A Numerical Study of the Effects of Material Properties on Performance of a Non-sprayed Porous Burner (NSPB)

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Abstract

A mathematical model based on the implicit finite difference schemes, have been developed for investigating the performances of a non-sprayed liquid kerosene porous burner (NSPB). The study is carried on the investigation in terms of temperature profiles and radiant output efficiency. The porous media are designed and utilize for both evaporation and combustion process. The NSPB consists of a porous evaporator (PE) and a porous combustor (PC). The PC is surrounded by a wall air jacket for air preheating. The liquid kerosene is completely evaporated within the PE that is self-sustained via thermal radiation from the PC. Afterwards, the vaporized kerosene and preheated air have been mixed in the small mixing chamber which is located between the PE and the PC. Then, the homogeneous combustion occurs in the PC, instead of heterogeneous combustion as is occurred in the conventional sprayed burner. The results show that, the prediction of temperature profiles has similar trends to those of the experiment. The radiant output efficiency and maximum temperature is decreased with increasing porosity while the optical thickness dose not significantly affect radiant output efficiency and maximum temperature..

Keywords*:* Porous burner, Porosity, Absorption coefficient, Evaporation, Liquid fuel

1. Introduction

There is a wide interest in a porous burner due to the advantage of its higher thermal efficiency and lower emission (e.g. NO and CO) than a conventional burner. The porous burner achieves excess enthalpy combustion because it provides a means of internal heat recirculating without an external heat exchanger surrounding the combustion chamber (i.e. the porous medium acts as a heat exchanger). The combination of conduction, convection and radiation heat transfer causes heat recirculation in the porous burner. At a post-flame zone of the burner, heat is transferred from the burned gases to the solid matrix by convection. Immediately, heat is recirculated by solid to solid conduction and radiation from the post-flame zone to the pre-flame zone. Thus, heat is transferred convectively from the hot solid to the incoming gas mixture at the pre-flame zone. This results in increasing flame temperature, burning velocity and extension of lean flammability limit. Because a porous burner can be operated at a higher excess air ratio, CO emission is decreased. Moreover, the radiative heat transfer inside a porous matrix is greater than the gases. This enhances heat transfer from the hot gas to the solid at the reaction zone, and suppresses the flame temperature, i.e. low NO_x formation. In other words, the combustion zone temperatures can be controlled so that CO and NO_x emissions are minimized. Therefore combustion within porous media have been fused by many researchers. Useful review papers of combustion within porous media were published by Howell et al. [1], Trimis and Durst [2], Kamal et al. [3], Wood and Harris [4], and Mujeebu et al. [5]. However, almost all of the studies have been focused on premixed gaseous fuel porous burners.

A few studies of liquid fuel sprayed combustion within porous media have been reported by Martynenko et al., [6]; Kayal and Chakravarty [7]; Kaplan and Hall, [8]; Vijaykant and Agrawal [9]. However, the liquid atomization was required for fuel vaporization. The utilization of porous media both in evaporation and combustion process were not studied.

Beneficial uses of porous medium to enhance combustion and evaporation process were proposed by Jugjai and Pongsai [10] and Wongwatcharaphon et al. [11]. The evaporation and combustion process within porous media was investigated by numerical and experimental study. Results indicated that evaporation within a upstream porous medium followed by a matrix stabilized flame within downstream porous medium can be achieved. Moreover, the effect of firing rate and equivalence on the thermal structures was reported. However, the effects of properties of porous media on temperature profile and burner performance was not reported.

In order to broaden the knowledge, the effect of properties of porous medium i.e., porosity and optical thickness on thermal structure and burner performance in term of radiant output efficiency are reported in this study. A numerical model of non-sprayed porous burner based on previous work [11], so-called NSPB was performed.

