subject. Echigo et al. [2] investigated the use of porous media for combustion augmentation, by considering a heat generation zone inside a porous medium. It was found that the energy recirculation by radiation preheated the combustion mixture, resulting in significant combustion augmentation. Wang and Tien [3] extended Echigo's analysis by including the effect of radiation scattering in the analysis. Transient heat transfer characteristics of an energy recovery system using porous media was investigated by Gandjalikhan Nassab [4] in which the two–flux radiation model was used for calculating the radiative fluxes. Echigo et al. [5] extended their earlier work to investigate the use of porous media for combustion augmentation, by considering a heat generation zone inside a porous medium. It was found that the energy recirculation by radiation preheated the combustion mixture, resulting in significant combustion augmentation. Tong and Sathe [6] analyzed the porous radiant burners considering a one–dimensional conduction, convection and radiation model with combustion treated as a spatially dependent heat generation zone. For radiative part, they used spherical harmonic method. Numerical results show that the

properties. Tong and Sathe [7] extended their earlier work by considering the actual combustion instead of

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constant heat generation zone. They found that for maximizing radiant output, the optical depth should be large and the flame should be stabilized near the center of porous medium.

Brenner et al. [8] used a two–dimensional pseudo homogeneous heat transfer and flow model for porous burners with an objective to optimize the combustion process and to adapt the burner design. They considered conservation equations for 20 species, two momentum equations and one energy equation. They assumed local thermal equilibrium between the gas and the solid phases. They did not consider any radiation model, such that the radiative part was taken care by the experimental value of the effective thermal conductivity. The effective thermal conductivity accounted for the effects of conduction, radiation and the convective dispersion.

For calculating the radiative fluxes in PRBs, the physical model is complicated by the directional characteristics of scattering, and exact analytical expressions for radiative heat flux and solid matrix temperature, like those for non scattering cases, cannot be achieved. The first 2–D numerical analysis of PRBs with detailed radiation model for scattering case was done by Talukdar et al. [9]. Both transient and steady state characteristics were studied. The combustion was considered as a spatially dependent heat generation zone. The energy equations for gas and solid phase were solved numerically such that the radiative part was found using the collapsed dimension method [10]. It must be noted that, this method needs a very complicated computations for determining the radiative flux distribution in an emitting absorbing and scattering media. Malico and Pereira [11] considered a 2-D cylindrical radiation field to investigate the influence of radiative properties on the performance of porous burner. They found that the temperature profiles were very sensitive to a perturbation in the radiative coefficient, particularly when the scattering albedo was increased. Mishra et al. [12] investigated a 2-D rectangular porous burner. Separate energy equations for gas and solid phases were solved for methane-air combustion. The radiative part of the energy equation is modeled using the collapsed dimension method. They studied the effects of power density, equivalence ratio, extinction coefficient and volumetric heat transfer coefficient on temperature and concentration profiles.

The analysis here employs, discrete ordinate method for computing radiative heat flux in PRBs. This method remains simple in expressions but retains important physical insights. To the best of authors' knowledge, the discrete ordinates method has not been used in thermal analysis of porous burners by other investigators. Besides, an attempt is made to obtain the radiative output from a porous burner depends on optical thermal characteristics of PRB,s running under severe operating conditions Therefore, the present work mainly aims at the heat transfer analysis of a one–dimensional cylindrical porous burner with the detailed radiation model based on discrete ordinate method. As a result of high porosity of porous medium, thermal conductivity in this medium is low, and thus it is assumed that only convection and radiation take place in the porous matrix. However, in the gas flow, because of non-radiative gas assumption, heat transfer occurs by conduction and convection. In the present work, the combustion process is considered as a spatially dependent heat generation zone.

The governing equations consist of two energy equations for gas and solid phases and one radiative equation based on discrete ordinate method are solved numerically. For validation, the numerical results are compared with those obtained by other investigators and good agreement is found.

