Liquid Fuels Porous Medium Burner

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ABSTRACT

Present observations of the progress in liquid fuels combustion technology strongly suggest that utilization of a porous medium burner is a promising approach for future applications. The porous medium burner for liquid fuels is more advantageous than the conventional open spray flame burner for several reasons. These include enhanced evaporation of droplet spray owing to regenerative combustion characteristics, low emission of pollutants, high combustion intensity with moderate turndown ratio and compactness. Existing designs of liquid fuel porous medium burners have typically relied on a spray atomizer with combustion flame contained within the porous medium. Against this background, a novel porous burner system was developed for burning kerosene without the need of using a spray atomizer. This burner system had an insulated porous medium burner section and a combustion chamber section, which is 80 mm in diameter. Kerosene was supplied dropwise to the top surface of the porous medium burner and burnt on the lower side where the swirling combustion air was supplied and mixed with the fuel vapor. Observations of vaporization mechanism and combustion characteristics occurring inside the burner system were investigated by measuring both axial and radial temperature profiles and emission characteristics. Interaction between the kerosene fuel and the porous medium burner was irradiated by thermal radiation from the downstream combustion zone enhancing the fuel vaporization. Stable combustion with low emission of pollutants was achieved at an equivalence ratio of 0.37-0.55 at thermal input of 2.62-3.49 kW. The effects of various parameters including equivalence ratio, thermal input and downstream installation of the porous emitter on the combustion characteristic were clarified.

1. INTRODUCTION

Research on combustion of hydrocarbon fuels within a porous inert media (PIM) burner has received attention from numerous researchers. Driven by the desire for higher radiant heating rates and the need to emit minimal pollutants, their interest in porous media combustion has been growing. It emphasizes combustion within the PIM, rather than combustion that occurs at the surface of the PIM. During the past decade, emphasis was placed on the 'gaseous fuel' combustion in the PIM, wherein the heat transfer phenomena and the combustion regime taking place were clarified. It has been well understood that the combustion of gaseous fuel within the PIM has many advantages over conventional open premixed flame burners.

Firstly, a porous ceramic burner can provide a regenerative combustion and contribution to complete reaction because of energy feedback by thermal radiation and conduction through the porous matrix. Heat from the burned downstream gases is recirculated to preheat the unburned mixture upstream of the reaction zone in the PIM by a combined mode of the heat transfer mechanism yielding the so-called 'excess enthalpy flame' [1], which has a peak temperature higher than the corresponding adiabatic flame temperature of the mixture. Thus, higher burning speeds and combustion intensity together with high radiant output were yielded, which allowed a more compact burner design.

Secondly, the ability to extend lean flammability limits because of the efficient internal energy recirculation means that the chemical reaction can occur at a relatively low temperature because of the presence of a large amount of air. Thus, a low emission of pollutants such as NO_X was yielded since its chemistry of production is very temperature dependent. CO and unburned hydrocarbon emission were also low due to efficient internal energy recirculation or fuel preheating effect.

Thirdly, there is an advantage due to the ability to maintain stable combustion during any minor fluctuations of the flame temperature or even to restart the combustion immediately by simply restoring the flow of the fuel and oxidant after flame extinction caused by momentary interruption of the fuel or oxidant flow or other reasons. The reason is that the specific heat of most PIM materials is sufficient to maintain its temperature above the ignition temperature for some time after the flame has been extinguished. Details of the combustion of gaseous fuels within the PIM and its practical application can be found in various sources, where the topic of this area has been well-researched [2].

