Numerical simulation of porous radiant burners under transient condition

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Abstract

The purpose of this work is to analyze two dimensional rectangular porous radiant burners for investigating the thermal characteristics of this type of burners in starting time period. Since, the solid and gas phases are not in thermal equilibrium, two separate energy equations for these two phases are solved numerically with alternative direction implicit scheme. The gas is considered non-radiative medium and for computation of radiative heat flux through the solid phase, the radiative transfer equation (RTE) is employed and solved with the discrete ordinates method (DOM). It is obtained that due to the dominant radiation effects, the required time to reach the steady gas temperature is very low. Furthermore, the effects of optical thickness and scattering albedo on the performance of porous radiant burner (PRB) are investigated.

Keywords: Porous radiant burner, transient condition, discrete ordinate method

1. Introduction

Nowadays, using devices that utilize more efficient combustion technologies is of particular interest to industries. Using porous materials in the structure of burners makes better air-fuel mixing and eliminates hot spots which leads to more efficient combustion with lower pollutants [1]. Combustion in porous media has further advantages compared to conventional combustion devices, such as high radiant output, high flame speed, high power density and high combustion efficiency.

Porous radiant burners (PRBs) through conduction and radiation heat transfer in solid phase can recirculate heat from combustion zone toward the entrance section where air-fuel mixture enters to the burner. In the flame upstream, solid to gas convection heat transfer preheats the reactants which results in increasing flame speeds and achieving higher local combustion temperatures without getting high temperature at the burner exit which reduces NOx formation.

In the last decades, many numerical studies and experimental investigations have been done on the combustion of porous media [1-6]. In order to optimize the combustion process, a 2-D pseudo homogeneous porous burner was studied by Brenner et al. [7] in 2000. They did not consider any radiation model, such that the radiative part has been affected by the experimental value of the effective thermal conductivity.

Talukdar et al. [8] examined a two dimensional porous radiant burner with the detailed radiation model. Both transient and steady state characteristics were studied, but in the transient condition, only the results of increasing time on the convective flux, radiative flux, gas conductive flux and solid conductive flux were presented. They did not show the temperature distributions of gas and solid phases at different time steps. In addition, the transient behavior from the start of the system operation up to the steady state was not represented. A one-dimensional porous burner in the transient state was analyzed by Lari and Gandjalikhan Nassab [9]. In their work, the coupled energy equations for the gas and the porous medium and also the radiative transfer equation based on the two-flux method were solved numerically. The effect of some parameters on the performance of porous radiant burner were investigated. Farzaneh et al. [10] simulated a two-dimensional porous burner in steady condition. They solved numerically Navier-Stokes, the energy and the chemical species transport equations. The sensitivity of flame front and heat transfer characteristics to the some factors were assessed. They concluded that any perturbation in heat transfer properties influences the temperature difference between the gas and solid phases and consequently the volumetric heat transfer between two phases.

Keshtkar and GandjalikhanNassab [11] studied a two- dimensional rectangular porous radiant burner in steady condition. The gas phase was considered transparent to radiation, but the porous medium was considered to absorb, emit and isotropically scatter radiation. They conclude that layers with large optical thickness and lower scattering albedo have high values of radiant output. In a study done by second and third authors [12], fluid flow in a 2-D porous heat recovery system was simulated using lattice Boltzmann method and thermal analysis was performed by solving governing equations of solid and gas phases. In this study, momentum and heat transfer equations were solved separately and no temperature variation was taken into account in fluid flow simulations. Also, no considerable effect was observed on simulations due to considering real fluid flow in contrast with plug flow.

A useful method that is used to predict the thermal behavior of the porous radiant burners is asymptotic analysis under simplifying assumptions. This analysis provides a practical tool for qualitative prediction of porous burner characteristics. This model successfully predicts the operational features of porous burners such as flame location and radiant efficiency. Chandra and Nakamura [13] considered one dimensional mathematical model for combustion of solid fuel over the porous medium. A two-temperature model with two energy equations for the gas and porous medium was employed to determine the effects of the thermal non-equilibrium between two phases. They used large activation energy asymptotic to solve the problem. They conclude that an increase of porosity of the porous medium, thermal conductivity of porous medium, and gas flow velocity, increases the moving the speed of the burning front. They showed that the asymptotic simulation gives a good agreement with the numerical simulation.

In the previous literature, only a few studies have focused on the transient behavior of porous radiant burners and most of them used a simple two-flux radiation model in radiation computations. In the present work, transient characteristics of a two-dimensional rectangular porous burner are numerically studied. The solid and gas phases are considered to be in non-local thermal equilibrium. Three modes of heat transfer take place simultaneously in the solid phase, while in the gas phase only conduction and convection are considered and the gas is considered to be transparent to radiation. Plug flow assumption is considered. The governing equations which are two energy equations for the gas and solid phases and also the radiative energy equation for the solid phase are solved numerically and simultaneously to determine the thermal behavior of the system. Due to that the porous matrix is assumed to be an emitting, absorbing, and scattering medium, a good radiation model is necessary. To this end, discrete ordinates method (DOM) is used to solve radiative transfer equation in order to obtain heat flux distribution through solid matrix. Moreover, for the purpose of validation, the computational results are compared with available data obtained by other researchers and good consistency exists.

2. Porous Burner Configuration

The geometry of the problem under investigation is schematically shown in Figure 1. The computational domain has*2cm*length and *1cm*height. The dimension of porous medium in the direction normal to the plane of paper (z- direction) is very large compared to other dimensions (x- and y- directions), so a two dimensional approximation can be used. Fuel-air mixture enters from the left side with uniform temperature T_{g_0} and uniform velocity u_g . Two main zones exist in PRBs, the preheat zone and the combustion zone. Due to solid conduction and radiation heat transfer from the reaction zone, the temperature of the solid phase in the preheat zone is higher than that of the gas phase. This makes enthalpy transfer by convection from solid to gas which results in preheating the gas flow, whereas in the combustion region, enthalpy is transferred from gas to solid, by reverse mechanism of the preheat zone.

2.1. Governing Equations

Because of non-local thermal equilibrium between the solid and gas phases, separate energy equations in transient condition are considered. In addition, radiative heat transfer equation is used to find the radiative term in the solid energy equation. The following assumptions are considered in the present simulation:

- Plug flow assumption is considered
- The solid is assumed to be a gray medium that can emit, absorb and scatter radiation.
- Gas radiation is ignored.
- Scattering is assumed to be isotropic.

To save space, only the non-dimensional form of governing equations are presented here as follows.

Non-dimensional gas energy equation:

$$\frac{\phi}{\Gamma.P_1}\left(\frac{\partial^2\theta_g}{\partial\eta_x^2}+\frac{\partial^2\theta_g}{\partial\eta_y^2}\right)-\phi\frac{P_2}{\Gamma}\frac{\partial\theta_g}{\partial\eta_x}-(1-\phi)\frac{P_3}{\Gamma}(\theta_g-\theta_p)+\phi\frac{P_4}{\Gamma}\delta(x)=\phi\frac{\partial\theta_g}{\partial t^*} \quad (1)$$

Non-dimensional solid energy equation:

$$(1-\phi)P_5\left(\frac{\partial^2\theta_p}{\partial\eta_x^2}+\frac{\partial^2\theta_p}{\partial\eta_y^2}\right)-\nabla^*.Q_{rad}+(1-\phi)P_3(\theta_g-\theta_p)=(1-\phi)\frac{\partial\theta_p}{\partial t^*} \quad (2)$$

Where, $\delta(\mathrm{x})$ is a function which defined as unity in the combustion zone and zero elsewhere. All non-dimensional parameters used in Eqs. (1), (2) are given in the nomenclature.

