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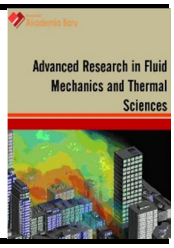

Experimental and Numerical Studies of Porous Media Combustion in Micro Burner

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ABSTRACT

Porous media combustion has been key point of interest for researchers from past couple of decades due its numerous advantages, such as remarkable low emission levels without compromising thermal efficiency. In this present work, dual layered micro burner was built with predefined thickness of reaction and preheat layer. Reaction layer was made up of alumina (discrete type) while preheat layer porcelain (foam type) material. Burner was successfully built to undergo both surface and submerged flames. A concept of equivalences ratio was enabled since its premixed combustion with natural air as one of the inlet along with butane. Trials were not just restricted at stoichiometric ratio but also carried out up to ultra-lean region. Additionally, numerical simulation was performed using commercially available computational fluid dynamics package so that porous media combustion phenomenon can be better analyzed and predicted. Finally, Thermal efficiency was calculated at critical equivalence ratios and emission parameters such as NOx and CO was continuously monitored which were under controlled limits.

1. Introduction

Day to day huge needs of people to make their life more comfortable has given rise to many devices and systems which mainly run on curdle oil and its other products. These products directly affect our surrounding weather and yearly seasons. Definitely the amount of fuel consumption and

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emissions from these products are negligible if considered individually, but when taken globally it becomes remarkable [1-3]. Hence it's a right time to think about energy systems which run mainly on fossil fuels. To solve any global energy issue, it's a recommended to start thinking its occurrence from grass root level. Hence attempts are made by many researchers to save single drop of fuel [4, 5]. Porous media burners (PMB) are the devices plays active role in modern market to ensure smooth flow of small, medium and large scale industrials needs for heat supply [6-10]. Burner with porous media combustion (PMC) shall play a vital role in enhancing both thermal efficiency and lower emissions.

From a decade, PMB are getting popular day by day due to availability of strong and better performing porous media (PM) materials. While on the other hand, size of the burner does play significant role in performance of the burner characteristics, specifically when burner is made to run at lean equivalences ratio (ER) [11, 12]. Wan *et al*., [13] made experimental study on ceiling gas temperature and made a point that due to much higher temperature in the flame region than the smoke temperature, multiple peak ceiling gas temperatures exist right above the multiple energy sources. While Valera-Medina *et al*., [14] made research work on combustion in tangential swirl burners, by involving numerical approach using GASEQ and CHEMKIN-PRO tool kit, and commented on flame stability and emissions parameters in ammonia methane flames. Song *et al*., [15] worked on ultra-low calorific gas combustion in a gradually-varied porous burner. Implied two major pathways of heat recirculation to reactants, firstly by enhancing the axial heat conduction and radiation heat transfer of porous media and next by increasing the temperature near combustor wall [16-20].

This present work mainly deals with running the micro porous media burner from stoichiometric to lean ER. So that optimum ER can be finalized thus making burner to run its best state. In addition, numerical simulation is performed at validate results surface flame temperature.

2. Methodology

A. Experimental setup

A two dimensional sketch of the burner housing is as shown in Fig. 1, while full overview of the different parts involved in this system is as shown in Fig. 2. A clear picture of laboratory setup is as shown in Fig. 3. Construction of burner housing involves two sections, upper and lower. Upper section is also called as reaction zone or combustion zone, while lower is popularly called as preheat zone. In the present work, reaction and preheat zone are kept at 20mm and 10 mm thick in size. Reaction zone is filled with discrete PM made up of alumina sphere (10mm in diameter). Porcelain material (foam type) was used in preheat zone with 8ppcm.

Since the combustion performed here is premixed type, two gas mixers are used to ensure complete mixing action. Four K types of thermocouples (T1, T2, T3 and T4) are used to collect temperature data from burner housing. T1 is kept at 5mm above the surface of reaction zone to fetch surface flame temperature. While, T2 and T4 are placed at mid of reaction and preheat zone respectively. Finally, T3 is inserted at the junction between preheat and reaction zone. To receive information form thermocouple DAQ (data acquisition system) was utilized. Butane was used as fuel along with compressed air. Fuel mixture was fed from bottom the burner for stable combustion. Flow of air and butane was measured in L/min while emissions were continuously captures using KANE-9106 QUINTOX combustion analyzer in terms of ppm (parts per million).

