Parametric Studies of Thermal Efficiency in a Proposed Porous Radiant Recirculated Burner (PRRB): A Design Concept for the Future Burner

Sumrerng Jugjai and Surachai Sanitjai
Department of Mechanical Engineering
King Mongkut's Institute of Technology Thonburi Bangkok 10140
THAILAND

ABSTRACT

The present study proposes a new design for a gas burner in order to improve the thermal efficiency of conventional open-flame atmospheric burners which are now widely used in most domestic appliances. The study proposes an efficient means for improving the thermal efficiency of the burners by utilizing a porous medium technology. An internal heat recirculation system was constructed in order to capture and recirculate some of the enthalpy of the exhaust gas. A porous medium of an appropriate optical thickness was used, based on the principle that the porous medium will convert the enthalpy of the exhaust gases to thermal radiation which is then fed back to the burner inlet. The experimental results show that the porous medium was very effective in increasing the thermal efficiency due to the efficiency of the internal heat recirculation taking place in the burner. The first-law thermal efficiency of the proposed porous radiant recirculated burner (PRRB), is on an average approximately 10 % higher than those of the standard burner (SB) and the porous radiant burner (PRB).

1. INTRODUCTION

It is well known that the thermal efficiency of conventional open-flame atmospheric burners (SB) for domestic appliances markedly depends on the size of the vessel and the quantity of the substance inside the vessel which has to undergo a thermal process at the burners. The thermal efficiency of standard burners increases with the size of the vessel and the volume of the substance in the vessel. However, these are considered as external factors since an increase in the thermal efficiency of the burners does not arise from the introduction of a specific means of heat transfer enhancement and of combustion augmentation technique to the design of the burners. In order to improve the thermal efficiency of a standard burner, an efficient heat transfer enhancement technique from flame to the load as well as an efficient combustion augmentation method is needed.

In standard burners (SB) for most domestic appliances, most of the energy supplied in the fuel is transferred to the thermal load by convective heat transfer, the rate of which is limited by the flame temperature and the flow characteristics of the flame over the heat transfer surface (normally laminar flow). Therefore, improvement in thermal efficiency of the burners may be brought about by change of type of fuel and flow characteristics which is actually difficult to do in view of the available fuels for domestic use. Recently, Sathe et al. [1] performed some detailed studies of the heat transfer characteristics of a porous radiant burner (PRB) and stated that the concept of the PRB is a promising development in gas-burner technology. The PRB operates by stabilizing a premixed flame inside or near to the surface of a non-combustible porous medium. The enthalpy of combustion released in the gas phase heats the porous matrix which then emits thermal radiation to a heat load. Unlike the SB, heat is transferred to the load, not only by convection but also by thermal radiation in the PRB. The output radiative efficiency (ratio of radiation output to heat input) is generally about 15% - 20% depending on the burner characteristics such as flame location, optical thickness and emissivity of the porous medium. In small-scale applications, the PRB has already shown a performance gain over the standard burner SB measured by higher efficiencies, lower NO_x emissions and more uniform heating. However, both the SB and the PRB are lacking in combustion augmentation in order to increase the flame temperature. This could be done by several means such as heat recirculation from the product gas to the reactant gas, the oxygen enrichment and/or better fuel quality by mixing in a high calorific fuel, combustion augmentation by catalytic reactions and by plasma jets. Among these techniques, heat recirculation is the most promising since it raises the combustion temperature above the adiabatic flame temperature without any additional energy. Weinberg [2], who pioneered heat-recirculation burners, stated that there is no ceiling on the maximum temperature. It depends on the amount of heat recoverable, which in turn depends on the type of heat exchanger used. A conventional convective heat exchanger may not be suitable due to its relatively large size when compared with the burner size. Therefore, special purpose heat exchanger design is urgently needed, based on state-of-the-art technology.

This study proposes a new design of atmospheric burner called a porous radiant recirculated burner (PRRB), in which there is a heat feed back process transferring some gas enthalpy to the premixed mixture by means of a porous medium technology. Echigo [3] discovered that a porous medium with an optimum optical thickness can effectively convert the enthalpy of the hot gas into thermal radiation or, indeed, vice- versa. This is because the porous medium is acting as an emitter or an absorber due to its high surface area to volume ratio and its greater emissivity when compared with gas. By combining these two functions, a compact heat exchanger can be fabricated and an internal heat recirculation from the exhaust gas to the mixture of fuel and air is expected to be established in the burner. The main purpose of the study is to evaluate the first-law thermal efficiency of the proposed burner (PRRB) which is equipped with a porous medium. The performance of the burner is evaluated by comparing the thermal efficiency and emission characteristics with those of the SB and the PRB.