Convection heat transfer between the gas and solid phases couples their equations together. Therefore, it is necessary to have an appropriate relation for the convective heat transfer coefficient. To this end, the following relationship is used [14]:

$$h=k_g\left[2+1.1\mathrm{Pr}^{1/3}\,\mathrm{Re}_p^{0.6}\right]/d_p \quad (3)$$

Where, Re_p is the particle-based Reynolds number which is defined as: $\mathrm{Re}_p=\dfrac{\rho_g u_g d_p}{\mu_g}$.

The dimensionless divergence of radiative heat flux, $\nabla^*.Q_{rad}$, which should be calculated from the radiative transfer equation can be written as follows:

$$\nabla^*.Q_{rad}=\frac{\partial Q_x}{\partial\eta_x}+\frac{\partial Q_y}{\partial\eta_y} \quad (4)$$

The non-dimensional radiative transfer equation in an emitting, absorbing and scattering medium can be written in general form as [15]:

$$\hat{s}.\nabla^* I^*(\vec{r},\hat{s}) = -\tau_1 I^*(\vec{r},\hat{s}) + \tau_1 I_b{}^*(\vec{r}) + \frac{\tau_2}{4\pi}\int_{\omega'=4\pi} I^*(\vec{r},\hat{s}')\varphi(\vec{r},\hat{s},\hat{s}')d\Omega' \tag{5}$$

The critical issue in the solution of the radiative transfer equation for the porous burner is the calculation of the radiative properties of the medium. In this study, optical thickness and scattering albedo are considered as representative of other properties. Optical thickness τ_0 , which shows the ability of the media to absorb and emit thermal radiation, is defined as $\tau_0 = \beta \times L_x$**,** where β is the extinction coefficient. Also, optical thickness measures the attenuation of the transmitted radiant power in a material. Another important radiant property of porous materials which can affect the thermal behavior of PRBs is scattering albedo and defined as the ratio of scattering coefficient to extinction coefficient ($\omega = \sigma_s / \beta$). In general, extinction coefficient is a function of the type of material, construction of the material and operating temperature. The scattering albedo is a strong function of material and relatively independent of structure and temperature [16]. It should be noted that the range of present values considered for optical thickness and scattering albedo are according to Ref. [16].

Once the RTE with appropriate boundary conditions was solved for computation of radiant intensities, the radiative heat flux can be calculated from:

$$Q_{rad}(\vec{r}) = \int_{4\pi} I^*(\vec{r},\hat{s})\hat{s}d\Omega \tag{6}$$

2.2. Initial and Boundary Conditions

For numerical solution of the gas and the solid phases, energy equations and also the RTE, an appropriate set of initial and boundary conditions are needed.

Solid and gas phases are assumed initially to be at ambient temperature as:

$$\theta_g = \theta_\infty \quad and \quad \theta_p = \theta_\infty \quad at \quad t^* = 0 \tag{7}$$

The following boundary conditions are applied to solve the gas energy equation:

$$\theta_g = \theta_{g_0} \qquad at \quad \eta_x = 0 \tag{8}$$

$$\frac{\partial \theta_g}{\partial \eta_x} = 0 \qquad at \quad \eta_x = 1 \tag{9}$$

$$\frac{\partial \theta_g}{\partial \eta_y} = P_6(\theta_g - \theta_\infty) \qquad at \quad \eta_y = 0 \tag{10}$$

$$\frac{\partial \theta_g}{\partial \eta_y} = -P_6(\theta_g - \theta_\infty) \qquad at \quad \eta_y = \eta_{Ly} \tag{11}$$

At inlet and outlet sections of the solid phase, due to convection heat transfer between the gas and solid phases and radiation heat transfer between solid matrix and its surrounding, the following boundary conditions are used:

$$Bi\left(\theta_g - \theta_p\right) + \frac{\varepsilon_{p_1}}{P_5}\left(\theta_i^4 - \theta_p^4\right) = -\frac{\partial \theta_p}{\partial \eta_x} \qquad at \quad \eta_x = 0 \tag{12}$$

$$Bi\left(\theta_p - \theta_g\right) + \frac{\varepsilon_{p_2}}{P_5}\left(\theta_p^4 - \theta_e^4\right) = -\frac{\partial \theta_p}{\partial \eta_x} \qquad at \quad \eta_x = 1 \tag{13}$$

Boundary conditions at upper and lower surfaces of the solid matrix are as follows:

$$\frac{\partial \theta_p}{\partial \eta_y} = P_7\left(\theta_p - \theta_\infty\right) \qquad at \quad \eta_y = 0 \tag{14}$$

$$\frac{\partial \theta_p}{\partial \eta_y} = -P_7\left(\theta_p - \theta_\infty\right) \qquad at \quad \eta_y = \eta_{Ly} \tag{15}$$

In order to solve the RTE, suitable boundary conditions should be considered. It is assumed that radiant intensities B1 and B2 are entered to the domain from inlet and outlet sections. So, the following boundary conditions are applied to the radiative transfer equation:

$$I^{*m}(0,\eta_y) = \frac{B_1'}{\pi} \qquad at \ \eta_x = 0 \tag{16}$$

$$I^{*m}(1,\eta_y) = \frac{B_2'}{\pi} \qquad at \ \eta_x = 1 \tag{17}$$

$$I^{*m}(\eta_x,0) = \varepsilon_B \frac{\sigma \theta_B^4}{\pi} + \frac{\rho_B}{\pi} \int_{\hat{n}.\hat{s}'<0} \left|\hat{n}.\hat{s}'\right| I^*\left(\eta_x, 0, \hat{s}'\right) d\Omega' \qquad at \ \eta_y = 0 \tag{18}$$

$$I^{*m}(\eta_x,\eta_{L_y}) = \varepsilon_T \frac{\sigma\theta_T^4}{\pi} + \frac{\rho_T}{\pi} \int_{\hat{n}.\hat{s}^{'}<0} \left|\hat{n}.\hat{s}^{'}\right| I^*\left(\eta_x,\eta_{L_y},\hat{s}^{'}\right) d\Omega^{'} \qquad at\ \eta_y = \eta_{L_y} \tag{19}$$

2.3. Discrete Ordinates Method ($S_N$approximation)

Due to the complex nature of RTE, many methods have been proposed for simplification of this equation. One of commonly used methods to solve the radiative transfer equation is the discrete ordinates method. This method is based on the discrete representation of the directional behavior of the radiative intensity. DOM transforms the equation of heat transfer into a set of simultaneous partial differential equations. The name S_N approximation shows that N different direction cosines are used for each principal direction. Altogether, there are $n = N(N+2)$ different directions to be considered in three dimensional computational domain. The general form of RTE is solved for a set of n different directions $\hat{s}_i$, *i=1,2,...,n*[14], such that the integrals over direction are replaced by the following relation:

$$\int_{4\pi} f(\hat{s})d\Omega \approx \sum_{i=1}^{n} w_i f(\hat{s}_i) \tag{20}$$

Where, w_i are the quadrature weights associated with the directions $\hat{s}_i$.