2. Numerical Model

2.1 Burner geometry

Fig. 1 shows the geometry of NSPB. The system consists of two porous inert media which is surrounded by an air jacket. The upstream and downstream porous are a porous evaporator (PE) and a porous combustor respectively. Both porous media are coupled by thermal radiation emitted from the PC to the PE for preheating and evaporating the liquid kerosene. The fuel and air supplies are separated for safety. When the liquid kerosene at flows into the PE at section 1, it is preheated by the hot porous matrix of PE. When the liquid temperature reaches its boiling point T_b, it is completely evaporated and changed to vapor. The fuel vapor is further preheated by a solid, and flows out from PE at section 2 with temperature T_2 and mass flow rate of $\dot{m}_F$. At the same time, the combustion air with inlet temperature T_{ai} and flow rate of $\dot{m}_a$ flows through an annular air jacket surrounding the PC at section 4, and is preheated to T_{ao} at section 3. The preheated vapor fuel and the hot air meet and mix together in the mixing chamber to form a combustible mixture with flow rate $\dot{m}_{mix}$ and temperature T_{mix} prior to combustion within the

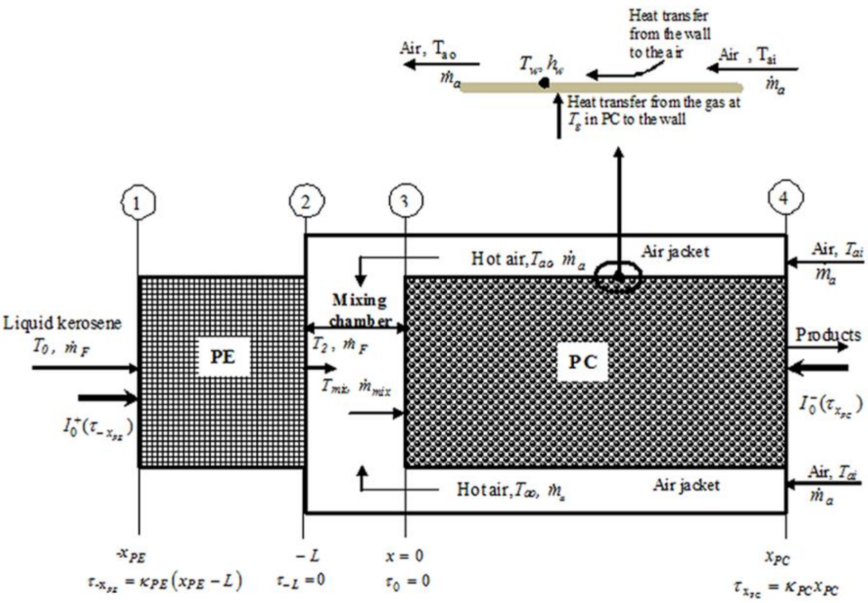

Fig. 1 Burner geometry

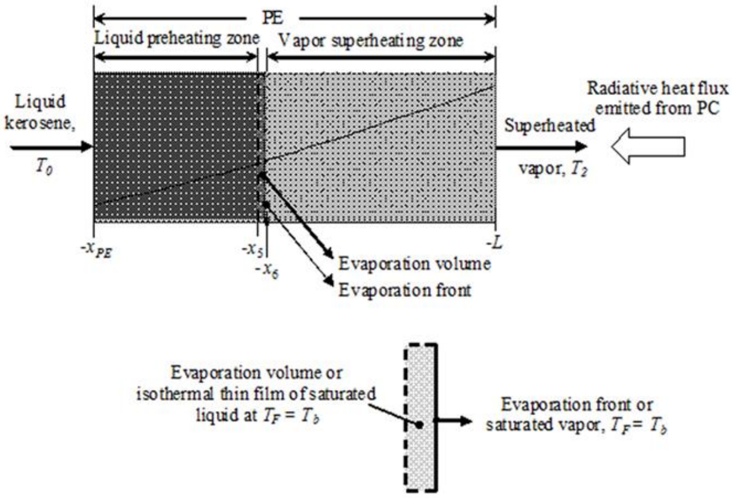

Fig.2 Evaporation model

PC. The total length of the burner is 260 mm (the PE section is 80 mm long, the mixing chamber is 20 mm long, and the PC section is 160 mm long). The computation domain is discretized into 200 grid points (100 uniform grid points for PE and 100 uniform grid points for PC). Both ends of the NSPB are exposed to black surroundings maintained at an ambient temperature providing incident radiation intensity $I_0^+\left(\tau_{-x_{FP}}\right)$ at section 1 and $I_0^-\left(\tau_{x_{PC}}\right)$ at section 4, respectively.

