

The Pack-Bed Sphere Liquid Porous Burner

B. Krittacom, P. Amatachaya, W. Srimuang and K. Inla

Abstract—The combustion of liquid fuel in the porous burner (PB) was experimented to investigate evaporation mechanism and combustion behavior. The diesel oil was used as fuel and the pebbles carefully chosen in the same size like the solid sphere homogeneously was adopted as the porous media. Two structures of the liquid porous burner, i.e. the PB without and with installation of porous emitter (PE), were performed. PE was installed by lower than PB with distance of 20 cm. The pebbles having porosity (ϕ) of 0.45 and 0.52 were, respectively, used in PB and PE. The fuel was supplied dropwise from the top through the PB and the combustion was occurred between PB and PE. Axial profiles of temperature along the burner length were measured to clarify the evaporation and combustion phenomena. The pollutant emission characteristics were monitored at the burner exit. From the experiment, it was found that the temperature profiles of both structures decreased with the three ways swirling air flows (Q_A) increasing. On the other hand, the temperature profiles increased with fuel heat input (Q_F). Obviously, the profile of the porous burner installed with PE was higher than that of the porous burner without PE.

Keywords—Liquid fuel, Porous burner, Temperature profile.

I. INTRODUCTION

COMBUSTION of liquid fuels is commonly conducted using a spray burner for industrial applications such as furnaces and boilers. The famous research of the combustion in porous media using liquid fuel oil was begun experimentally by Kaplan and Hall [1]. The stable and complete combustion of their experiment can be achieved by using a pressure atomizer for spraying heptane fuel into a cylindrical porous ceramic burner. Tseng and Howell [2] investigated the theoretical and experimental studies of sprayed liquid heptanes combustion in the porous inert media (PIM). The experimental setup was constructed similar to those of Kaplan and Hall [2], and the numerical model was performed in a one-dimensional laminar

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flow. They reported that the stable combustion was achieved as lean as equivalence ratio equal 0.3, and the comparison results between numerical and experimental results showed good agreement. Takami et al. [3] operated experimentally to find out the combustion of kerosene oil on the ceramic plate in which the fuel was supplied dropwise, instead of droplet spray, to the top surface of a horizontal porous ceramic plate burner. They revealed that the porous ceramic plate was effective for preheating of vaporized kerosene and stabilization of the flame on the surface of the porous ceramics. Martynenko et al [4] proposed the mathematical model of self-sustaining combustion of the gaseous mixture with simultaneous evaporation of fuel droplet in the porous medium by considering the phase change under the complex heat transfer including convective, conductive and radiative heat transfer. Jugjai and co-worker [5]-[6] studied experimentally based on the work of Takami et al. [3]. They presented more detail of the heat transfer phenomena, the evaporation mechanism inside the porous burner, and the combustion characteristics with the effects of the swirling air supply and effect of the packed bed emitter installed downstream of the burner system. Park and Kaviany [7] examined the model of a regenerative diesel engine using an in-cylinder reciprocating, porous regenerator. They found that thermal efficiency of this model approached 53 percent, and was higher than the conventional diesel engine (43 percent). Fuse et al. [8] performed experimentally to substantiate the concept of combustion self-sustained by kerosene vaporization enhanced by radiation through a higher Al_2O_3 porous ceramic plate, for the development of the practical work in electric power saving of the boiler for home use. However, these previous works did not use a lower porous media as PIM burner directly, particular the packed-bed sphere shape.

Recently, the authors [9] have adopted the pebbles as the pack-bed sphere porous burner to observe the combustion of diesel oil. We found that the complete combustion could be carried out. However, the phenomenon of evaporation was not very good owing to the pebbles only absorb radiant energy from flame. Therefore, the present work aimed to extensively investigate the improvement of evaporation in PB. A packed-bed sphere porous emitter (PE) was installed downstream of burner. Moreover, the merit of this result is to prepare the fundamental data to a practical work such as the diesel engine of automobiles, industrial furnace, and industrial boiler.

II. EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows the schematic diagram of the experimental apparatus. The burner consists of 4 sections: a fuel injection chamber, porous burner (PB), mixing & combustion and porous

emitter (PE). The experimental setup is constructed similar to those of Pipatana and Krittacom [9] but developed by combining the downstream porous emitter (PE). The cylindrical shape of the present burner is based on a 104-mm inner diameter stainless steel pipe with the length of 100 mm, 100 mm, 180 mm and 120 mm for the fuel injection, PB, mixing & combustion and PE section. Both porous sections are made of the pebble having porosity of 0.45 for PB and of 0.52 for PE. The pebble was carefully chosen in the same size like the solid sphere homogeneously.