2. FORMULATION

The problem under consideration is shown schematically in Fig. 1. A gaseous fuel–air mixture enters the duct at $x = -x_i$. The porous segment acts as PRB is located between $0 < x < x_3$ when $x_3 << R_0$ to insure the validity of the one–dimensional analysis.

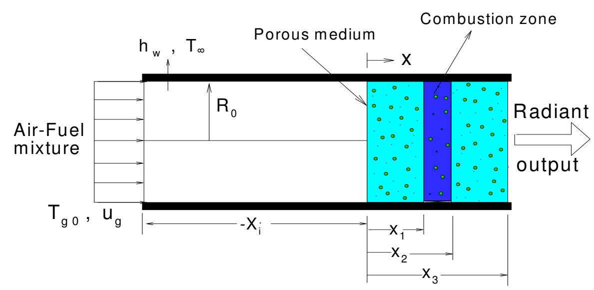


Figure 1: Schematic diagram of the porous radiant burner.

Slug flow with velocity u_g is assumed in the duct. Gaseous radiation is neglected compared to solid radiation. Thus, radiation is considered only between the particles that comprise the porous layer. The heat generation zone, representing the flame, is situated in the porous medium at $x_1 < x < x_2$. Since the solid and gas are not in local thermal equilibrium, separate energy equations are needed to describe energy transfer in these two phases. The governing equations inside the porous region which are the energy equations for the fluid flow and porous segment and also the radiation equation (based on discrete ordination method) can be written as follows:

$$u_g\rho_g c_g \frac{dT_g}{dx}(\pi R_0^{\ 2})+h_w(2\pi R_0)(T_g-T_\infty)+h_sN_sA_s(\pi R_0^{\ 2})(Tg-T_P)-$$
$$\frac{\partial}{\partial x}(K_g\frac{\partial T}{\partial x})(\pi R_0^{\ 2})-\dot{Q}\delta(x)(\pi R_0^{\ 2})=0 \quad (1)$$

$$\frac{dq}{dx}+h_sN_sA_s(T_P-T_g)=0 \quad (2)$$

In the gas flow out of the porous media, the conductive heat transfer can be ignored because of small gas thermal conductivity along with the existence of small temperature gradient [3]. Thereby, the gas energy equation outside the porous layer can be obtained from Eq. 1 by omitting the convective term between gas and solid phases and also the conductive term as follows:

$$u_g\rho_g c_g \frac{dT_g}{dx}+\frac{2h_w}{R_0}(T_g-T_\infty)=0 \quad (3)$$

In the above equations, h_s is the heat transfer coefficient between the fluid and the solid matrix, N_s is the number density of the particles, A_s is the surface area of an individual particle, σ_a and σ_s represent the absorbing coefficient, the scattering coefficient respectively. The term $\dot{Q}\delta(x)$ is the unit heat generation and $\delta(x)$ is the delta function defined as unity for $x_1 \le x \le x_2$ and zero elsewhere.

A. *Discrete ordinate method*

The discrete ordinate method is a tool to transform the equation of radiative transfer (for a gray medium, or on a spectral basis) into a set of simultaneous partial differential equations. The discrete ordinate method is based on a discrete representation of the radiation intensity in specified directions. A solution to the transport problem is found by solving the equation of transfer for a set of discrete directions spanning the total differencing of the directional dependence of the equation of transfer. Integrals over solid angle are approximated by numerical quadrature. Most of the credit, in our opinion, for the introduction and development of the discrete-ordinate method should go to Chandrasekhar (1950), who in his fundamental work on radiative transfer did much to define the method as an effective computational tool.

The rediative transfer equation for absorbing, emitting, and anisotropically scattering medium is as follows [13]:

$$\frac{dI}{ds}=\hat{s}.\nabla I(r\hat{s})=\sigma_a(r)I_b(r)-\sigma_e I(r,\hat{s})+\frac{\sigma_s(r)}{4\pi}\int_{4\pi} I(r,\hat{s}')\phi(r,\hat{s}',s')d\Omega' \quad (4)$$

and is subjected to the following boundary condition

$$I(r_w,\hat{s})=\varepsilon(r_w)I_b(r_w)+\frac{\rho(r_w)}{\pi}\int_{\hat{n}.\hat{s}'<0} I(r_w,\hat{s}')\left|\hat{n}.\hat{s}\right|d\Omega' \quad (5)$$

where we have limited ourselves to an enclosure with opaque, diffusely emitting and reflecting walls.