Until recently, the concept of combustion of hydrocarbon fuels within the PIM has been applied to burn 'liquid fuels'. Typical applications of this technology are quite new and have had a strong impact on industrial application. These may range from stationary incinerators for liquid hazardous wastes [3-6] to a new version of a diesel engine used in automobile [7]. Kaplan et al. [3] achieved stable and complete combustion of liquid heptane spray within a cylindrical porous ceramic burner. Heptane was impinged on the combustion section using a pressure atomizer with a fixed flow rate of approximately 0.025 lpm and with the fuel droplet diameters in the range of 50 to 100 μm. The burner had an insulated combustion section that was 0.01 m in diameter and consisted of several 0.025 m thick ceramic plates. Stable complete combustion with low emissions of NO_X and CO was achieved at an equivalence ratio of 0.57-0.67 when using porous ceramic plate made of magnesia stabilized zirconia with an appropriate design configuration of the ceramic plate. However, the possibility of stable and complete combustion requires very special conditions, which makes such burners not very practical for applications. These are types of ceramic material, optical thickness or pore size of the porous ceramic plate, configuration of the porous burner and pore size. Also droplet size and distance between the spray nozzle and the upstream burner section were found critical to maintain stable operation.

Tseng et al. [4] have conducted both experimental and numerical studies of sprayed liquid heptane combustion in the PIM burners. They constructed an experimental setup which is similar to that of Kaplan et al. [3] and determined the flash-back and blow-off limits of the combustion within the zirconia porous burner. Stable combustion was still realized at a small equivalence ratio of 0.3 with an average droplet diameter of about 10 μm. Emission levels were similar to those found by Kaplan et al. [3]. Numerical results showed good agreement with the experimental ones. The predicted relationship between the burning velocity and the equivalence ratio, and also the thermal structure in terms of the gas and the solid phase temperature inside the PIM behave more like those of the premixed flame. Regenerative combustion characteristics can be observed in the numerical studies where the predicted peak temperature and burning velocity, respectively, are higher than the corresponding adiabatic flame temperature of the fuel and the burning velocity of the free-burning flame. Liquid fuel droplets experienced early complete evaporation in the low temperature region before the flame front due to the small droplets size (< 25 μm) and high volatility of the fuel. It was revealed that energy feedback mechanism by combined mode of heat transfer was not large enough to effectively enhance the enthalpy required for phase change. This may be a critical problem when using other fuels having higher heats of vaporization than that of heptane. The higher the heat of vaporization of the fuel is, the finer the droplet spray that is needed; otherwise, stable combustion flame of the liquid fuel spray within the PIM may be no longer possible.

To eliminate the need of using very fine droplet spray and a fuel atomizer, in particular when using fuel having a relatively high heat of vaporization, a special vaporization method of the fuel is needed. Takami et al. [5] supplied kerosene (instead of heptane) dropwise (instead of in droplet spray) to the top surface of a horizontal porous ceramic plate burner through a steel wire net which served as a distributor for the fuel to be uniformly distributed over the surface of the burner. Stable combustion flame, which is similar to a pool of fire, was realized on the lower surface of the burner where the fuel vapor was ignited upon which it mixed with the swirling combustion air supplied tangentially from the sidewall of the combustion chamber. Thermal structures in terms of temperature distribution and various species concentration profiles such as CO, CO_2 and NO_X along the centerline of the combustion chamber were measured to evaluate the burner's performance. Almost complete combustion in the equivalence ratio ranging from 0.5-0.9 was achieved. A favorable flammability region having the minimum equivalence ratio of 0.1 and maximum turndown ratio of 5.8 at the range of input load of about 670-3,880 kW/m^2 was also reported. Effects of kerosene input and equivalence ratio on the thermal structures were clarified. However, only the temperature profiles within the combustion chamber were measured and discussed, whereas those of the porous burner were not. Thus, the interaction between the liquid fuels and the porous burner (vaporization mechanism) that occurred inside the porous burner together with interaction between the porous burner and the adjacent combustion chamber have not yet been discussed and well-understood.