2. THEORETICAL BACKGROUND OF ENERGY CONVERSION BY POROUS MEDIUM

Fig. 1 shows the energy conversion process from gas enthalpy to thermal radiation using the porous medium which was pioneered by Echigo [3]. Due to the large surface area to volume ratio of the porous medium and its higher emissivity compared to the gas, the interaction between the hot gas flowing through the porous medium and the solid phase results in a convective heat transfer from the gas to the solid. Under thermal equilibrium conditions, the solid phase will emit thermal radiation which is mainly directed towards the upstream direction ($F^-(0)$) of the flowing gas leading to a uniform heat flux and high temperature over the region. Based on this principle and using a porous medium technology, combustion augmentation was successfully observed with fuel mixtures of low heating value [4,5] and in an enhanced heat transfer to the cooling water pipe [6].

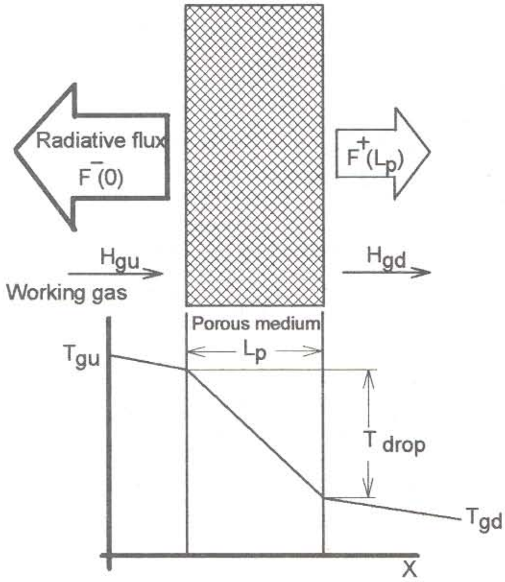

Fig. 1. Principle of the porous medium.

3. EXPERIMENTAL APPARATUS AND PROCEDURE

Three types of burners were used in the experiment. Fig. 2, Fig. 3 and Fig. 4, respectively, show the standard burner (SB), the porous radiant burner (PRB) and the proposed porous radiant recirculated burner (PRRB). The SB and the PRB were used in the experiment to compare the first-law thermal efficiency with the PRRB. The importance of the SB and the PRB hardly needs to be stressed since they are now widely used on most domestic appliances and have very many low-temperature industrial applications. The structure and function of the SB and the PRB are similar in that the formation of the combustible fuel/air mixture is by the entrainment of primary air arising from a momentum exchange between the gas jet and the surrounding air of the mixture tube. In most atmospheric burners, the supply pressure of the gas is normally about 625 Pa to 2000 Pa. The amount of air entrained for combustion is generally between 50% to 70% of the stoichiometric air requirement. However, it is feasible to entrain the whole of the stoichiometric air requirement in this way if the supply pressure of the gas is higher than 2000 Pa. In the experiment supply pressure of the gas (LPG containing C_3H_8 40 % and C_4H_{10} 60 % by vol.) was 14000 Pa. and thus a stoichiometric air entrainment was achieved. Fuel/air mixture is uniformly distributed to the flame port which in the case of the SB, is a circular brass ring with small slots around its circumference. The flame port in the case of the PRB is a ceramic porous medium. A vessel containing 1500 cm^3 of water was placed on the burner for measuring the thermal efficiency. A mercury thermometer was used to indicate the water temperature. A portable Bacharac gas analyzer with probe was inserted underneath the vessel to measure the exhaust gas compositions. This gas analyzer is capable of monitoring some important emissions such as NO_x, SO_2, CO, CO_2 and percent of excess O_2.

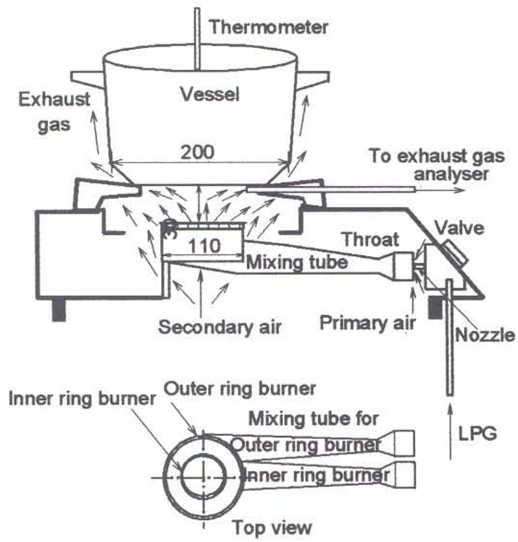

Fig. 2. Standard burner (SB).