Fig. 2 shows an evaporation model within the PE. The computational domain is divided into three zones: a liquid preheating zone ($-x_{PE}$ to $-x_5$), an evaporation volume ($-x_5$ to $-x_6$), and a superheated vapor zone ($-x_6$ to $-L$). The liquid kerosene at T_0 is counter supplied with the net radiation flux direction from PC to PE. The liquid kerosene is gradually preheated until its temperature reaches its boiling point T_b. The sub-cooled boiling at saturation temperature is considered in this model. The evaporation process is considered to occur in the evaporation zone at the interfaces between the liquid preheating zone and the vapor superheated zone. At the interface, the fuel is assumed to be an isothermal thin film of a saturated liquid at T_b within an evaporation volume that have a thickness of 8×10^{4} m. The evaporation process immediately occurs at an evaporation front that is an exit of the evaporation volume. At the evaporation front, the saturated liquid immediately changes to saturated vapor at T_b and the fuel density significantly decreases because of phase change. Cause to the flow velocity of vapor is greater than the liquid, while the mass flow rates are conserved. Then, the vapor fuel gets preheated and its temperature rise in the superheated vapor zone.

The principal assumptions used in the model are as follows:

(a) One-dimensional for flow and heat transfer.

(b) Steady state and steady flow process.

(c) Non-radiating working gas behaves as ideal gases.

(d) PE and PC are able to emit and absorb thermal radiation in local thermal equilibrium, while radiative scattering is ignored.

(e) The flow is incompressible, because the flow velocity is very small when compared to the sound speed.

(f) The Lewis number is unity.

(g) The physical properties are constant.

(h) Good thermal insulation system.

(i) Porous media are non-catalytic both in PE and PC.

(j) The boiling temperature of kerosene is assumed constant, T_b = 250 ^{o}C. The flash evaporation process occurs only in a small volume in PE and the temperature of fluid in this volume is constant and equal to boiling temperature, T_b of kerosene.

(k) The evaporation process in PE is considered to be operated under relatively small liquid fuel mass flow rate. The latent heat of kerosene is very small when compared to heat radiation from PC. Therefore, the evaporation rate is assumed equal to the fuel mass flow rate.

(l) In PE, the mechanisms of hydrocarbon thermal cracking are negligible.

(m) In the mixing chamber, the superheated vapor fuel is assumed to be mixed immediately with swirling air to form homogeneous combustible mixture at temperature T_{mix}, before it enters the PC at section 3.

(n) The combustion reaction is described by an irreversible first-order reaction: Reactants$\longrightarrow$ Products.

(o) The reaction starts and gets completed in PC.

(p) The PC wall temperature at T_w is constant because of good thermal conductivity.

2.2. Basic Equation

The numerical model solves liquid, gas and solid phase energy equation for analyzing evaporation and combustion process. A one-dimension, single-step global reaction model and steady state approach are considered in this work. The influence of heat transfer between fluid and solid phase, fluid and solid conduction, and solid radiation are also considered. The conservation equations of species and energy both in PE and PC are discretized by finite differential approximations. An implicit difference scheme is adopted with respect to time, and a central difference scheme is adopted with respect to space. The convergence criteria for numerical computation of all variables are set to 10^{-6}. Time step and mesh size were tested using different values. As a compromise between an accuracy and computational time, 100 uniform grid points both in the PE and PC, and 0.1 s time steps are selected. The final error in the energy balance is usually less than 1%. The governing equations for solving problems have been reported by Wongwatcharaphon et al. [11].