Based on the work of Takami et al. [5], this research went one step further and considered more detailed vaporization mechanism inside the porous burner. A simpler porous burner system for burning kerosene was proposed. Understanding the heat transfer phenomena, the vaporization mechanism, and the combustion characteristics, which occur simultaneously in the burner system were the primarily important goals in this study. These were done by measuring thermal structures in terms of temperature profiles along the centerline of both the porous burner and the combustion chamber. Also radial temperature profiles at any location along the centerline of the combustion chamber were measured to evaluate the performance of the downstream porous emitter. The effect of parameters such as thermal input, equivalence ratio, installation of the downstream porous emitter and its optical thickness on the combustion temperatures and emission characteristics of the burner system were clarified. Possible practical applications of this type of the burner were also suggested.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows the experimental apparatus used for the determination of the evaporation mechanism and the combustion characteristics of kerosene by the porous burner. Design and operational function of the experimental apparatus are similar to that of Takami et al. [5]. Major components and details of the present apparatus including, fuel injection chamber, porous burner, combustion chamber, fuel/air supply system and measurement apparatus are almost the same as that of [5].

However, some modifications were made, causing the present experimental apparatus to differ from the former one [5]. These are the size of the chambers, type of the porous burner, configuration of the burner, swirling air sup
ply, and temperature measurement. The chambers become larger than those of the previous one [5] and are based on a stainless steel tube 80 mm in diameter (in comparison to 55 mm in [5]) and 200 mm long for the fuel injection chamber (including the porous burner) and 300 mm long for the combustion chamber. The porous burner is formed by a stack of stainless steel wire net (instead of a porous mullite plate) having 16 mesh per inch with its overall geometrical length of 70 mm and apparent optical thickness τ_b (production of an extinction coefficient and a geometrical length) of 30. The burner configuration of the present work is a single stack of stainless steel wire net, whereas that of the previous one [5] is composed of the fuel distributor made of wire net and a porous ceramic plate with a small gap between them. The downstream porous emitter with optical thickness τ_e is installed near the exit of the combustion chamber for studying the effect of the difference of the boundary condition at the chamber outlet on the combustion behaviors. Structure of the porous emitter is similar to that of the porous burner but having different optical thickness. For simplicity, a single swirling air tube, instead of double tubes as in the previous one [5], is jointed tangentially to the combustion chamber wall at a location of about 30 mm from the lower surface of the porous burner. Temperature measurement along the axis of the chamber is made, not only in the combustion chamber as in the previous work [5] but also in the porous burner, so as to understand the vaporization mechanism and the interaction between the porous burner and the flame that occurs in the combustion chamber.

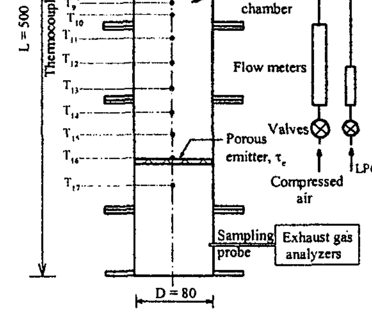

Figure 1. Experimental apparatus.

The combustion characteristics are determined from profiles of the temperature and the composition of the combustion gases at the combustion chamber outlet. The temperatures are measured by setting 17 locations of N-type sheath thermocouples (0.1 mm diameter) to the centerline of the chamber except the porous burner section (a stack of stainless steel wire net), where the thermocouples are located at the interface of the porous burner and the cement insulator. This is to prevent the kerosene flow pattern from being disturbed by the protrusion of the thermocouples. The thermocouple signals are digitized by a general-purpose data logger, and then transmitted to a personal computer. Emission of the dry combustion products at the chamber exit is carried out by using a standard combustion analyzer, which is capable of measuring CO, CO_2, O_2 and NO_X.

Operating the burner is first started by preheating the burner with LPG/Air premixed flame with appropriate equivalence ratio Φ. A flat blue flame is stabilized near the lower surface of the porous burner after ignition is initiated by inserting a pilot flame through an ignition port located in the side wall of the combustion chamber. The swirling airflow is then increased to an amount corresponding to the kerosene flow rate to be subsequently supplied with an appropriate equivalence ratio Φ. Then, the LPG is gradually decreased until it is completely turned off. During this course, the equivalence ratio is adjusted in such a way that stable combustion of the kerosene is maintained while promoting complete combustion. When combustion in the combustion chamber attains a continuously stable state, the measurements described above are performed.