Hence, the RTE is approximated by a set of n differential equations as follows:

$$\hat{s}_i.\overset{*}{\nabla} I^*(\vec{r},\hat{s}_i) = -\tau_0 I^*(\vec{r},\hat{s}_i) + \tau_1 I_b^*(\vec{r}) + \frac{\tau_2}{4\pi}\sum_{j=1}^{n} w_j I^*(\vec{r},\hat{s}_j)\varphi^*(\vec{r},\hat{s}_i,\hat{s}_j) \qquad i = 1,2,...,n \tag{21}$$

For 2-D Cartesian coordinate problems, Eq. (21) becomes:

$$\xi^m \frac{\partial I^m}{\partial x} + \eta^m \frac{\partial I^m}{\partial y} = -\beta I^m + \sigma_a I_b + \frac{\sigma_s}{4\pi} \sum_{m'} w^{m'} \varphi^{m'm} I^{m'} \tag{22}$$

Where, $m \;\; and \;\; m'$ are outgoing and incoming directions, respectively, and $\xi^m \; and \;\; \eta^m$ are the components of unit vector $\hat{s}_i$. Finally, the radiative heat flux inside the porous medium can be determined from the following equation:

$$Q_{rad}(\vec{r}) = \int_{4\pi} I^*(\vec{r}, \hat{s}_i)\hat{s}_i d\Omega \approx \sum_{i=1}^{n} w_i I^*{}_i(\vec{r})\hat{s}_i \tag{23}$$

Numerically, using the finite difference method, the radiative intensity at each node in desired directions can be determined from:

$$I_{i,j}{}^m = \frac{IX^m + IY^m + \sigma_a I_{b,i,j} + S^m}{\beta + X^m Sign(X^m) + Y^m Sign(Y^m)} \tag{24}$$

Where,

$$X^m = \frac{\xi^m}{\Delta x}$$

$$Y^m = \frac{\eta^m}{\Delta y}$$

$$S^m = \frac{\sigma_s}{4\pi} \sum_{m'} w^{m'} \phi^{m'm} I_{i,j}{}^{m'}$$

$$IX^m = X^m u_0(X^m) I^m{}_{i-1,j} - X^m u_0(-X^m) I^m{}_{i+1,j}$$

$$IY^m = Y^m u_0(Y^m) I^m{}_{i,j-1} - Y^m u_0(-Y^m) I^m{}_{i,j+1}$$

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

$$u_0(x) = \begin{cases} 1 & x > 0 \\ 0 & x < 0 \end{cases}$$

Therefore, from Eq. (24), the value of radiant intensities at each nodal point (i,j) can be computed.

2.4. Method of solution

In order to obtain the temperature distribution of the gas and solid phases and also the values of radiative heat fluxes in the solid phase, the governing equations with related boundary conditions are solved iteratively. Because of transient behavior of the energy equations, the Alternative Direction Implicit (ADI) scheme is used to discretize these equations. In numerical analysis, the ADI scheme is an implicit finite difference method for solving parabolic, hyperbolic, and elliptic partial differential equations. The advantage of the ADI is that the equations have to be solved in each step, have a simpler structure and can be solved efficiently with the tri-diagonal matrix algorithm. The idea behind the ADI is to split the finite difference equations into two ones, one with the x-derivative taken implicitly and other with the y-derivative.

By the above numerical methods, the temperature distributions in the solid and gas phases and also the values of radiant heat flux at each nodal point and each time step can be computed by the following procedure:

1. Initial values for temperature and radiative heat flux are considered.
2. The tri-diagonal matrix algorithm is applied to solve the discrete form of the gas energy equation to obtain the gas temperature distribution.
3. Radiative transfer equation is solved to determine the values of I^*, Q_{rad} and $\nabla^*.Q_{rad}$ at each nodal point.
4. With the values of θ_g *and* $\nabla^*.Q_{rad}$, obtained from the steps 2 and 3, the porous energy equation (Eq. (2)) is used to compute the temperature of solid phase at each node.
5. Steps 2 to 4 are repeated until convergence is obtained.

It should be noted that convergence is achieved when changes in temperature and radiative intensity between two consecutive iterations do not exceed 10^{-3} at each grid point. Steady state condition is achieved when temperature changes in two consecutive time steps are less than 10^{-5}.

3. Comparison of the numerical results in the steady condition with those in the literature

In order to examine the validity of the present numerical code, the numerical results for a test case in steady condition are compared with those obtained by Talukdar et al. [8]. They analyzed a 2-D rectangular porous radiant burner. In their work, the solid phase was assumed to be absorbing, emitting and scattering while the gas phase was considered transparent to radiation. They used the collapsed dimension method to solve the radiative term of solid energy equation. The non-dimensional parameters of the PRB system are based on Ref. [8] and given in Table 1. The steady state gas temperature distribution is shown in Fig.2. Radiation and conduction heat transfer of the solid phase recirculates heat from the combustion zone to the entrance zone of the porous burner and consequently the fuel-air mixture is preheated by convection heat transfer with porous matrix. Due to the heat released in the combustion region, the maximum temperature of the gas phase occurs in this zone. Downstream this zone, the gas temperature decreases by converting the gas enthalpy to thermal radiation towards the outlet section. The increase in the gas temperature up to heat generation zone is clearly seen in this figure after which the temperature decreases towards the outlet section. However, a satisfactory agreement is found between the present results and those of Ref. [8]. Small discrepancy seen downstream the combustion zone of the porous burner may be due to different boundary conditions used at the exit section in these studies which detoriate the numerical results in the upstream of this section.

4. Results and Discussion

The main purpose of this study is to analyze the transient behavior of a porous radiant burner. The optical thickness τ_0 is the potential of a medium in radiation heat transfer. The greater the optical thickness, the better the performance of the medium in radiation transfer. To illustrate this effect, time history of the burner for two optical thicknesses is shown in Figs.3 and 4. In both figures by increasing time, the gas temperatures increases until reaching steady state condition. The heat generation zone is situated in the middle of porous layer, so the maximum values of temperatures occurs in the combustion zone due to heat releasing phenomenon. Also it is seen in these figures that, the gas phase is preheated by increasing time and this is due to convection heat transfer with solid phase. The required time to reach steady state condition for the case of $\tau_0 = 1$due to the dominance of radiation effects is less than that of $\tau_0 = 0.001$. It means that radiation phenomenon caused the temperature field become steady in a short time period.

Besides, one can found from Figs.3 and 4 that the temperature distributions have similar patterns at each time step and they reach to steady condition along a very short time period.

Gas and solid temperature distributions in the steady condition are presented in Fig.5. In the premixed region, the solid temperature due to the radiative fluxes from combustion zone is higher than the gas temperature. Due to the heat generation in the combustion zone, the gas temperature increases sharply before this domain and the porous temperature has similar trend due to high rate of convection heat transfer between these two phases. Also it is depicted from Fig.5 that the maximum values of gas and solid temperatures occur inside the combustion zone, where there is a heat source domain inside the gas flow.

Fig.6 shows the radiative heat flux distribution of the solid phase along the flow direction. The value of downstream radiative flux, Q^+, at the end of solid matrix, $\eta_x = 1$, is called radiant output, which is a very important parameter in increasing radiant burner efficiency. The upstream radiative heat flux, Q^-, at the inlet of porous burner, $\eta_x = 0$, is the energy loss which decrease the burner efficiency. It is desirable that this value should be very low to have more efficient PRB.

The effect of optical thickness on thermal characteristics of PRBs are studied in Fig.7, in which the gas and solid temperatures distributions along the centerline at steady condition are plotted. These figure shows that the temperature distributions are much affected by the optical thickness, such that both gas and porous temperatures decrease with increasing in τ_0. This behavior can be explained by noticing to this fact that, the porous burner with high optical thickness can convert more gas enthalpy to thermal radiation which causes a decrease in maximum gas temperature and increasing in radiant output. As it is depicted in Fig.7, high values of gas and porous temperatures take place in porous burners with zero optical thickness in which there is not any energy conversion from gas enthalpy into thermal radiation, such that all of the thermal energy released from combustion process leads to high temperature gas flow through the burner. Accordingly, large variations of temperature with changing optical thickness from 0 to 2 are observed in Fig. 7. In addition, in the preheat zone where there is no heat generation, the effect of optical thickness on the temperature is less in comparison to the combustion zone.