3. Results

3.1. Model Validation

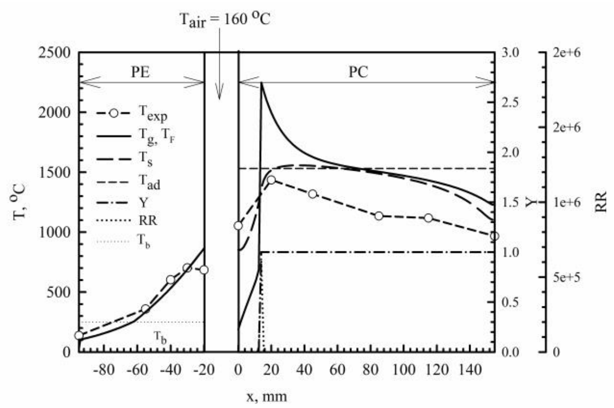

Fig. 3 Comparison between predicted and measured temperature profiles

Fig. 3 shows the fuel temperature (T_F), gas temperature (T_g), solid temperature (T_s), product mole fraction (Y), and dimensionless reaction rate (RR) at equivalence ratio of 0.64 and firing rate of 5.74 kW. The PE and PC were designed to couple with thermal radiation. The PE and the PC act as a radiative heat absorber and radiative heat emitter respectively. In the PE, the fuel temperature and the solid temperature are nearly identical because of a high heat transfer between the fluid and solid phases; however, solid temperature is higher than fuel temperature for along the PE. In the porous PC, the temperature profiles show similarity to the conventional premixed porous burner. The evaporation front is defined as a location that T_F is equal to T_b. The liquid kerosene is preheated before is evaporated in the evaporation volume. After that, the vaporized kerosene is continually preheated and flows out the PE as a superheated vapor.

In the PC, at the pre-flame zone, T_s is greater than T_g; therefore heat is transferred from solid to gas. While at the post-flame zone of PC, the gas temperature is higher than the solid temperature. Thus the gas phase transfers heat to the solid phase. Then, the solid recirculates heat from the post-flame zone to the pre-flame zone by solid to solid conduction and radiation for preheating the mixture upstream. This result in the maximum temperature of gas phase is greater than the corresponding adiabatic flame temperature (T_{ad}) according to internal heat recirculation. A preheated air temperature of T_{ao} = 162 °C before mixing with the vaporized kerosene followed by combustion in the PC. The numerical results have similar trends as the experimental and the model can be used to predict the importance information of the NSPB, e.g., maximum temperature, flame position, and the evaporation front. Moreover, the global energy balance calculation has a small error (<1%). This indicated that the proposed model can be used to predict the NSPB with high accuracy.

After model validation and performing a baseline simulation, a parametric study is done to investigate the effect of porosity and absorption coefficient on burner performance.

3.2 Effect of Porosity

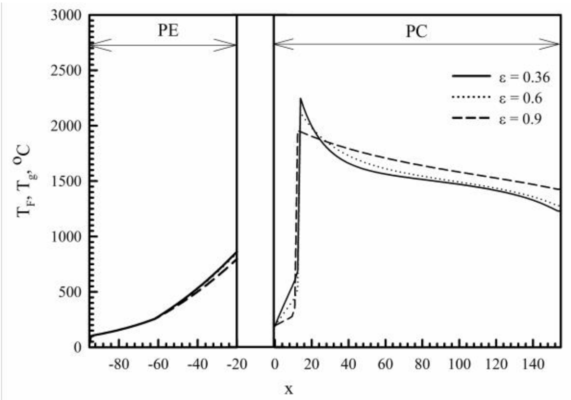

Fig. 4 Effect of the porosity of PC on thermal structure

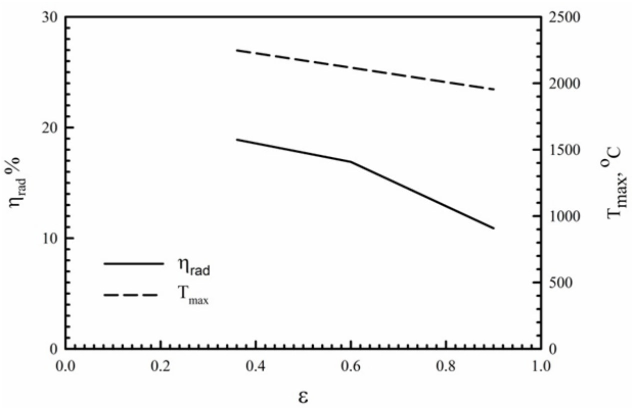

Fig. 5 Effect of porosity of the PC on radiant output efficiency and maximum temperature

Fig. 4 show the effect of porosity of the PC on fuel and gas temperature distribution at Φ = 0.64 and firing rate of 5.74. The variation of porosity of PC has effect on flow velocity, volumetric heat transfer, and effective thermal conductivity of porous medium. A higher porosity leads to increasing void volume and decreasing in volume of solid phase. The effective thermal conductivity, flow velocity, and volumetric heat transfer is decreased while convection loss at the downstream end increase. This results in lowering flame temperature (maximum temperature of gas phase). Moreover, porosity of PC dose not significantly affect evaporation front.