In the discrete ordinates method, equation (4) is solved for a set of *m* different direction and the integrals over direction are replaced by numerical quadratures approximation:

$$\int_{4\pi} f(\hat{s})d\Omega \cong \sum_{i=1}^{m} w_i f(\hat{s}_i) \quad (6)$$

where the w_i are the quadrature weights associated with the directions s_i . Equation (4) is approximated by a set of *m* differential equations as follows:

$$\hat{s}_i\nabla I(r\hat{s}_i)=\sigma_a(r)I_b(r)-\sigma_e(r,\hat{s}_i)+\frac{\sigma_s(r)}{4\pi}\sum_{j=I} w_j I(r,\hat{s}_j)\phi(r,\hat{s}',\hat{s}_j)\ i=1,2,...m \quad (7)$$

with the following boundary conditions:

$$I(r_w,\hat{s}_i)=\varepsilon(r_w)I_b(r_w)+\frac{\rho(r_w)}{\pi}\sum_{\hat{n}.\hat{s}'<0}^{m} I(r_w,\hat{s}_j)\left|\hat{n}.\hat{s}_j\right| \quad \hat{n}.\hat{s}_j>0 \quad (8)$$

Once the intensities have been determined in the desired direction, the radiative heat flux inside medium at any surface may be found from:

$$q(r)=\int_{4\pi} I(r,\hat{s})\hat{s}d\Omega \cong \sum_{i=1}^{m} w_i I_i(r)\hat{s}_i \quad (9)$$

There are two direction cosine (±μ) in x-axis for one dimensional radiation transport equation. In the present work, S_2 approximation is employed, because it is so match with one – dimensional system.

The radiative transfer equation in one–dimensional form becomes as follows:

$$\mu^m\frac{\partial I_j^m}{\partial X}=\sigma_a\ I_{b,j}-\sigma_e\ I_j^m+\frac{\sigma_s}{4\pi}\sum_{m'} w^{m'} I_j^{m'} \quad (10)$$

By differencing of equation (10), the following relation can be obtained for computation of radiative intensity:

$$I_j^m=\frac{X^m u_0(X^m)I_{j-1}^m-X^m u_0(-X^m)I_{j+1}^m+KI_{b,j}+s_j^m}{\sigma_e+X^m sign(X^m)} \quad (11)$$

where

$$u_0(x)=\begin{cases}1 & x>0\\ 0 & x<0\end{cases} \quad , \quad \text{sign}(x)=\begin{cases}1 & x>0\\ -1 & x<0\end{cases}$$

$$X^m = \frac{\mu^m}{\Delta X} \quad , s_j^m = \frac{\sigma_s}{4\pi} \sum_{m'} w^{m'} I_j^{m'}$$

Once the radiation intensity is computed by iterative scheme, the radiative heat flux can be calculated as:

$$q_j = \sum_m w^m \mu^m I_j^m \tag{12}$$

and for one–dimensional condition , it became [14]:

$$q_j^+ = w^{(1)} \mu^{(1)} I_j^{(1)} \tag{13}$$

$$q_j^- = w^{(2)} \mu^{(2)} I_j^{(2)} \tag{14}$$

where $\mu^{(1)} = \mu^{(2)} = 0.5$.