The gas and solid temperatures for different scattering albedo are represented in Fig. 8. This figure shows that decreasing the scattering albedo reduces the maximum temperatures of the gas and solid phases due to decrease the conversion of enthalpy to thermal radiation.

Porosity or void fraction is a measure of the void spaces in the porous material, and is defined as the volume of voids over the total volume. To show the effect of this parameter, gas temperature distributions for three porosities are shown in Fig. 9. As it is seen in this figure, by increasing porosity which leads to decrease in the volume of solid phase, the rate of energy conversion between gas enthalpy and thermal radiation decreases and finally high temperature gas flows through the burner without considerable temperature change.

Furthermore, the performance of PRBs is usually defined as radiant efficiency which is the ratio of the radiant output to the total amount of energy released via the combustion process as follows [17]:

$$\eta_r = \frac{heat\ released\, by\ radiation\ at\ output}{heat\ released\, by\ complete\ combustion} \times 100$$

In Fig.10 time history of the variation of radiant efficiency is presented. This figure shows zero value for the burner efficiency at time of starting, such that it increases rapidly with time and reaches to its steady state value. Fig. 11 shows the effect of the optical thickness on the radiant efficiency. Also, in Fig. 12 the effect of scattering albedo on the radiant efficiency is seen. As it is observed in Fig. 11, the higher the optical thickness, the more the radiant efficiency. In addition, reverse phenomenon happens about scattering albedo. It can be concluded that using porous layers with high optical thickness and small scattering albedo in the construction of radiant burners lead to increase in the performance of PRBs by converting more thermal energy from gas enthalpy to thermal radiation.

5. Conclusions

The thermal characteristics of a porous radiant burner in the transient condition was studied numerically in this work. To this end, a 2-D solid matrix in Cartesian coordinates with combustion chamber located in its middle was considered. The governing equations include the gas and solid energy equations, and also the radiative transfer equation were solved by numerical method. Alternative direction implicit scheme and discrete ordinates method are used to discretize the energy equations and solve the RTE, respectively. Numerical results show that the required time to reach steady state condition is relatively short in porous radiant burners. The reason of this phenomenon is that the nature of radiation mode which has the most effective contribution in the transferring energy is instantaneous, such that the transient time period is only due to the existence of conduction and convection heat transfer modes. It can be concluded, in the analysis of such thermal systems, the duration of transient thermal behavior is not important and therefore the investigation can be considered in steady state condition. In addition, findings show that two important radiative properties on the behavior of PRBs are the optical thickness and scattering albedo. It is better that in construction of porous radiant burners, porous media with large optical thickness and small scattering albedo are used that leads to having PRBs with high performance.

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Figures:

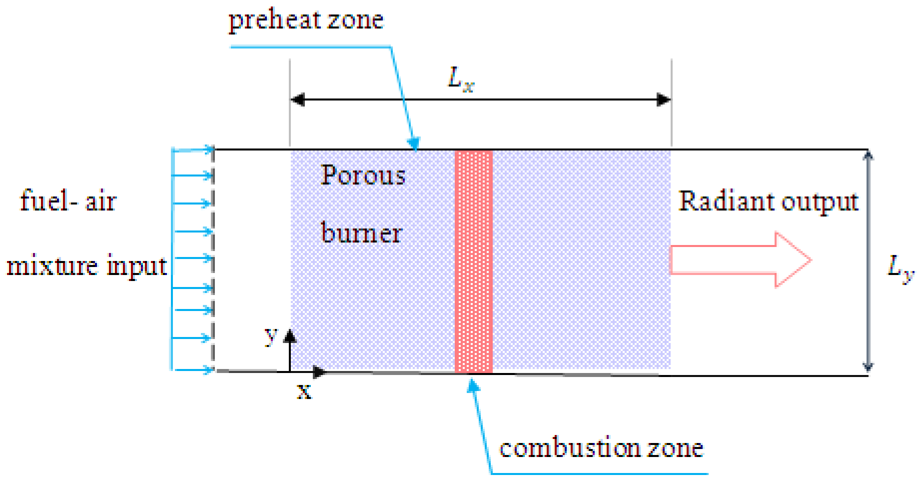

Fig.1: The schematic representation of porous radiant burner.