Fig. 5 show the effect of porosity of the PC on radiant output efficiency at Φ = 0.64 and firing rate of 5.74. The graph shows that increasing porosity result in decreasing the radiant output efficiency and maximum temperature. Because of lowering volumetric heat transfer between gas and solid phase, the solid temperature at the downstream is decreased and the convection loss at downstream end increase.

3.3 Effect of optical thickness

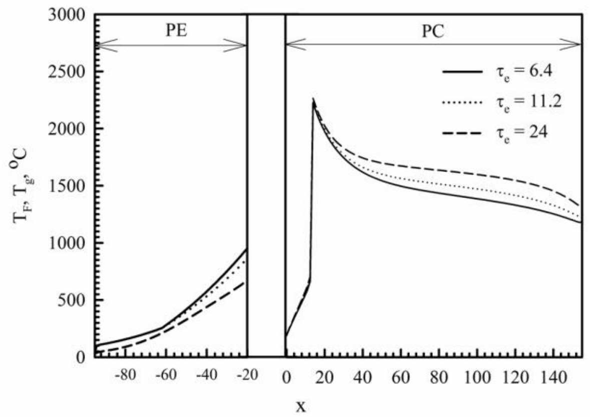

Fig. 6 Effect of the optical thickness of PC on thermal structure

Fig. 6 show the effect of the optical thickness of the PC on fuel and gas temperature distribution at Φ = 0.64 and firing rate of 5.74. The absorption coefficient is changed to vary optical thickness while the length of PC remains constant. Increasing in optical thickness, the maximum temperatures are not changed while the exit gas temperatures at downstream increase. Moreover the variation of optical thickness dose not significantly affects temperature different between the phases because the optical thickness does not affect volumetric heat transfer between gas and solid phase.

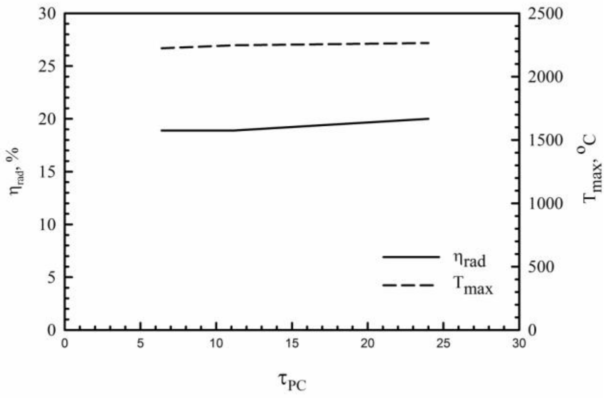

Fig. 7 Effect of the optical thickness of PC on radiant output efficiency and maximum temperature

Fig. 7 show the effect of optical thickness of the PC on radiant output efficiency at Φ = 0.64 and firing rate of 5.74. The radiant output efficiency and maximum temperature are quite constant with increasing optical thickness. Because not only the radiative energy feedback from post-flame zone to reaction zone increase but also the convection loss at the downstream is increased with increasing in exit gas temperature at downstream end that show in Fig.6. This may be attributed to a tradeoff between radiative energy feedback and convective heat losses. Therefore increasing in optical thickness does not affect maximum temperature and radiant output efficiency.

4. Conclusions

This paper presents the study of effect of porous material on performance of NSPB, in terms of maximum temperature and radiant output efficiency by mean of numerical modeling. The conclusions of the study are as follows:

1. The numerical results show the same trend as experimental results.
2. The radiant output efficiency and maximum temperature is decreased with increasing porosity.
3. The optical thickness does not significantly affect the radiant output efficiency and maximum temperature.

5. References

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