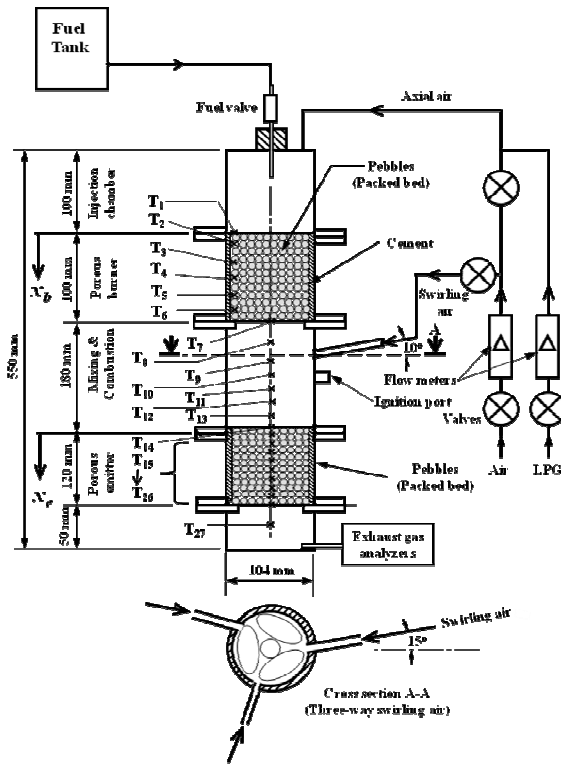


Fig.1 Schematic diagram of the experimental apparatus

Two average diameters of the pebbles d_p used in examinations were 5 mm and 15 mm, and thus the apparent porosity ϕ were 0.45 and 0.52 respectively. Optical thickness of packed bed τ_0 was given by geometrical length x as follows;

$$\tau_0 = x\beta, \quad (1)$$

where β is an extinction coefficients (m^{-1}) that can be evaluated by Kamiuto and Yee [10]:

$$\beta = (2\gamma - 1) \left(\frac{\pi d_p^2 n_p}{4} \right) = \frac{1.5(2\gamma - 1)(1 - \phi)}{d_p}, \quad (2)$$

$$\gamma = 1 + 1.5(1 - \phi) - 0.75(1 - \phi)^2. \quad (3)$$

Here, d_p , n_p , γ and ϕ denote the diameters of a packed sphere

(m), the number density of a packed sphere and the extinction enhancement factor, respectively. The porous thickness of PB and PE were 0.1 m and 0.12 m. Thus, the optical thicknesses of the present packed-bed sphere were 36.24 for PB and 12.06 for PE.

The combustion characteristics were determined from temperature profiles along the burner axis and the composition of the gases at the exit of the burner. To gain the profile of temperature, 27 N-type thermocouples of 0.5 mm diameter were installed in the burner and rearranged into 3 groups. The first group, N-type thermocouples of T_1 to T_7 , was installed axially at the circumferential surface of the injection chamber (inserted through the cement insulator). This arrangement prevented the change of flow pattern of liquid phase from the disturbance of the protrusion of the thermocouples and to validly describe the evaporation mechanism. The second group, thermocouples of T_8 to T_{14} , was used to measure the flame temperature along the centerline in the mixing & combustion chamber in order to determine the combustion characteristic. Finally, 10 thermocouples were inserted in the PE section to observe the phenomena of energy absorption and thermal feedback. The thermocouple signals were digitized by a general-purpose data logger then interpreted to a personal computer. The Testo M350 model was used to monitor the emission of the dry sampling gas products at the chamber exit. The concentrations of NO_x and CO were especially tuned to electrochemical sensors to ensure long-time stability and accuracy of the measurement, which were corrected to 0% excess oxygen on dry basis.