For parametric studies, the governing equations are non–dimensionalized by introducing the following dimensionless parameters:

$$P_w = \frac{h_w}{u_g \rho_g c_g} \qquad Q = \frac{q}{\sigma T_{g0}^4} \qquad R = \frac{h_s N_s A_s}{\sigma T_{g0}^3} R_0$$

$$P_1 = \frac{u_g \rho_g c_g T_{g_0}}{x_3 \dot{Q}} \qquad Pe = \frac{u_g \rho_g c_g R_0}{k_g}$$

$$\Gamma = \frac{h_s N_s A_s R_0}{u_g P_g C_g} \qquad X = \frac{x}{R_0}$$

$$\omega = \frac{\sigma_s}{\sigma_e} \qquad \bar{I} = \frac{I}{\sigma T g_0^4}$$

In the above, Pe is the Peclet number, R is the rate of solid–fluid convective heat transfer to radiative heat transfer, P_w and Γ represent the ratios of heat loss through the duct wall to fluid energy and solid–fluid convective heat transfer to fluid energy, respectively. Using these dimensionless parameters, the non–dimensional forms of the governing equations can be written as follows:

$$\frac{d\theta_g}{dX} + 2p_w\left(\theta_g - \theta_\infty\right) + \Gamma\left(\theta_g - \theta_p\right) - \frac{\delta(x)}{p_1}\frac{R_0}{\delta} - \frac{1}{Pe}\frac{d^2\theta_g}{dX^2} = 0 \tag{15}$$

$$\frac{dQ}{dX} = R(\theta_g - \theta_p) \tag{16}$$

$$Q = \sum_m \omega^m \bar{I}^m \mu^m = Q^+ - Q^- \tag{17}$$

Equations (15) to (17) should be solved by prescribing appropriate boundary conditions**.**

B. Boundary conditions

At the inlet of the duct ($x = -x_i$), the gas is at ambient temperature T_∞ and at the outlet section ($x = x_3$), the gas temperature gradient is known from Eq. 3. Since, there is not any radiative sources outside the porous burner, the radiative flux q at $x = 0$ and at $x = x_3$ are equated to zero. Thereby, the boundary conditions for the gas and solid energy equations and for radiative equations can be written as follows:

$$\theta_g = \theta_\infty \quad \text{at} \quad x = -x_i$$

$$\frac{d\theta_g}{dx} = -2\, p_w(\theta_g - \theta_\infty) \quad \text{at} \quad x = x_3$$

$$Q^+(0) = 0$$

$$Q^-(x_3) = 0$$

3. SOLUTION TECHNIQUE

In thermal analysis of PRBs, the coupled equations (15) to (17) should be solved simultaneously to obtain the values of dependent variables θ_g , θ_p , and Q at each nodal point in the computational domain. Finite difference form of the gas energy equation is obtained using central differencing for derivative terms where the error of discretization is the order of $(\Delta X)^2$. Energy equation for solid phase is an algebraic one and the radiative equation which is written based on discrete ordinate method can be solved using Eq. 11. For obtaining the grid independent numerical results, a uniform grid of 60 to 100 nodal points in the computational domain based on the optical thickness of porous layers are used.

The sequence of calculations can be stated as follows:

1- A first approximation for each dependent variables θ_g , θ_p is assumed.

2- The finite difference form of radiative transfer equation is solved for obtaining the values of I, Q and $\frac{dQ}{dx}$ at each nodal point.

3- Using the values of $\frac{dQ}{dx}$ which were obtained in step 2.

4- The solid energy equation is solved to calculate the porous temperature θ_p.

5- Gas energy equation is solved to determine the temperature of gas phase.

6- Steps 2 to 4 are repeated until convergence is achieved.

4. VALIDATION OF COMPUTATIONAL RESULTS

In order to validate the computational results, a test case was analyzed and the numerical results were compared with the theoretical data in Ref. [6]. In the studies by Tong and Sathe [6], the dimensional and non–dimensional parameters of PRB system as a test case are given in Tables.1 and 2, respectively.