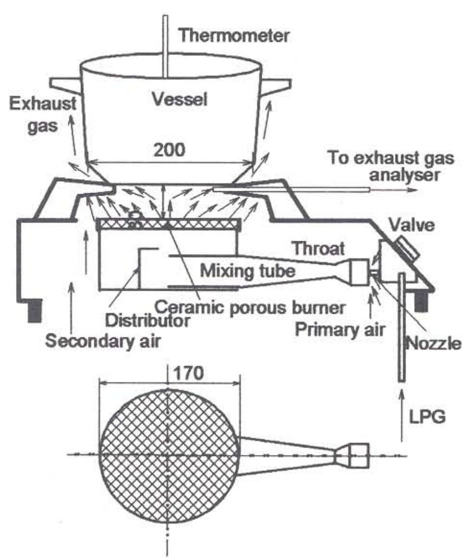

Fig. 3. Porous radiant burner (PRB).

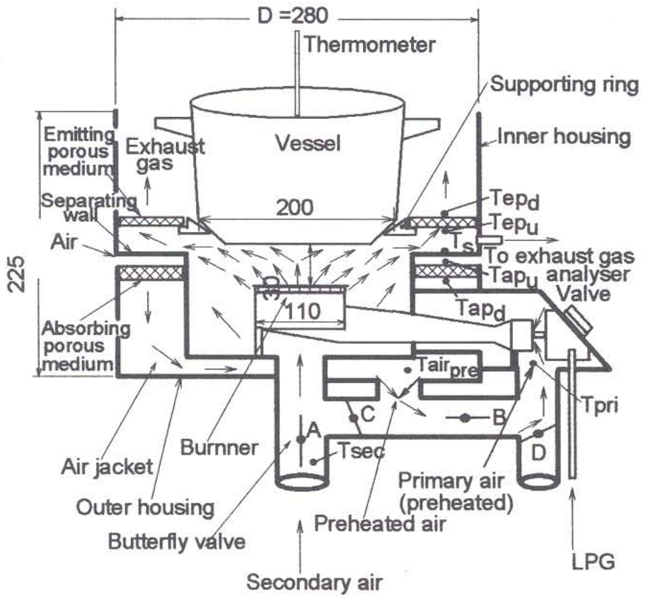

Fig. 4. Porous radiant recirculated burner (PRRB).

The proposed porous radiant recirculated burner (PRRB) is totally different in design concept from the SB and the PRB. It may be classified as a heat recirculating burner in which the fuel/air mixture is preheated by a specific method prior to combustion. The main feature of this study is to propose the application of a porous medium technology with product heat recovery. As shown in Fig. 4, the PRRB is first constructed by making use of a mixture tube identical to that of the SB. The heat feed back mechanism from product to reactant gases is obtained by suitably arranging the position of the porous medium (made of stainless steel wire mesh (40 meshes/inch) placed in layers) so as to allow operations according to the principle previously described. The PRRB is fabricated from stainless steel sheet. It has two major components, an inner housing and an outer housing. These components are cylindrical in shape and co-axially assembled. The gap between the housings forms an air jacket in which the combustion air will be preheated to temperature $Tair_{pre}$. Tap_u and Tap_d, respectively, denote upstream temperature and downstream temperature of the absorbing porous medium. The combustion air will be directed to the mixture tube (as a preheated primary air with temperature of T_{pri}) or to the bottom of the burner (as a preheated secondary air with temperature of T_{sec}) by controlling the position of the butterfly valves A, B, C and D. Combustion takes place at the flame ports of the burner and transfers heat to the vessel. The combustion gases are then directed to flow through the emitting porous medium by means of the splitting effect of the supporting ring. At the emitting porous medium, an energy conversion from enthalpy of product gas to thermal radiation takes place. This thermal radiation is then directed towards the absorbing porous medium via the separating wall which has a surface temperature T_s. Tep_d and Tep_u, respectively, represent the downstream temperature and upstream temperature of the emitting porous medium.