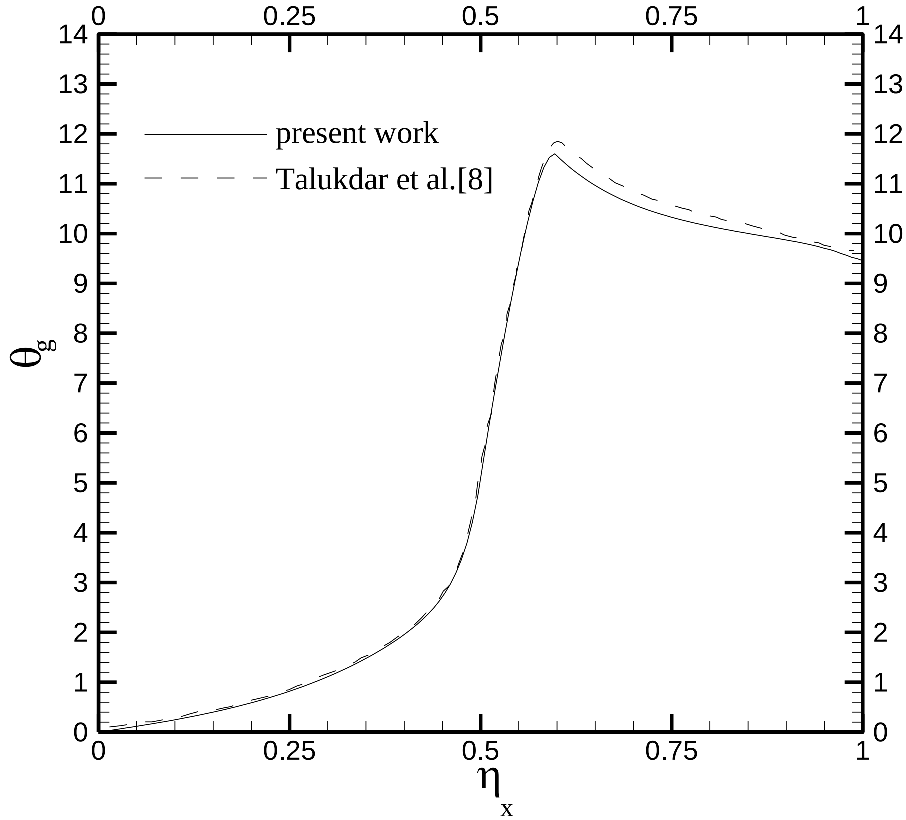

Fig. 2: Gas temperature distribution along PRB

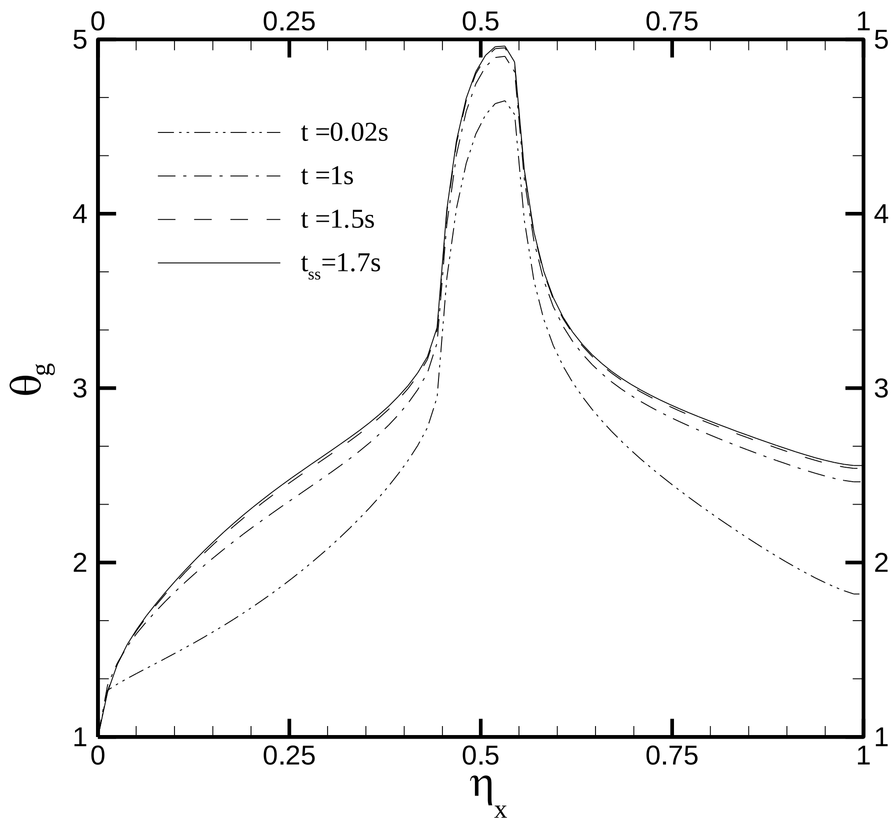

Fig.3: Time history of gas temperature at center line with $\tau_0 = 1.0$ $(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$ $P_6 = 16.0, P_7 = 0.2, Bi = 8.0,\ \omega = 0.0, \Gamma = 3.47\times10^{-3})$

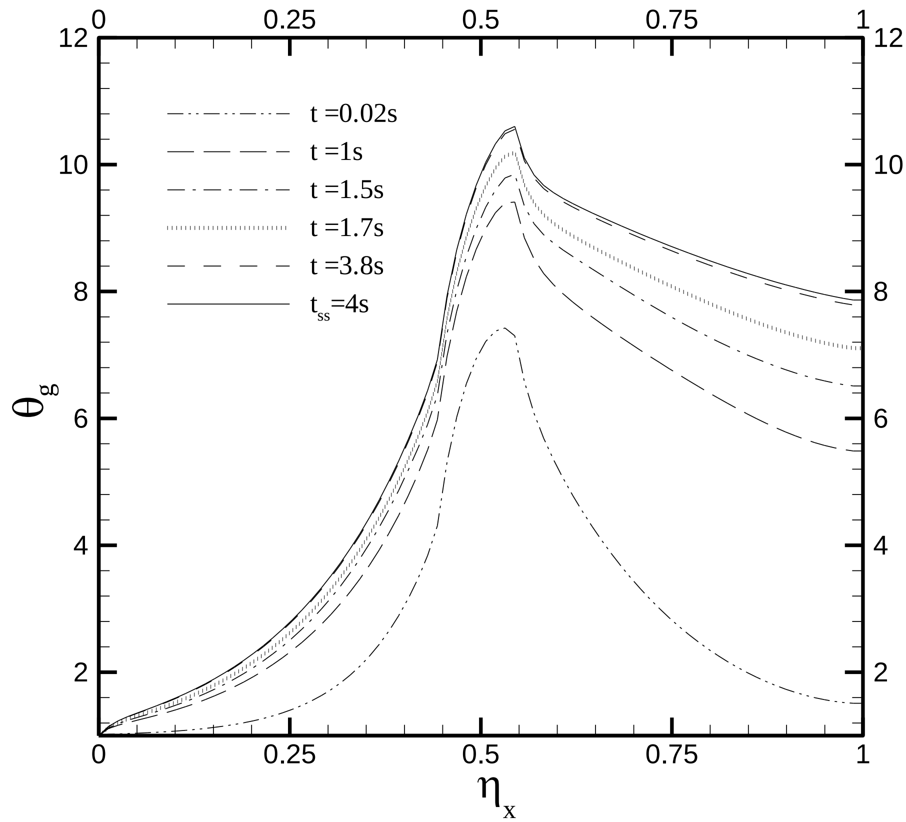

Fig.4:Time history of gas temperature at center line with $\tau_0 = 0.001$
$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \omega = 0.0, \Gamma = 3.47\times10^{-3})$

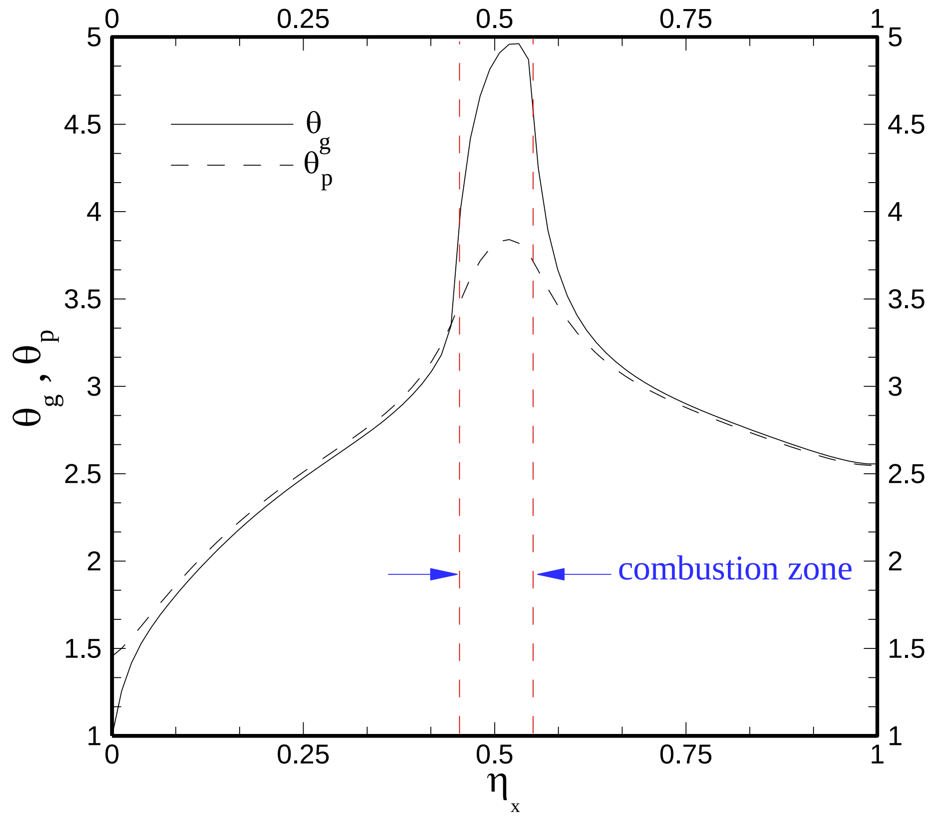

Fig.5: Gas and solid temperature distributions in steady condition at $\eta_y = 0.5$.
$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \tau_0 = 1.0, \omega = 0.0, \Gamma = 3.47\times10^{-3})$