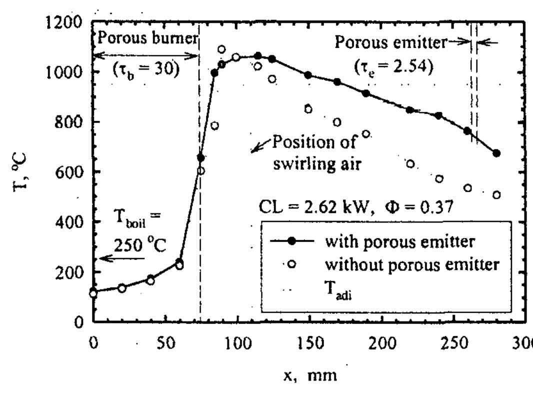

Figure 2. Temperature profiles.

3. RESULTS

3.1 Evaporation Mechanism and Combustion Regime

Steady state self-sustained combustion was realized in the experiment. Information on the thermal structure in terms of temperature profile within the porous burner and the combustion chamber is of primary importance in understanding the heat transfer phenomena, evaporation process and the combustion regime taking place in the burner system. Figure 2 illustrates a typical measured steady-state temperature profile of T_1 to T_{17} at CL = 2.62 kW, Φ = 0.37, τ_b = 30 and τ_e = 2.54. A temperature profile of the system without the porous emitter installed downstream of the combustion chamber is also included for comparison. Both temperature profiles are similar to that of the conventional gaseous premixed flame, with the maximum temperature and steep temperature gradient located close to the lower surface of the porous burner where highly luminous turbulent swirling flames were observed. Temperatures within the porous burner, especially at its downstream vicinity (T_4 and T_5), were significantly increased (above boiling temperature (T_{boil} = 250 °C) and Leidenfrost temperature [8]) owing to strong thermal radiation (emitted by the combustion flame, the combustion chamber wall and the porous emitter) and conduction through the porous burner. The higher the

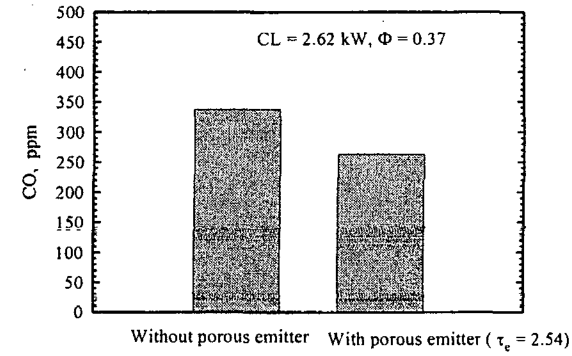

Figure 5. Comparison of CO emissions.

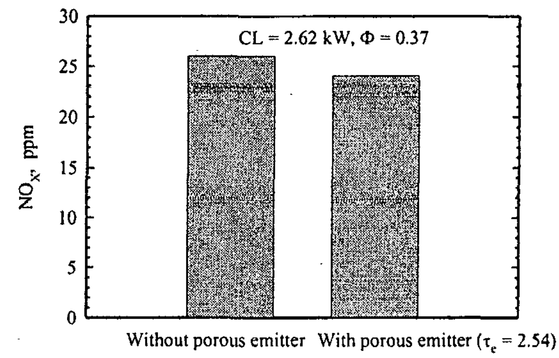

Figure 6. Comparison of NO_X emissions.