The first step of the experimental procedure for each burner was to warm up the burner by burning the fuel at a prescribed fuel flow rate without thermal load. Then an aluminum vessel containing 1500 cm^3 of tap water at ambient temperature T_{amb} of about 30 °C was placed on the burner. The water was allowed to boil and evaporate until about 1000 cm^3 of water was in the vessel. This period of time was recorded for computing the amount of fuel supplied. A typical first-law thermal efficiency η_{th} for each burner is the ratio of summation of the sensible heat and the latent heat to the amount of heat supplied as defined as

$$\text{First-law thermal efficiency, } \eta_{th} = (Sensible\ Heat + Latent\ Heat)/(Total\ Heat\ Supplied)$$

$$= \{(MCp)_W (100 - T_{amb.}) + (M_{Wevap.})\ (Latent\ heat)\} / (Fuel\ flow\ rate \times Heating\ value \times Burning\ time)$$

where M, M_{Wevap} and Cp_W represent, respectively, the initial mass of water, the mass of evaporated water and the mean specific heat of water. The experiment was conducted in order to compare the thermal efficiency η_{th} and emission characteristics of the three burners. For each burner and each power level, three repeats test were performed and the results were averaged to assure the reliability of the experiment. Parametric studies of the PRRB were conducted in order to obtain a better understanding of its characteristics. Parameters of the PRRB which are expected to be controlling factors for the improvement of the thermal efficiency η_{th} include the presence of a porous medium (with or without installation of the emitting porous medium), the optical thickness of the emitting porous medium and type of air preheated (primary or secondary air). Experiment were performed to vary these parameters.

4. RESULTS AND DISCUSSION

4.1 Typical Thermal Structure of the PRRB

Fig. 5 shows a typical thermal structure in terms of temperature change with the combustion load at various positions in the burner. In this case, the butterfly valves A, B, C and D are set in order to preheat the primary air prior to mixing with the LPG in the mixing tube. The optical thickness, used in this experiment, of the emitting porous medium (emitting PM) and of the absorbing porous medium (absorbing PM), respectively, are 0.5 and 1.5. As a first step, the outside surfaces of the burner were not covered with any insulator. Fig. 5 clearly shows a dramatic increase in $Tair_{pre}$ with combustion rate (Lc) which indicates an effective preheating of the primary air. $Tair_{pre}$ has a maximum value of about 210 °C at 4 kW and it gradually decreases as the Lc exceeds 4 kW. A quite large temperature drop across the emitting porous medium is shown by the difference between Tep_u and Tep_d. This means that an effective energy conversion from gas enthalpy to thermal radiation is occurring in the emitting porous medium and is based on the stated principle and careful suitable arrangement of the porous medium. This thermal radiation is responsible for the occurrence of internal heat recirculation from the exhaust gas to the combustion air leading to a significant increase in $Tair_{pre}$.

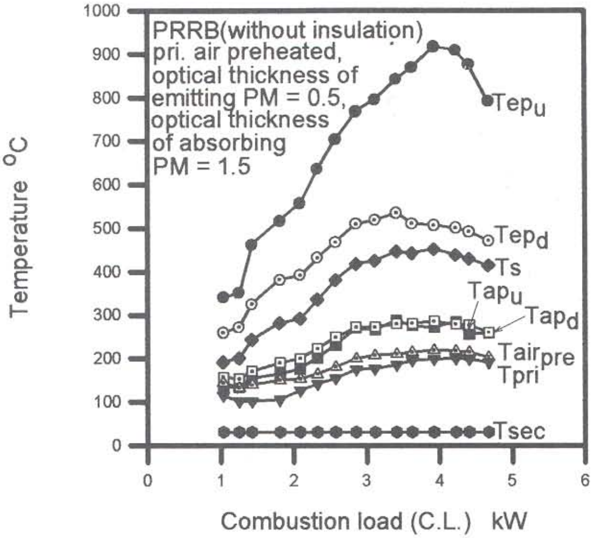

Fig. 5. Thermal structure of the PRRB.

4.2 Comparison of First-law Thermal Efficiency η_{th} and Emission Characteristics

Figs. 6 to 10 show a comparison of the first-law thermal efficiencies and emission characteristics for the three burners. The performance of the PRRB is superior to that of the SB and the PRB both in terms of thermal efficiency and combustion characteristics. The PRB offers a higher thermal efficiency than that of the SB at low Lc in particular. The relative value of thermal efficiency and Lc of the three burners show trends similar to each other. The thermal efficiencies of the three burners are relatively high at low Lc. However, the thermal efficiencies decrease as the Lc increases. This may be attributed to an increase in convective heat loss as the Lc increases. The contact time for heat transfer is reduced as the flow velocity increases.