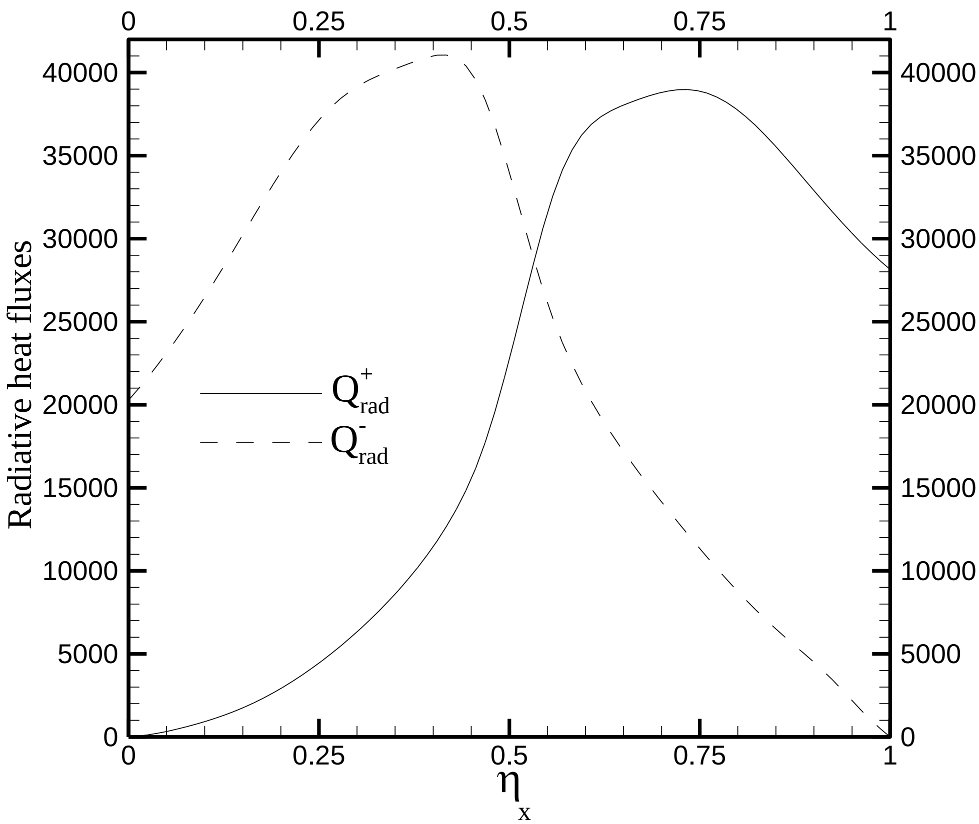

Fig.6: Radiative flux distribution at steady state condition in centerline
$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22, P_6 = 16.0$
$, P_7 = 0.2, Bi = 8.0, \tau_0 = 1.0, \omega = 0.0, \Gamma = 3.47\times10^{-3}, B_1 = 0.0, B_2 = 0.0)$

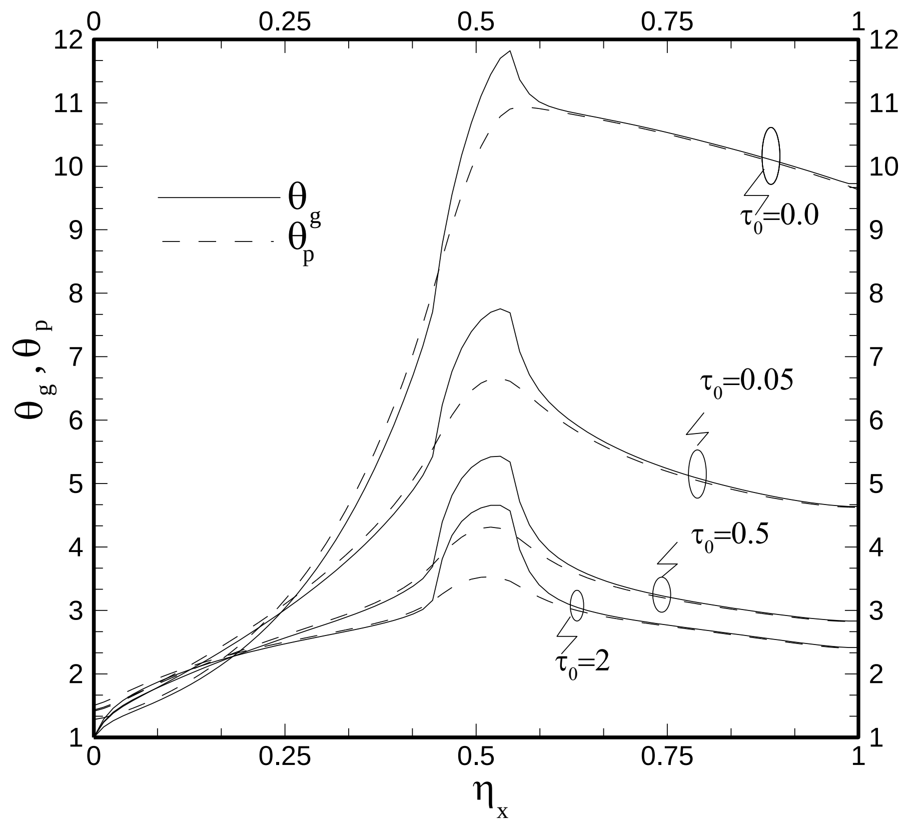

Fig.7: Effect of optical thickness on the temperatures of the gas and solid phases ($P_1 = 0.24, P_2 = 405.2, P_3 = 18.66 \times 10^4, P_4 = 44.7 \times 10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \omega = 0.0, \Gamma = 3.47 \times 10^{-3}$)

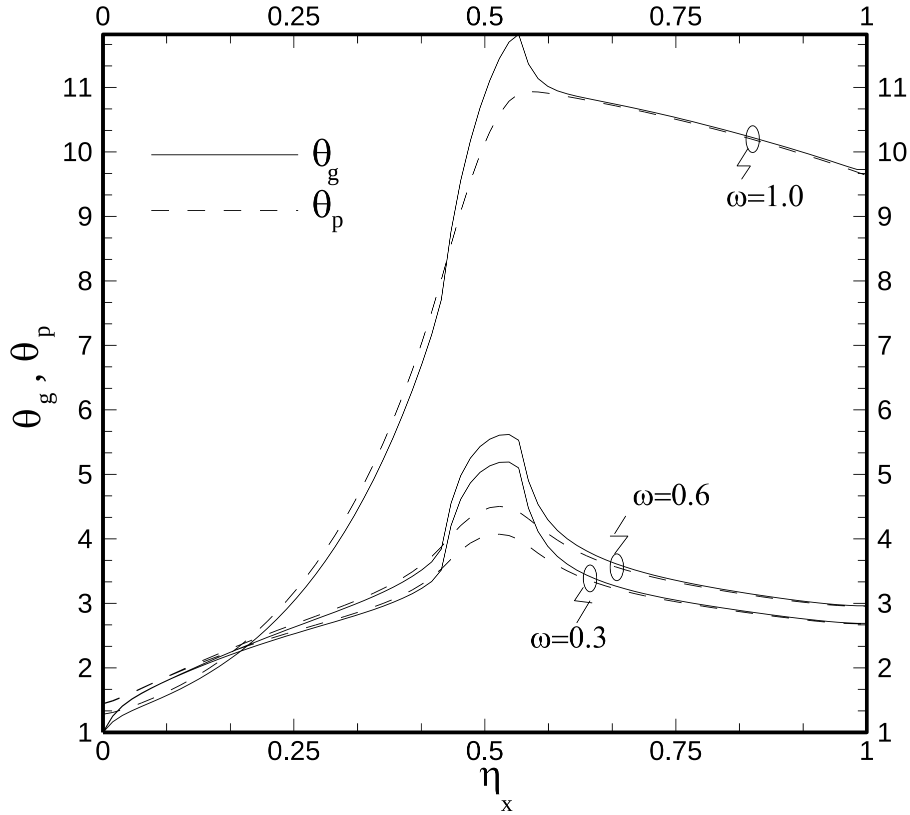

Fig.8: Effect of scattering albedo on the temperatures of the gas and solid phases
$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \tau_0 = 1.0, \Gamma = 3.47\times10^{-3})$