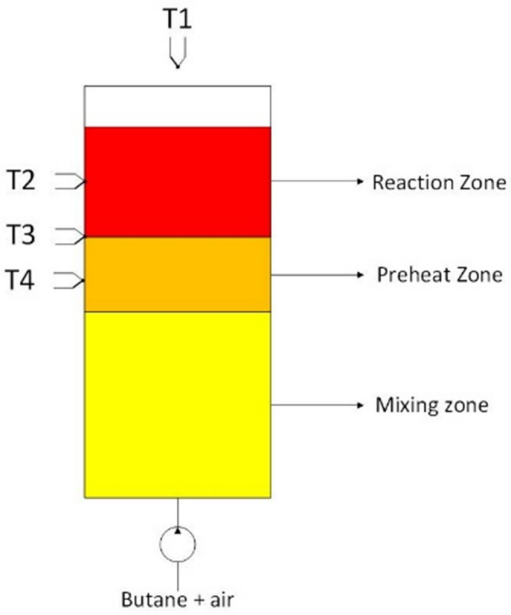

Fig. 1. Sectional view of burner housing

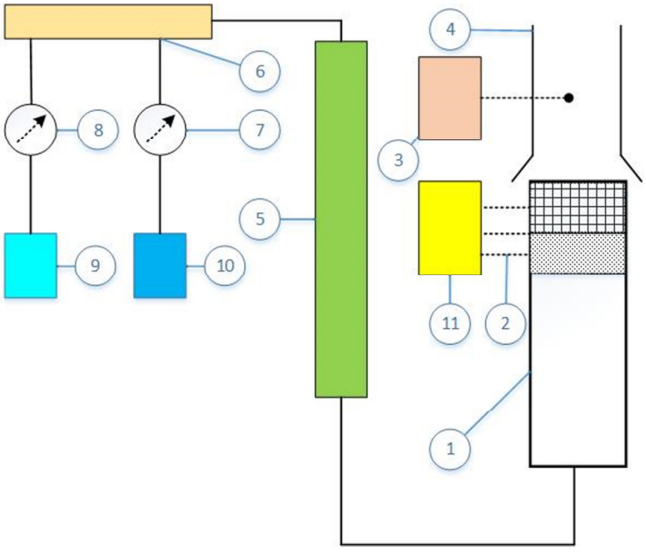

Fig. 2. Layout of combustion system. (1) Housing, (2) Thermocouple, (3) Emission gas analyzer, (4) Gas conduit, (5) Mixing unit, (6) Pre-mix, (7) Air flow meter, (8) Butane flow meter, (9) Butane supply, (10) Air inject, (11) DAQ (Data acquisition system)

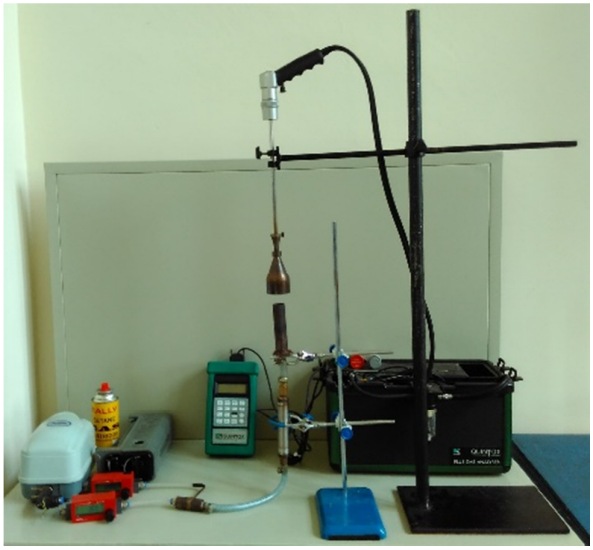

Fig. 3. Experimental setup

B. Mathematical model

In the present work, numerical simulations are performed using ANSYS 18.0. Single step reaction mechanism was carried out to predict the surface flame temperature with 2D modelling of the burner section was made with 10140 cells. The standard k-ε model with species transport equations was enabled during simulations. Furthermore, Outlet was kept at default pressure outlet, pressure was selected as atmospheric, with wall indicating stationary without any slip. Equation (1) is solved by CFD software as given below

$$\nabla.[\vec{v}(\rho E + p)] = \nabla.\left[k_{eff}\nabla T - \sum_j h_j \vec{J}_j + (\bar{\bar{\tau}}_{eff}.\vec{v})\right] + S_h \quad (1)$$

where, $\vec{v}$ = overall velocity vector, ρ = density, E=totoal energy, k_{eff} = effective conductivity, h_j = species enthalpy, $\vec{J}_j$ = diffusion flux of species j. On the right-hand side of equation (1), the equations represent energy transfer due to conduction, species diffusion and viscous dissipation, where $S_h$is the heat of chemical reaction [12]. Equation (2) was used with modification to the conduction flux.

$$\nabla.\left[\vec{v}(\rho_f E_f + p)\right] = \nabla.\left[k_{eff,pm}\nabla T - (\sum_i h_i J_i) + (\bar{\bar{\tau}}.\vec{v})\right] + S_f^h \quad (2)$$

where, E_f= total fluid energy, $k_{eff,pm}$= effective thermal conductivity of the medium and S_f^h= fluid enthalpy source term. Equation (3) and (4) indicate continuity and momentum equations respectively. Equation (5) indicates momentum equation for porous medium. In addition, PM was modelled by the addition of a momentum source term.