There is an increase in CO at a certain combustion rate Lc for the SB and the PRRB, whereas the PRB produces nearly constant emissions of CO over the range of Lc examined in the experiment (Fig. 7). This may be due to the fact that the mixing process between the secondary air and the combustion flame in the PRB is not very good. Unlike the SB and the PRRB, the PRB has a solid cylindrical box supporting the ceramic porous burner and this does not allow the secondary air to flow through the burners or the flame. Thus, the mixing process takes place only at the circumferential edge of the flame. The jump increase in CO for the PRRB occured at Lc = 3.5 kW which is higher than that of the SB (Lc = 3 kW). This indicates that the PRRB provides a wider operating range with a low emission of CO.

The PRRB produces a sharp increase in NO_x as the combustion rate Lc increases (Fig. 8). This is because the PRRB operates at a higher flame temperature due to the internal heat recirculation from the enthalpy of the exhaust gas to the combustion air by means of thermal radiation. However, the

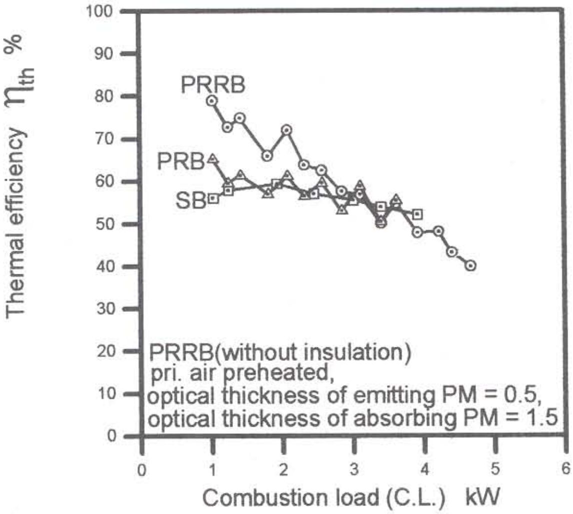

Fig. 6. Comparison of first-law thermal efficiencies η_{th} of the three burners.

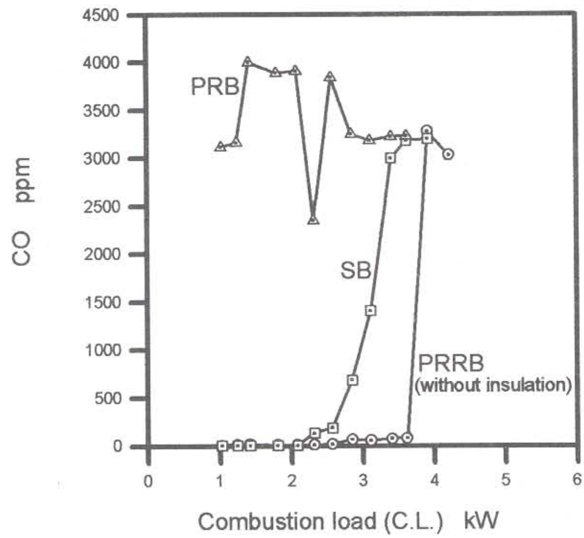

Fig. 7. Comparison of CO.

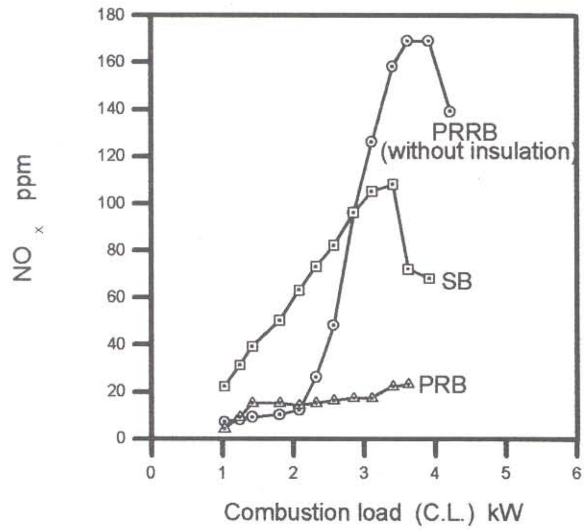

Fig. 8. Comparison of NO_x.

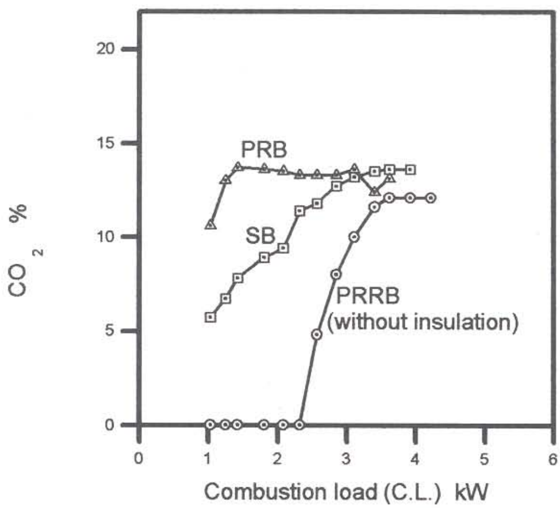

Fig. 9. Comparison of CO_2.