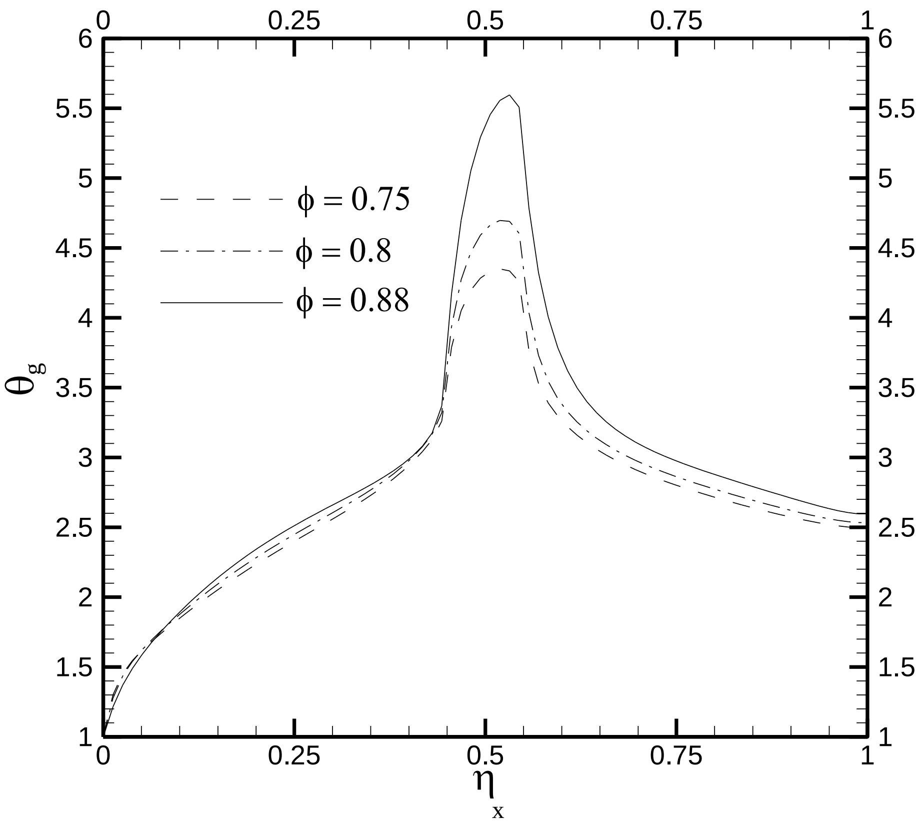

Fig.9: Effect of porosity on the gas temperature.

$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \tau_0 = 1.0, \omega = 0.0, \Gamma = 3.47\times10^{-3})$

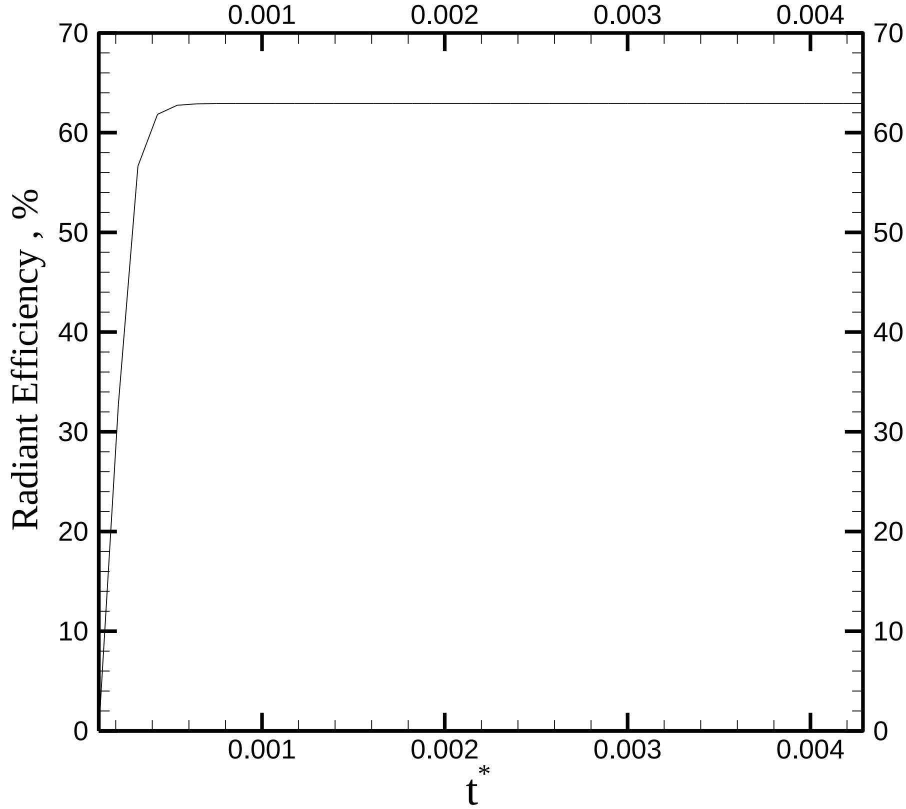

Fig.10: Time history of burner efficiency

$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \tau_0 = 1.0, \omega = 0.0, \Gamma = 3.47\times10^{-3})$

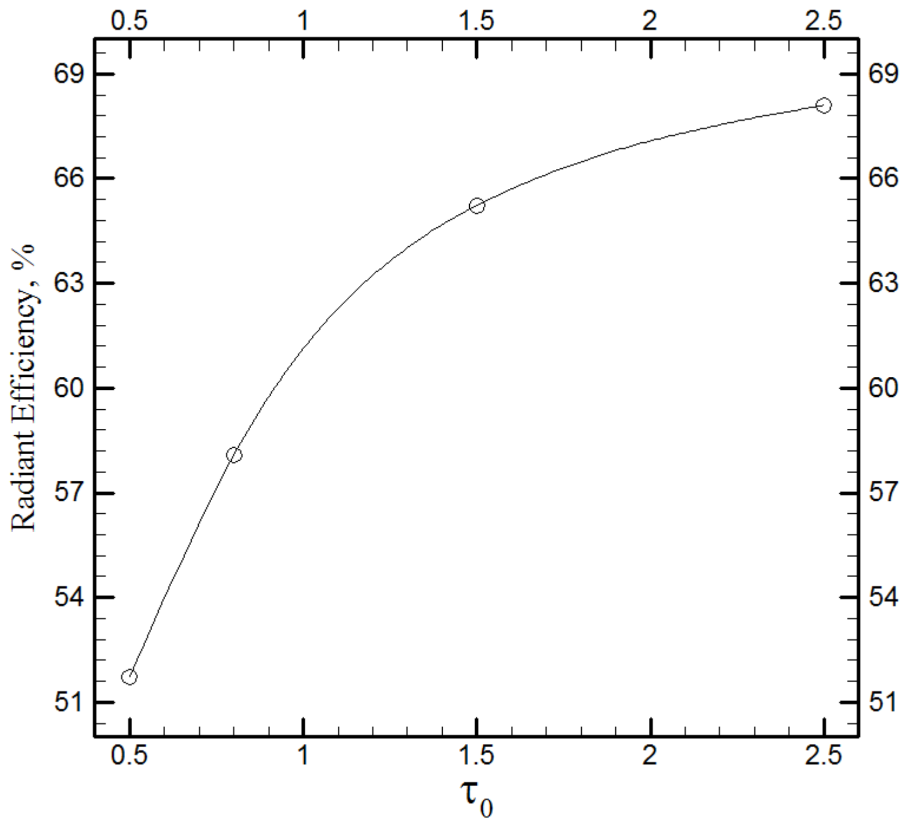

Fig 11: The effect of optical thickness on the radiant efficiency

$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66 \times 10^4, P_4 = 44.7 \times 10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \omega = 0.0, \Gamma = 3.47 \times 10^{-3})$