The procedure of the experimental operation of the present burner is similar to those of Pipatana and Krittacom [9]. The burner was first preheated with a premixed LPG-Air fed by axial flow whereas the swirling air was closed. The combustion was occurred by inserting a pilot flame through an ignition port. The diesel droplet and the swirling air were then supplied simultaneously with an appropriate condition for combustion. If the diesel flame was achieved, the LPG-Air and the axial air were then turned off. After obtaining stable combustion, the concentrations of NO_x and CO, the flow rate of swirling air (Q_A), the fuel heat input (Q_F) and temperature along the burner length were observed.

III. RESULTS AND DISCUSSION

A. Effects of three swirling air flow, Q_A

Figure 2 illustrates the effects of three swirling air flow Q_A on the temperature distribution along the burner length at a constant fuel load input $Q_F = 2.51$ kW for the case of without PE. With decreasing Q_A from 2.82 m/s to 8.47 m/s, the temperature profiles inside of PB are slightly increased. The trend of profile in the mixing & combustion chamber should be discussed as two important points. The first point is the peak temperature is appeared in position of T_8 above the position of swirling air supply owing to the stable flame is placed in this region. For the second point, the temperature

profiles below the position of swirling air supply are increased with Q_A decreasing. This profile is described by the fact that when a higher air-flow rate of the mixing between the fuel vapor and swirling air is supplied, a stronger combustion of the diesel flame is achieved, leading to the flame movement toward the lower surface of PB. In addition, the flame length become further extends as Q_A decrease.

Figure 3 shows the effects of Q_A on the level of CO and NO_x emissions under the same condition of Fig. 2. As seen in Fig. 3, with Q_A increasing, the CO concentrations are decreased. It is clarified that the better combustion of diesel vapor oil is dominated by a higher air-flow rate of swirling air fed to the burner. Then, an excess air of the combustion known well as lean combustion is occurred. The level of NO_x emission is almost unchanged as Q_A increased and is not over 50 ppm. Commonly, the quantity of NO_x will be appeared by two ways. The first way is a peak temperature (T_8) and the high temperature of combustion is obtained for the second one. However, the trend of NO_x can deduce that it is slightly decrease as Q_A increasing because of the peak temperature (T_8) increase with Q_A .

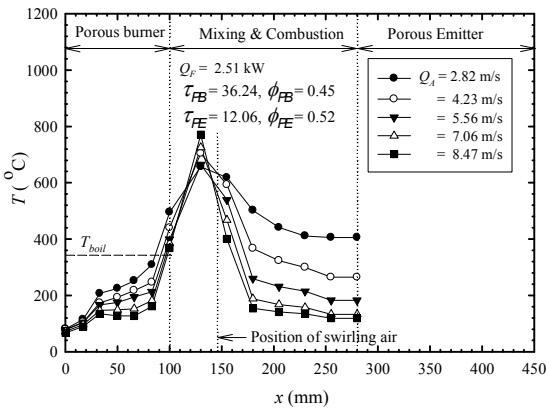


Fig. 2 Effect of Q_A on temperature profile along the burner length for without PE

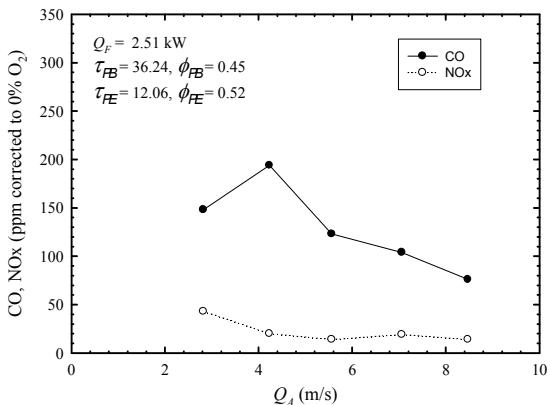


Fig. 3 Effect of Q_A on the level of CO and NOx emission for without PE

Figure 4 also illustrates the effects of Q_A on the temperature

distribution along the burner length at $Q_F = 2.51$ kW for the case of installation PE. The temperature profiles inside of PB are unchanged but can be said that they are increase slightly with decreasing Q_A . The temperature profiles inside of PE are evidently increased. The trend of profiles in the mixing & combustion chamber should be discussed as two regions separately: above and below the position of swirling air. For the above region, the peak temperature (T_8) is increased with Q_A . On the other hand, the temperature profile is evidently increased as Q_A increasing corresponding to the temperature in PE section. This profile is again described by the effect of a higher air-flow rate of swirling air or an excess air of the combustion.