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (3)$$

$$u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(v + v_T)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_i}\right)\right] + \frac{f_j}{\rho} \quad (4)$$

$$S_i = \left(\sum_{j=1}^{2} D_{ij\mu v_j} + \sum_{j=1}^{2} C_{ij}\frac{1}{2}\rho|v|v_j\right) \quad (5)$$

where, S_i is the source term for the ith (x or y) momentum equation, $|v|$is the magnitude of the velocity and D and C are prescribed matrices [12].

3. Results and Discussion

Equation (6) indicates expression used to calculate ER. Where in actual air–fuel ratio (AF_a) and while stoichiometric air fuel ratio (AF_s) for butane gas was considered. Value of stoichiometric air fuel ratio for butane at room temperature was found out to be 30.95 [11, 12]. Actual supply of air is as shown in Table 1. In addition, the energy input is denoted as Q_{in}, whereas energy liberated from combustion can be represented as Q_{out}. Other terms like volumetric flowrate (V_{flow}), density (ρ), mass flow rate (M_f), calorific value (C_v), quantity of heat taken away by water (M_w) was the mass of water (kg), and M_c was the mass of container (kg). C_w and C_p were the specific heat of water and the container, respectively. The final temperature was 50 °C from the initial temperature (T_{in}) with time (t') in seconds. Equations (7-10) indicate correlation among these parameters.

$$\mathrm{ER} = \frac{AF_s}{AF_a} \quad (6)$$

$$M_f = V_{flow}\,\rho \quad (7)$$

$$Q_{in} = M_f C_v \quad (8)$$

$$Q_{out} = \frac{[(M_w C_w + M_c C_p)(50^{\circ} - T_{in})]}{t'} \quad (9)$$

$$n_{th} = \frac{Q_{out}}{Q_{in}} \quad (10)$$

Table 1
Calculated value of Er

Butane (Lpm)	0.1				
Air (Lpm)	3.1	3.4	3.8	4.4	5.1
ER	1.0	0.9	0.8	0.7	0.6

Initially burner was made run with low fuel input from 0.2 to 2 Lpm in an increments of 0.01 Lpm. Trial and error method indicate the minimum amount of fuel to generate stable flame was 0.1Lpm. Surrounding temperature was kept at 32^{o}C. Once the minimum amount of fuel was calculated based on flame stability, corresponding temperature profiles were generated at various ER. Output from DAQ helps to plot the Fig. 4 and 5. Burner can withstand only five ER based on experimental trails. Initially from ER = 1 to 0.9, burner was in surface flame mode and at ER = 0.6 submerged mode was seen clearly. Optimum performance was seen at ER=0.7 reaching surface temperature up to 631^{o}C. While submerged flame can able to generate maximum surface temperature of 470^{o}C with an average wall temperature of 170^{o}C. Since combustion is complex phenomenon predication of actual reasons for such abrupt behavior becomes blurring. The first possible reason is due to supply of incoming air, in addition installed PM do play very important role since surface area directly proportional to PM specification. Moreover, incoming fuel mixing technique does affect the temperature profiles.

Actual pictures of both surface and submerged can be seen in Fig. 6. While with the help of thermal imager, a top view of burner was taken as shown in Fig. 7. With the help of thermal image, temperature variation on the burner surface can be better understood. Thermal image indicate temperature >620^{o}C since that is upper limit to the device. Maximum thermal efficiency was calculated during surface flame was recorded as 84%. Finally, with the help of certified gas analyzer, emitted emissions from the burner surface where fed to gas probe to measure NOx and CO. It was found out that emission where less than 1ppm since the minimum amount to detect the ppm for the utilized gas analyzer is 1pmm. A 2D simulations using ANSYS 18.0 was performed which predicted surface temperature around 689^{o}C as shown in Fig. 8. Which is 5mm above the reaction layer, which can be considered to be acceptable limits, since error is only 9%.