TABLE 1
Dimensional parameters of the test case

Parameter	Units	values from Tong and Sathe [6]
ρ_g	kg / m^3	0.19
u_g	m / s	2.8
c_g	KJ / kgK	1.28
T_∞	K	298
T_{g_0}	K	298
$\dot{Q}$	W / m^3	1.5×10^9
x_3	m	0.01
$h_s N_s A_s$	$W / m^3 K$	2×10^7
k_g	$W / m°C$	0.125
$x_2 - x_1$	m	0.001

TABLE 2
Non–dimensional parameters of the test case

Parameter	values from Tong and Sathe [6]
τ_0	1
Pe	272
R	66.6×10^3
P_1	0.0136
p_w	0
Γ	1468

Fig. 2 shows the variation of gas temperature along the porous burner. It is seen that the incoming air–fuel mixture is preheated by radiation in the region upstream to the combustion zone. The maximum temperature occurs inside the heat generation domain after which the gas temperature decreases by converting gas enthalpy to thermal radiation. However, the agreement between the present results with those obtained in Ref. [6] is satisfactory with small inconsistency which is due to employing different methods in solving radiative transfer equation. Thus, it can be concluded that the assumptions and numerical scheme used in the present analysis are valid although the radiation model is much more simpler than the other models.

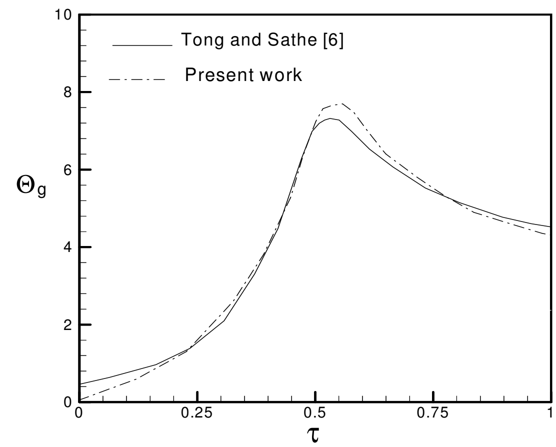


Figure 2: Gas temperature distribution along the Burner.

δ / R_0 =0.2, τ_0 =1, Γ=1468, Pe=272

$p_W = 0, \xi_{f\ell} = 0.5, \theta_\infty = 1, P_1 = 0.0136$

In order to compare the present numerical results with experiment, a heat recovery system which was analyzed theoretically by Ben Kheder et al. [15] and experimentally by Olade [16] is considered as a test case whose dimensional parameters are given in Table 3. It should be noted that in this heat recovery system, a large amount of incident thermal radiation $B_1 = 6.5\times10^6\ W / m^2$ is applied toward a porous layer in which a gas flow is passing through. The incident radiation is absorbed by the porous layer and the gas flow is heated by the convection heat transfer with the porous matrix. The gas and porous temperature distributions are presented in Fig.3. As this figure shows, the porous temperature at inlet section increases sharply as a result of absorbing the incoming radiation and the low temperature gas flow is effectively heated by convective heat transfer between gas and solid phases. However, Fig.3 shows a good consistency between the present results and those obtained theoretically and experimentally in Refs. [15] and [16], respectively.