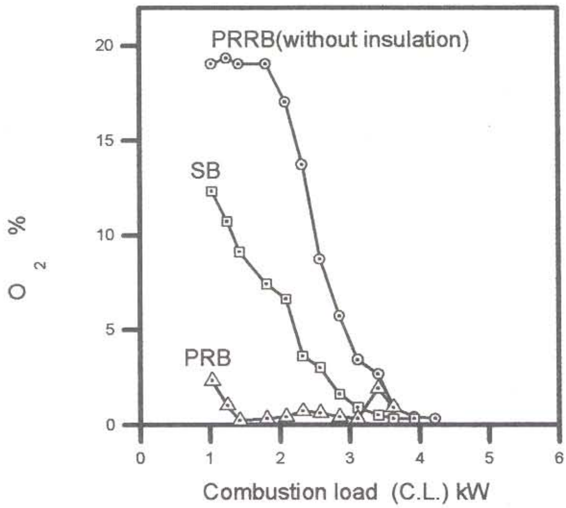

Fig. 10. Comparison of O_2.

maximum value of NO_x produced by the PRRB does not exceed the limits set by the EPA (1000 ppm). The PRB is notable for having the lowest NO_x emission amongst the three burners. This may be for two main reasons. First, the flame temperature is reduced by the interaction between the gas and the solid ceramic porous burner. A large proportion of the gas enthalpy is used to heat the ceramic from which thermal radiation energy is emitted to the thermal load (vessel). Second, in the PRB , excess oxygen is the lowest as shown in Fig. 9. Therefore, thermal NO_x is suppressed. This may be due to a limited air entrainment capability of the PRB when compared with the SB and the PRRB. Thus, of the three burners, only the PRB operates at a nearly stoichiometric combustion condition.

Fig. 10 shows comparison of CO_2 concentration for the three burners. It is obvious that the level of CO_2 up to about Lc = 2.4 kW for the PRRB is closed to zero. At first glance it seems that incomplete combustion may be occurring and a high combustion efficiency or a high thermal efficiency corresponding to this region for the PRRB may be impossible. However, the PRRB has produced very low concentration of CO up to about Lc = 3.6 kW as shown in Fig. 7. Meanwhile the corresponding O_2 concentration is relatively high of about 19 % as shown in Fig. 9. These mean that a very lean combustion situation is taking place in the region of the PRRB. This may be attributed to the improvement of the natural draft of the secondary air and an occurrence of an efficient energy recirculation in the PRRB.

4.3 Effect of Type of Preheated Air

Fig. 11 shows the effect of the type of preheated air on the thermal structure in terms of T_s, Tap_d and $Tair_{pre}$ within the PRRB. The experimental conditions of the PRRB were the same as those for

Fig. 5. Selection of type of the preheated air could be acheived by setting the position of the butterfly valves A, B, C and D. As can be seen in Fig. 11, preheating the primary air is more effective than preheating the secondary air in that the primary air reaches a higher temperature $Tair_{pre}$ than the secondary air. Preheating the primary air also extends the limit of good combustion stability and favorable emission characteristics, whereas preheating the secondary air results in a flame blow-off phenomena (when the combustion rate Lc is more than 3.6 kW). Furthermore when heating the secondary air a jump increase in CO occurred at smaller Lc when compared with preheating the primary air (Fig. 12). This may be for two reasons, firstly the volume of secondary air being preheated is small when compared with the volume of the primary air and secondly the mixing process between the secondary air and the LPG is poorer.