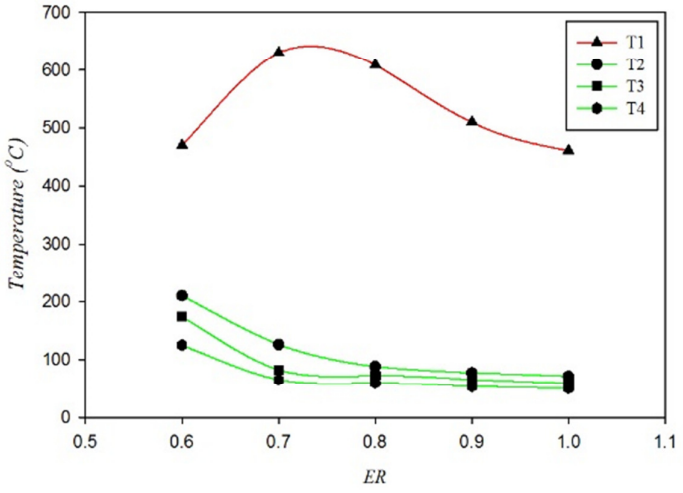

Fig. 4. Surface flame temperature along with ER

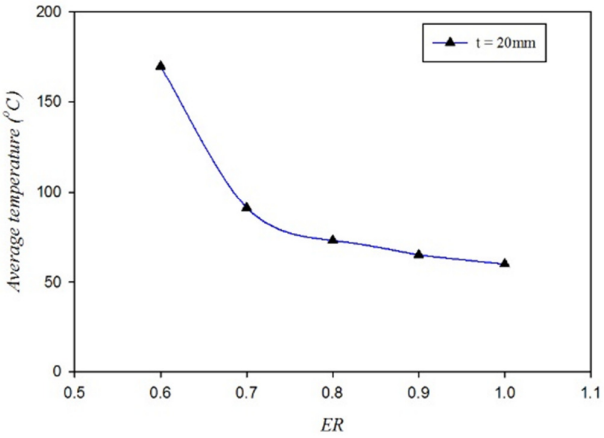

Fig. 5. Average temperatures along with ER

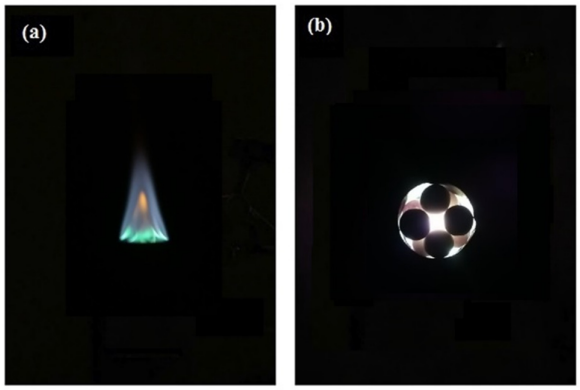

Fig. 6. Actual photo (a) Surface flame and (b) Submerged flame

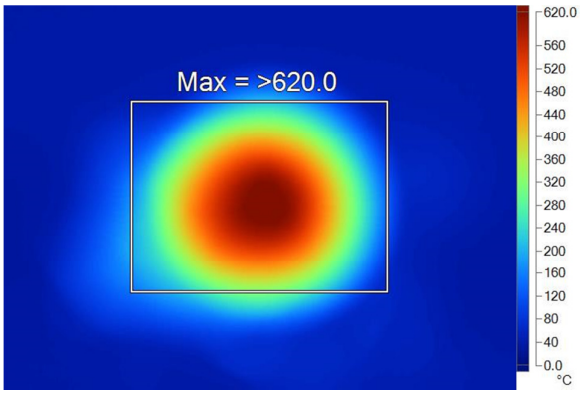

Fig. 7. Thermal image (Top view)

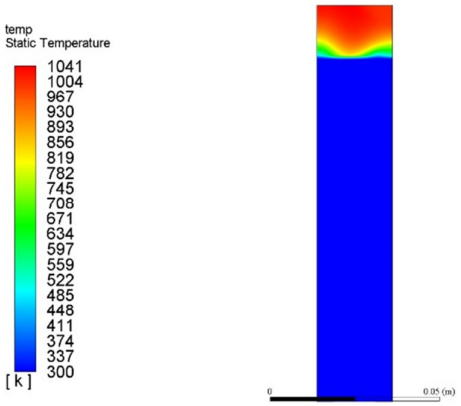

Fig. 8. Surface flame temperature distribution

4. Conclusion

Presented work deals with prediction of optimum equivalences ratio during surface and submerged flame conditions which comes out to be 0.7 and 0.6 respectively. Maximum surface flame temperature that can be archived was noted to be 631^{o}C, while during submerged flame was at 470^{o}C. Maximum thermal efficiency of the burner is 84%. Numerical simulation predicted surface flame temperature under acceptable value of 9% with respect to experimental data. Thermal images confirmed stable combustion at the center of the burner then gradually spreading across periphery. NOx and CO are under controlled limits.

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