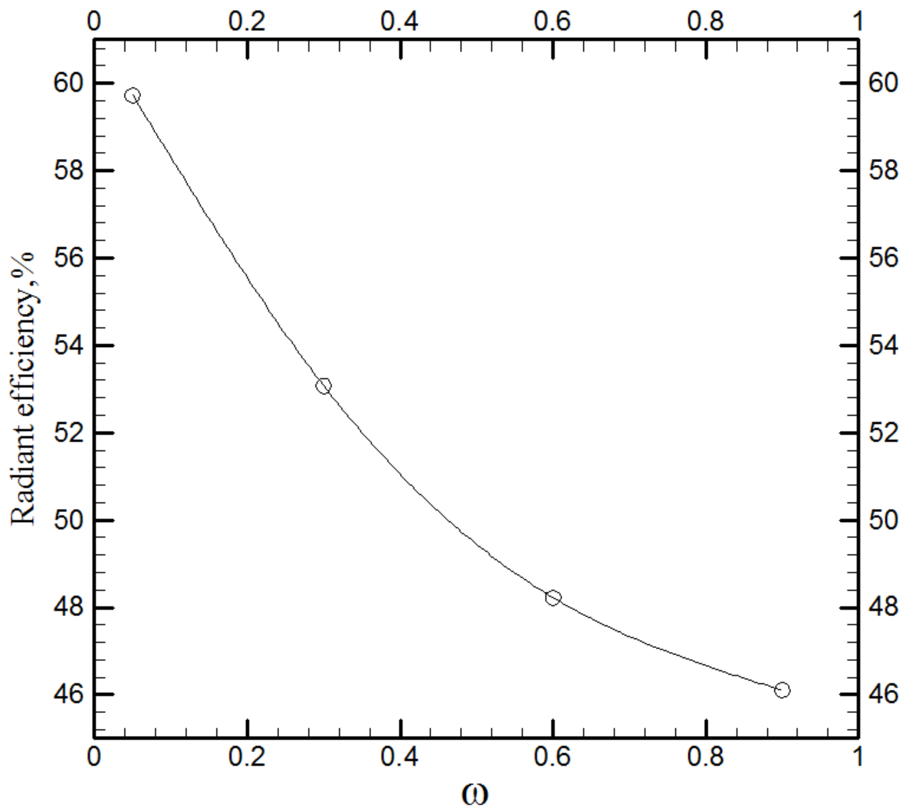

Fig 12: The effect of scattering albedo on the radiant efficiency

$(P_1 = 0.24, P_2 = 405.2, P_3 = 18.66\times10^4, P_4 = 44.7\times10^3, P_5 = 333.22,$
$P_6 = 16.0, P_7 = 0.2, Bi = 8.0, \tau_0 = 1.0, \Gamma = 3.47\times10^{-3})$

Tables:

Parameters	Values from Talukdar et al. [8]
P_1	1.2
P_2	33.33
P_3	333.33
P_4	10^8
P_5	66.66
τ_0	1.0
ω	0.0
Γ	3.33×10^{-3}
Bi	5

Table1: Non-dimensional parameters

Nomenclature

A	surface area per unit volume (m^2/m^3)
$B_{1,2}$	incoming radiations, (W/m^2)
$B'_{1,2}$	non-dimensional incoming radiations, $B_{1,2}/\sigma T_{g0}^4$
Bi	Biot number, hL_x/k_p
c_g	special heat of gas, (J/kgK)
c_p	special heat of solid, (J/kgK)
d_p	particle diameter, (m)
h	convective heat transfer coefficient, (W/m^2K)
I	intensity, (W/m^2)
I^*	non-dimensional intensity, $I/\sigma T_{g0}^4$
K_g	gas thermal conductivity coefficient, (W/mK)
K_p	solid thermal conductivity coefficient , (W/mK)
L_x	length of porous medium, (m)
L_y	height of porous medium, (m)
Nu	Nusselt number, hL_x/k_g
P_1	The ratio of solid radiation to gas conduction, $\sigma T_{g0}{}^3 L_x/k_g$

P_2	The ratio of gas convection to solid radiation, $\rho_g c_g u_g / \sigma T_{g0}^3$
P_3	The ratio of convection between two phases to solid radiation, $hAL_x / \sigma T_{g_0}^3$
P_4	The ratio of combustion heat to solid radiation, $L_x \dot{Q} / \sigma T_{g0}^4$
P_5	The ratio of solid conduction to solid radiation , $k_p / \sigma T_{g0}^3 L_x$
P_6	The ratio of gas convection to gas conduction at boundaries, $h_{wg} L_x / k_g$
P_7	The ratio of solid convection to solid conduction at boundaries, $h_{wp} L_x / k_p$
q_{rad}	radiative heat flux, (W/m^2)
Q_{rad}	non-dimensional radiative heat flux, $q_{rad} / \sigma T_{g0}^4$
Re_{d_p}	Particle-based Reynolds number, $\rho_g u_g d_p / \mu_g$
$\hat{s}_i$	direction vector in RTE
T	temperature, (K)
T_{g_0}	initial temperature of gas, (K)
T_∞	ambient temperature, (K)
t	time, (s)
t^*	non-dimensional time, $\sigma T_{g0}^3 t / \rho_p c_p L_x$
u_g	gas velocity along x direction, (m/s)
x	coordinate along flow direction, (m)

y	coordinate perpendicular to the flow direction, (m)

Greek symbols

β	extinction coefficient, (m^{-1}) , $\sigma_a + \sigma_s$
∇^*	non-dimensional gradient operator, $L_x \nabla$
$\Delta\eta_x$	non-dimensional grid spacing along x-axis
$\Delta\eta_y$	non-dimensional grid spacing along y-axis
ε	emissivity
η_x	non- dimensional x coordinate, x/L_x
η_y	non- dimensional y coordinate, y/L_x
Γ	non-dimensional parameter, $\rho_g c_g / \rho_p c_p$
θ	non-dimensional temperature, T/T_{g_0}
ρ_g	gas density, (m^3/kg)
ρ_p	solid density, (m^3/kg)
ρ_w	wall reflection coefficient
σ	Stephan-Boltzmann coefficient, (w/m^2K^4)
σ_a	absorption coefficient, (m^{-1})
σ_s	scattering coefficient, (m^{-1})
$\theta_{g,p}$	non–dimensional temperature, $T_{g,p}/T_{g_0}$
	Non dimensional temperature ($T_{g,p}/T_{g_0}$)

τ_0	optical thickness, βL_x
τ_1	non-dimensional parameter, $\sigma_a L_x$
τ_2	non-dimensional parameter , $\sigma_s L_x$
ϕ	Porosity
φ	scattering phase function
w	weighting constant

Subscripts

b	black body
B	Bottom
e	exit of the porous matrix
g	Gas
i	inlet of the porous matrix
p	Porous
T	Top

Superscripts

m	outgoing radiation direction
m'	Incoming radiation direction
$+$	upstream direction
$-$	downstream direction

Biography

H. Shabani Nejad is a Ph. D student in mechanical engineering in the Faculty of Engineering, Shahid Bahonar University, Kerman, Iran. Her M.Sc. thesis is about radiation effect in porous radiant burner. Her research interest includes: radiation heat transfer, porous media, and combustion.

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E. Jahanshahi Javaran is an assistant professor of Mechanical Engineering at Graduate University of Advanced Technology, Kerman, Iran. He received his BSc, MSc and PhD degrees from Shahid Bahonar University of Kerman, Iran in the field of mechanical engineering. He is now working on lattice Boltzmann method, solar energy and desalination systems. He has some publications in the field of radiative heat transfer, rheology of particle suspensions and solar energy.