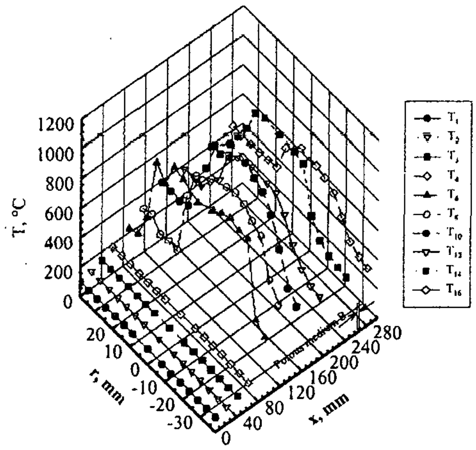

Figure 3. Radial temperature profiles.
(with porous emitter, τ_e = 2.54)

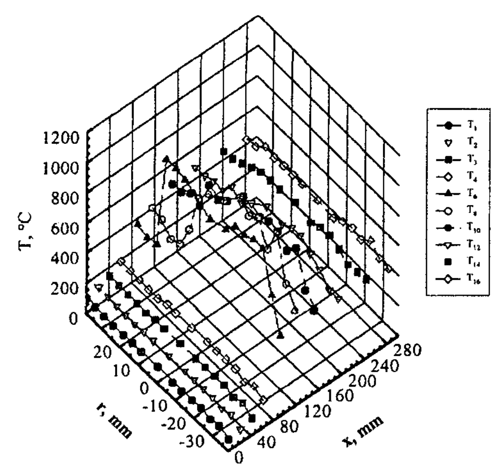

Figure 4. Radial temperature profiles.
(without porous emitter)

combustion temperature is, the higher downstream temperature (T_5) of the porous burner becomes, thus enhancing the evaporation of the kerosene fuel. The supplied liquid kerosene at normal temperature experiences initial preheating once in contact with the high temperature pore surface of the porous burner on its upper surface. The porous burner acts as not only a preheater but also a distributor for the liquid fuel to be uniformly distributed across the cross section area of the porous burner. Evaporation of the liquid kerosene commences once its temperature reaches the boiling point (T_{boil}) at an evaporation front located some where between T_4 and T_5 inside the porous burner. This makes certain a complete evaporation of the fuel takes place within the porous burner. In addition, this can help to provide the preheating effect of the fuel vapor enabling auto-ignition and the consequent intense premixed combustion with the swirling air supplied. This results in regenerative combustion having a peak combustion temperature relatively higher than the corresponding T_{adi}. The gas temperatures decrease with the axial distance x owing to heat losses through the combustion chamber wall. Nevertheless, the porous emitter being installed on the downstream side of the combustion chamber significantly improves the downstream temperatures. Also shown is that the temperature on the lower surface (T_5) for a system having the porous emitter is moderately improved. This may be attributed to an efficient energy feed-back by thermal radiation emitted from the porous emitter toward the combustion gases inside the combustion chamber, thus minimizing the convective heat loses from the combustion chamber, while enhancing fuel evaporation and fuel-vapor preheating inside the porous burner. The effect of the porous emitter on radial temperature profiles is shown in Fig. 3 and Fig. 4. The porous emitter system (Fig. 3) yields a higher peak temperature but less uniform radial temperature profiles on the downstream side than those of without the porous emitter (Fig. 4). The effects of the porous emitter on CO and NO_X emissions, respectively, are shown in Fig. 5 and Fig. 6. It was found that the combustion characteristic was improved by installing of the porous emitter on the downstream side of the combustion chamber. It is also evident in Fig. 6 that the NO_X emission in the combustion system with the porous emitter is reduced, though moderately, compared with that without the porous emitter system. This may be attributed to a staging combustion that may naturally take place in the flame, i.e., a fuel-rich flame is formed in the vicinity of the lower surface of the porous burner and then combustion of the unburned gas eventually terminates in the downstream. Thus, local temperature maximums during the combustion are suppressed.