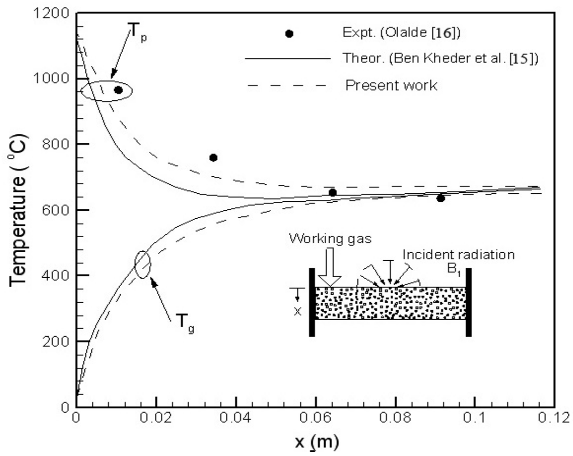


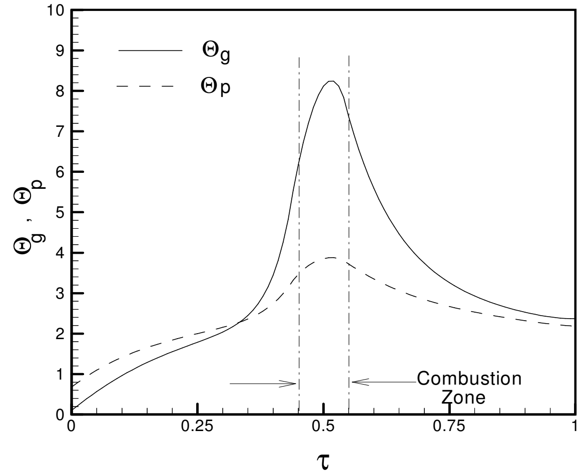
Figure 3: Gas and porous temperature distributions along the porous layer.

TABLE 3
Dimensional parameters of the test case from Ben Kheder et al. [15] and Olalde [16]

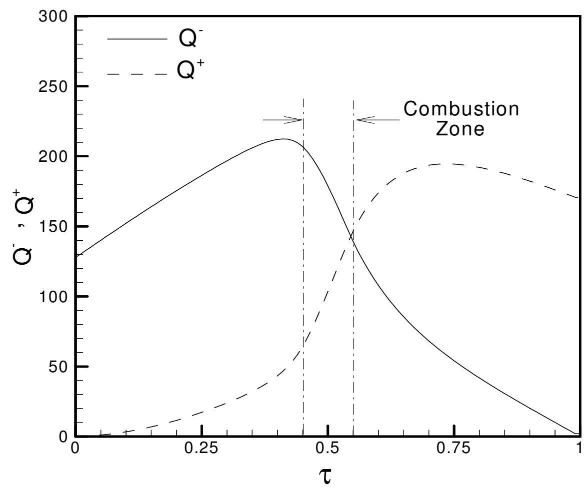
parameter	Unit	value
x_3	m	0.117
u_g	m/s	0.194
ρ_g	kg/m^3	1
c_g	$J/kg\ ^oC$	1000
σ_a	m^{-1}	8.547
σ_s	m^{-1}	0
k_g	W/m^oC	1.43
k_p	W/m^oC	358
φ	-	0.43

5. RESULTS AND DISCUSSION

In order to show the thermal behavior of PRBs, the gas and porous temperature profiles θ_g , θ_p and also Q^+, Q^- along the burner are shown in Fig. 4. The heat generation zone position represented by $\xi_{f\ell} = (x_1 + x_2)/2x_3$ is equal to 0.5, that is, the flame was regarded as suited in the middle of porous segment. The gas and porous temperatures increase along the flow direction at the entrance of the burner, such that maximum values of these variables occur in the combustion zone. Fig . (4-b) shows that the maximum values of Q^+ and Q^- occur outside the combustion zone.



(a): gas and porous temperature Distributions



(b): radiative heat flux distributions

Figure 4: Temperature and radiative heat flux distributions in the burner.

δ/R_0=0.1, Γ=200, Pe=153, θ_∞=1

$\omega = 0$, $\xi_{f\ell} = 0.5$, $P_1 = 0.006$, $P_w = 0.1$

Fig. 5 shows the effect of the optical thickness τ_0 on the gas temperature distribution inside the porous medium. In the computations, different optical thicknesses are generated by changing the extinction coefficient of porous media. So, for porous segments with small optical thickness, the main mechanism for heat transfer is convection in comparison to radiation. It is seen that the peak temperature decreases by increasing the optical thickness.