4.4 Effect of Optical Thickness

Figs. 13 and 14 show the effect of the optical thickness of the absorbing porous medium on the thermal structure of the PRRB. In this experiment, a ceramic fiber insulator is wrapped around the external surfaces of the PRRB in order to reduce heat loss. The optical thickness of the absorbing porous medium was changed from 1.5 to 3 by increasing the number of layers of the stainless steel wire mesh that form the porous medium. Increasing the optical thickness means increasing the surface area of heat transfer from the gas to the solid phase (wire mesh). Therefore, the preheating effect of the combustion air at the absorbing porous medium is expected to be increased. $Tair_{pre}$ at optical thickness of 3.0 is markedly higher than that of optical thickness of 1.5, eventhough T_s and Tap_d for the two optical thickness are not significantly different. A jump increase in CO of optical thickness of 3 took place at higher combustion rate than that of optical thickness of 1.5. NO_x emission for the optical thickness

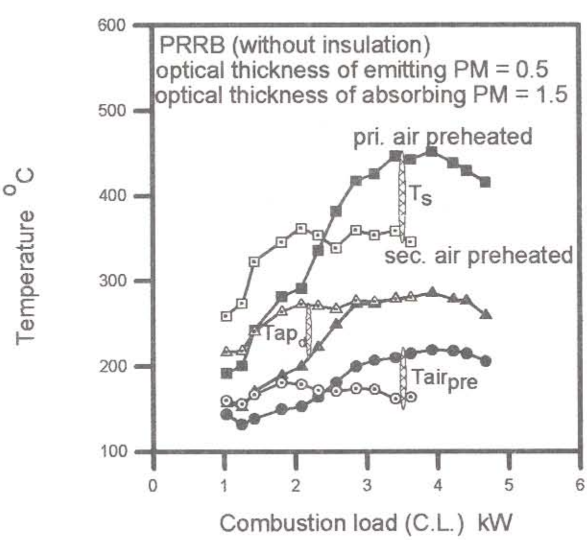

Fig. 11. Effect of preheated air on T_s, Tap_d and $Tair_{pre}$ of the PRRB.

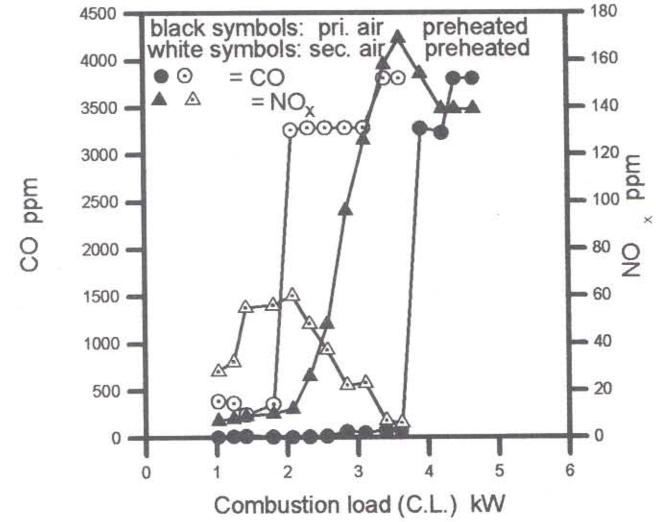

Fig. 12. Effect of preheated air on emission characteristics of the PRRB.

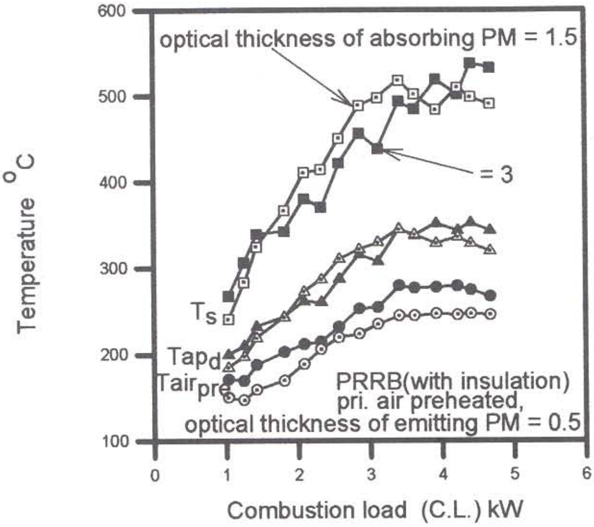

Fig. 13. Effect of optical thickness of the absorbing PM on Tap_d and $Tair_{pre}$ of the PRRB.

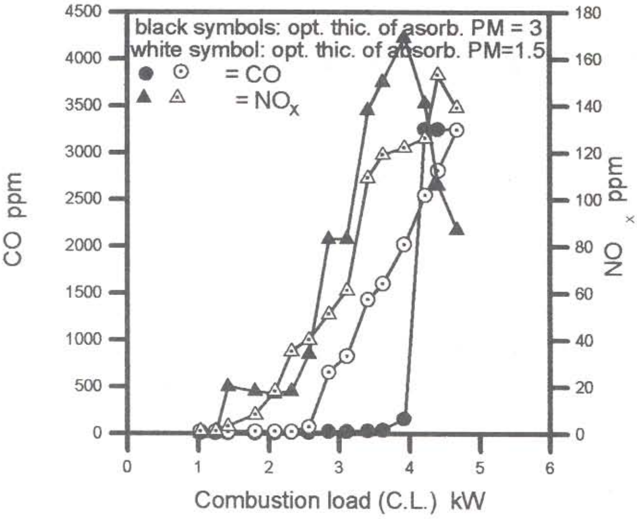

Fig. 14. Effect of optical thickness of the absorbing PM on the CO emission characteristics of the PRRB.

of 3 is relatively high due to a more efficient preheating effectiveness leading to higher combustion temperature.