Figures 5 show effects of the Q_A on the level of CO and NO_x emissions under the same condition of Fig. 4. As seen in Fig. 5, the CO concentrations are slightly increased as increasing Q_A which is in the range of 150 to 240 ppm. This is again clarified by effect of an excess air and much energy feedback from PE of a lower Q_A : the temperature profile in Fig. 4 is evidence. The level of NO_x emission is occurred in the range of 20 to 100 ppm that it is almost unchanged as Q_A increased. This trend is explained by a peak temperature (T_8) and the low temperature of combustion is not befallen.

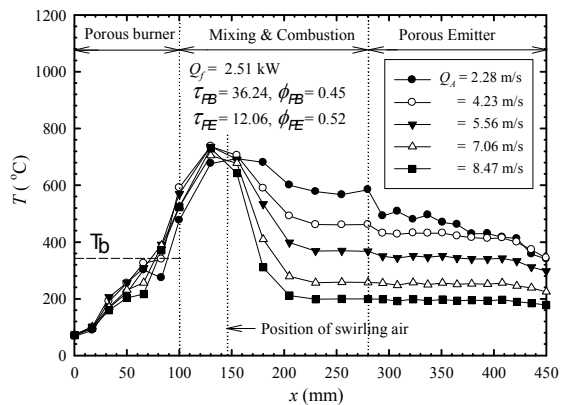


Fig. 4 Effect of Q_A on temperature profile along the burner length for installation PE

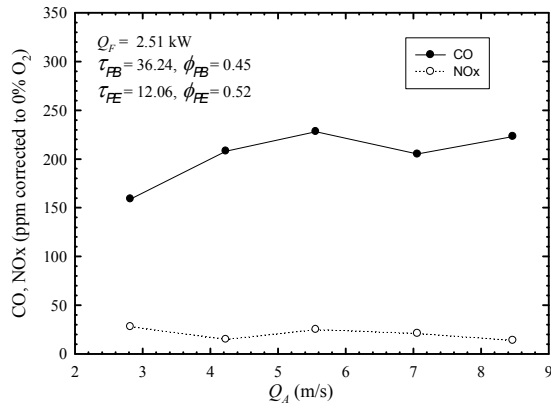


Fig. 5 Effect of Q_A on the level of CO and NOx emission for installation PE

B. Effects of Fuel Load Input, Q_F

Figure 6 illustrates the effects of heat input Q_F on the temperature distribution along the burner length at a constant $Q_A = 5.65$ m/s for the case of without PE. Varying fuel load input Q_F is achieved by varying droplet diesel input as Q_A is kept constant. For increasing Q_F from 0.71 kW to 4.42 kW, the temperature profiles are increased throughout the burner system. As seen in PB section, the temperature at lower surface of PB (T_7) is slightly higher than boiling temperature (T_{boil}) of diesel fuel [11] ($T_{boil} = 320$ C^o). It can be said that the evaporation is completed before the fuel passed through the PB. From this phenomenon, auto-ignition may be enabled by this fuel vapor, and thus the flame position is stabilized between the porous burner and the swirling air, which is presented by the maximum temperature (T_8).

Figure 7 shows the effects of Q_F on the level of CO and NO_x emissions under the same condition of Fig. 6. The CO concentration is increased with Q_F and is approached to 250 ppm. Obviously, a rapid increase in CO emission was established. The reason for this result is that the incomplete combustion of diesel vapor strongly depended on Q_F . This characteristic can be confirmed by a low level of NO_x concentration due to a lower maximum temperature of the combustion. Moreover, the level of NO_x emission can be deduced that is slightly increased with Q_F because of an increased temperature of the combustion.

Figure 8 illustrates the effects of heat input Q_F on the temperature distribution along the burner length at a constant $Q_A = 5.65$ m/s for the case of installation PE. The temperature profiles are increased throughout the burner system with Q_F . As seen in PB section, the PB can be absorbed the radiant energy feedback from PE leading to the temperature at T_7 is higher than T_{boil} of diesel fuel. From this behavior, the evaporation is completed before the fuel passed through PB. However, a rapid increase in CO emission is established as shown in Fig. 9. It is elucidated by the incomplete combustion of the rich mixture between diesel vapor and swirling air. The level of NO_x is relatively low because of no peak temperature and a good temperature distribution along the burner length.