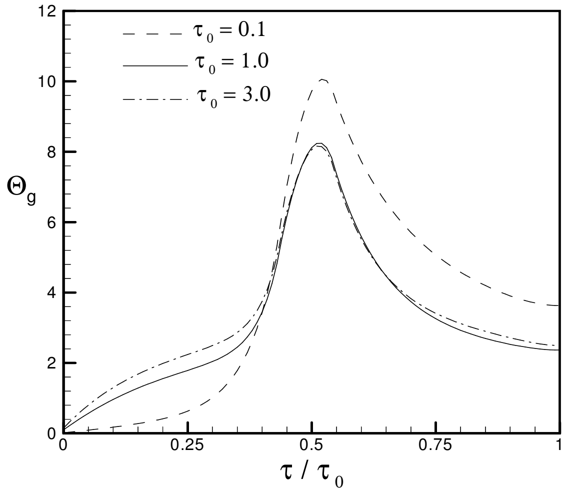


Figure 5: Effect of the optical thickness on the gas Temperature.

δ / R_0=0.1, Γ=200, Pe=153, θ_∞=1

$$\omega = 0 \, , \xi_{f\ell} = 0.5 \, , P_1 = 0.006 \, , P_w = 0.1$$

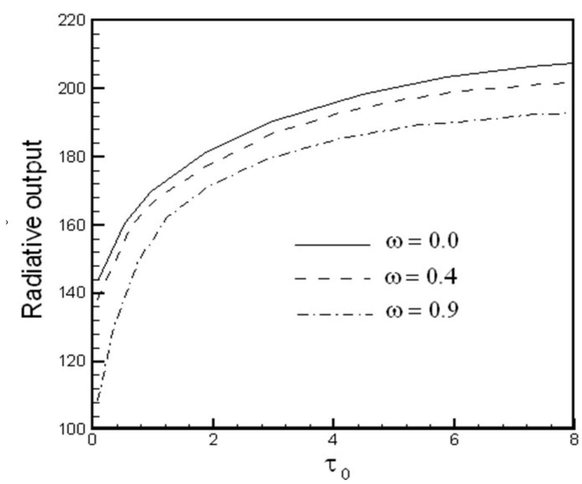


Figure 6: Effect of the optical thickness and the scattering albedo on the radiative output.

δ / R_0=0.1 ,Γ=200, Pe=153, θ_∞=1

$$\xi_{f\ell} = 0.5 \;\; P_1 = 0.006 \, , P_w = 0.1$$

The effect of optical thickness τ_0 on radiative output $Q^+(\tau_0)$ is shown in Fig. 6. Also, Fig. 6 shows the effect of radiative scattering on the radiant output. It is seen that a lower albedo results in a higher radiant output from the porous burner. The influence of radiative scattering to the thermal behavior of PRBs can further be explained in Fig. 7. It is seen that a lower albedo results in a smaller gas peak temperature due to stronger emissions by the solid phase.

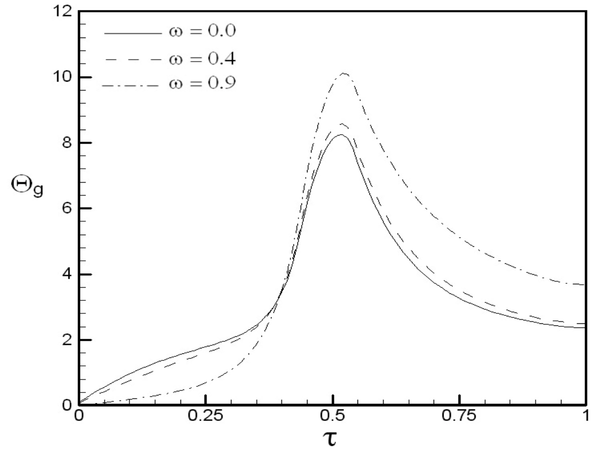
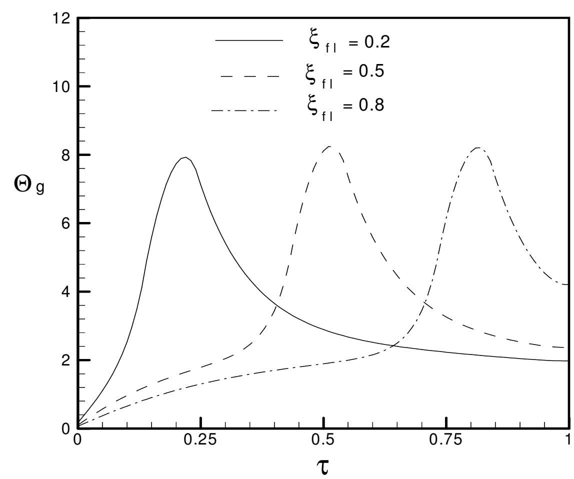


Figure 7: Gas temperature distributions at three different values of the scattering albedo.