4.5 Effect of Porous Medium

In order to clarify the effectiveness of the porous medium in generating an internal heat recirculation by thermal radiation from the exhaust gas to the combustion air, the emitting porous medium was dismantled, leaving the absorbing porous medium still in place. As was expected, without the emitting porous medium the preheating effectiveness is greatly reduced as shown in Fig. 15. Eventhough the emitting porous medium was excluded from the system, preheating is still effective in raising $Tair_{pre}$ up to 140 °C. This may be due to an increase in thermal inertia of the whole system and conduction of heat along the sheet metal to the combustion air. These experiments show that the emitting porous medium has very important role in enhancing the heat transfer from exhaust gas to the combustion air leading to a considerable increase in the various temperatures, and in $Tair_{pre}$ in particular.

Fig. 16 shows the entire picture in summary of the effect of the various parameters described above on the first-law thermal efficiency η_{th} of the PRRB. An effect of heat loss from the PRRB on the thermal efficiency was also conducted by wrapping around the PRRB by a ceramic fiber insulator. Performance of the SB and the PRB are also included for comparison. All of the thermal efficiencies are linearized for the sake of convenience in making comparison to avoid confusion due to scattering of the plotted symbols. The proposed PRRB is shown to be a promising technology for new domestic burners achieving a higher overall thermal efficiency than those of the other burners.

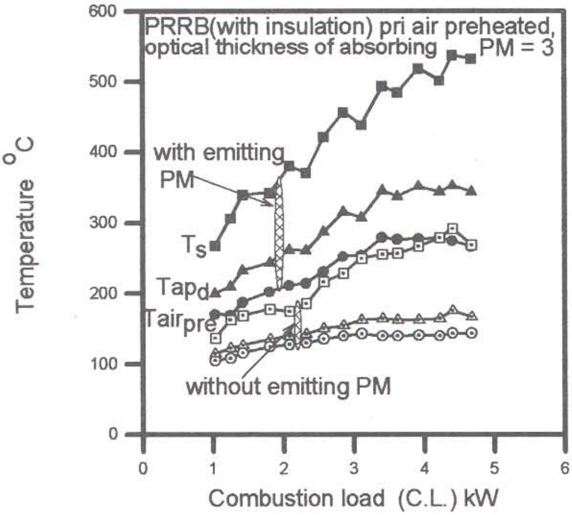

Fig. 15. Effect of optical thickness of the emitting PM on Tap_d and $Tair_{pre}$ of the PRRB.

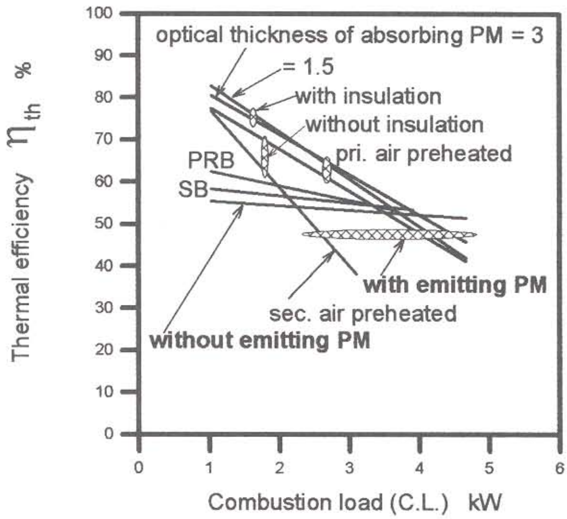

Fig. 16. Effect of various parameters on the thermal efficiencies η_{th} of the PRRB.

5. CONCLUSIONS

The porous radiant recirculated burner (PRRB) has been shown by experimental study to generate an efficient internal heat recirculation leading to significant improvement in the first-law thermal efficiency, an extension of combustion stability and good emission characteristics when compared with a standard burner (SB) and a porous radiant burner (PRB). Thus, the PRRB can be proposed as a worthy design concept for domestic burner.

6. REFERENCES

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