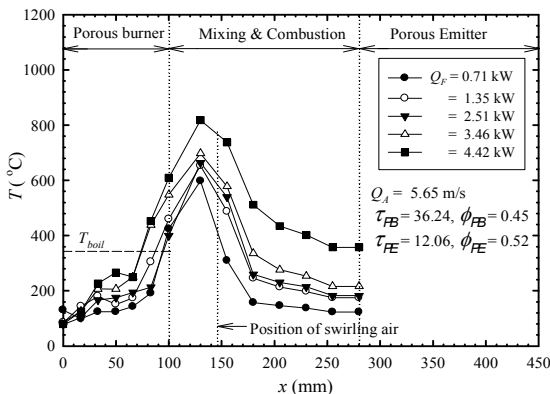


Fig. 6 Effect of Q_F on temperature profile along the burner length for without PE

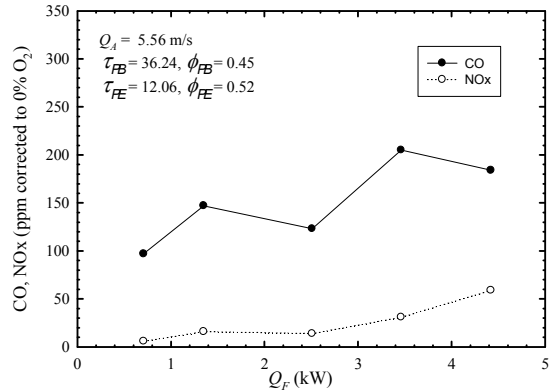


Fig. 7 Effect of Q_F on the level of CO and NOx emission the burner length for without PE

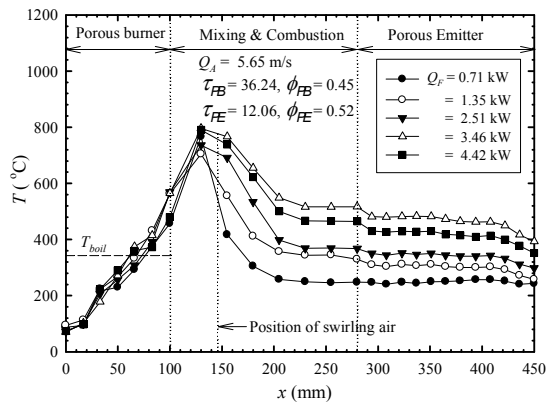


Fig. 8 Effect of Q_F on temperature profile along the burner length for installation PE

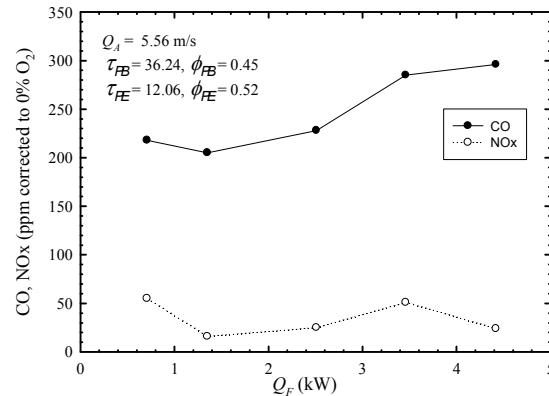


Fig. 9 Effect of Q_F on the level of CO and NOx emission the burner length for installation PE

IV. CONCLUSION

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

- 1) The profiles of temperature inside the porous burner section (PB) in both structures (with or without PE) are decreased as Q_A increasing, but increased with Q_F .

2) The profiles of temperature inside the porous emitter section (PE) in both structures depend on Q_F but the temperature profiles inside PE are decreased as Q_A increasing.

3) A peak temperature is located between the position of swirling air and PB section; that is T_8 . For the burner installed PE, the temperature at the lower surface of porous burner (T_7) is entirely higher than the boiling temperature of diesel fuel (T_{boil}): it is confirmed that the evaporation is completed before the fuel passed through PB.

4) The level of CO and NO_x emission observed under the operating range are relatively low and the CO concentration is strongly dependent on the fuel load input Q_F .

5) The evaporation of a diesel liquid fuel is possibly completed by the preheating in packed bed (PB) because the effect of the thermal radiation feedback from the PE. Therefore, the packed sphere porous medium can favorably serve as the liquid fuel distributor, the thermal radiation absorber-emitter, the liquid fuel evaporator and the fuel vapor preheated.

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