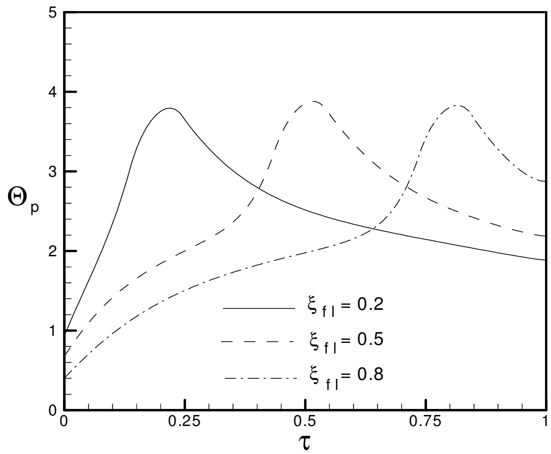
δ / R_0=0.1,Γ=200,Pe=153

$$\theta_\infty = 1 \, , \; \xi_{f\ell} = 0.5 \, , \; P_1 = 0.006 \, , P_w = 0.1$$

One of important factors in the performance of PRBs is the flame location. To study the effect of this parameter, the gas and porous temperature distributions along the burner, for three different flame locations are shown in Figs. 8. For all cases, the maximum values of gas and porous temperatures remain almost constant and occur inside the combustion zone. The influence of flame location to the thermal behavior of PRBs can further be explained by plotting the variation of radiant output as a function of ξ_{fl} in Fig. 9. The optimal value of ξ_{fl} depends on radiative properties of porous burner and in general, this position is more toward the downstream direction of the porous burner.



(a): gas temperature



(b): porous temperature

Figure 8: Effect of flame location on gas and porous temperature distributions.

δ / R_0=0.1,Γ=200,Pe=153

$\theta_\infty = 1$, $\omega = 0$, $P_1 = 0.006$, $P_w = 0.1$

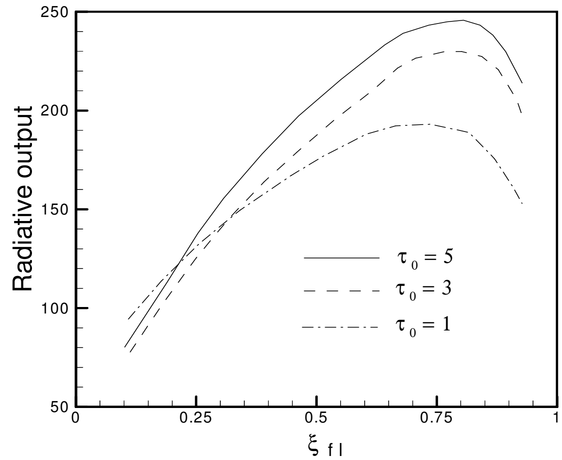


Figure 9: Radiative output for different flame locations.

δ / R_0=0.1, Γ=200, Pe=153, θ_∞=1

$\omega = 0$, $P_1 = 0.006$, $P_w = 0.1$

6. CONCLUDING REMARKS

A numerical study has been made on the porous radiant burners to obtain the thermal characteristics of these systems. The solid matrix may absorb, emit and scatter thermal radiation and the discrete ordinate method was used to describe the radiative flux. Combustion was modeled by considering a uniform heat generation zone inside the porous medium. Results for the temperature and radiative heat flux distributions in the burner were obtained for different parameters like the optical thickness and scattering coefficient. The numerical results compare well with theoretical results of other investigators, in spite of differences between the radiation models.

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