3.2 Effect of equivalence ratio Φ

Figure 7 shows effect of the equivalence ratio Φ on the thermal structure in terms of the temperature distributions at a constant thermal input CL = 2.62 kW, τ_b = 30 and τ_e = 2.54. When Φ was decreased from 0.50 to 0.37 by increasing the swirling airflow rate, the temperature profiles inside the combustion chamber were moderately increased, whereas those inside of the porous burner were almost unchanged. The improvement in the combustion temperature within the combustion chamber may be attributed to the improvement in mixing between the kerosene vapor and the swirling air supplied at higher inlet velocity. However, the improvement in the combustion temperature is counter-balanced by the increase in the convective heat losses through the combustion chamber wall due to highly turbulent flow regime, thus maintaining the radiative heat feedback to the porous burner. This specific feature may be responsible for the possibility of the extremely lean combustion of the kerosene-air flame, which occurred at an equivalence ratio Φ as low as 0.1 [5]. Figure 8 shows the effect of Φ on the CO and the NO_X emissions. Up to equivalence ratio Φ of 0.4, the effect of the improvement in mixing and the increased process temperature are dominant, causing relatively low CO emissions. For equivalence ratios Φ above 0.4, the normal behavior of increasing CO emissions with increasing Φ is observed. Because of the low temperature levels, low NO_X are expected. NO_X emission of lower than 30 ppm was observed for every run.

3.3 Effect of optical thickness of the porous emitter τ_e

Figure 9 shows the effect of the optical thickness of the porous emitter τ_e on thermal structure in terms of the temperature profiles in the burner system. Here the experiment was conducted at CL = 2.62 kW, τ_b = 30 and equivalence ratio Φ = 0.37. Three values of τ_e have been investigated. In general, radiative energy feedback by the porous emitter can be increased with a higher optical thickness [9]. The higher the optical thickness is, the better the radiative energy feedback becomes, thus enhancing the evaporation and the combustion processes. However, high optical thickness leads to a high-pressure drop and distortion of the flow pattern of the exhaust gases, thus leading to poor ventilation from the combustion chamber. Once the pressure drop is above a limited value, the temperature continues to fall until reaction ceases and the whole combustion process is stopped. With increasing τ_e from 1.27 to 5.08, the combustion temperatures increase until a maximum is reached at τ_e = 2.54. Thus, τ_e = 2.54 is an optimal optical thickness for the porous emitter. Figure 10 shows the effect of τ_e on the corresponding CO and NO_X emissions. It is clearly seen that with increasing τ_e, the CO emission decreases until a minimum is reached at τ_e = 2.54. NO_X emissions of lower than 30 ppm were observed for every run, because of the low combustion temperature levels.

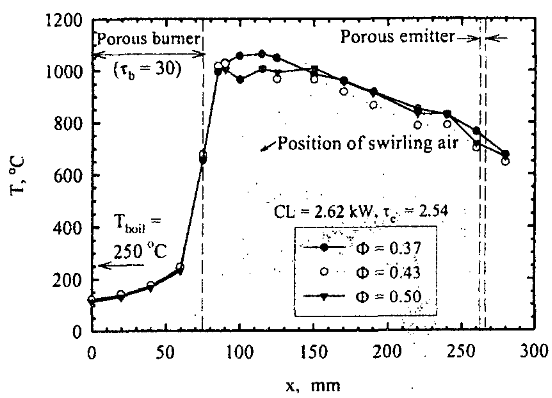

Figure 7. Effect of Φ.

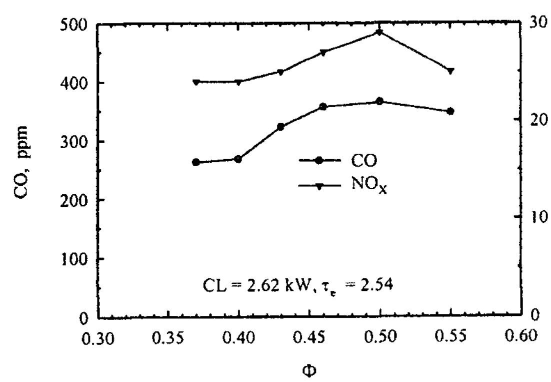

Figure 8. Effect of Φ on CO and NO_X emissions.

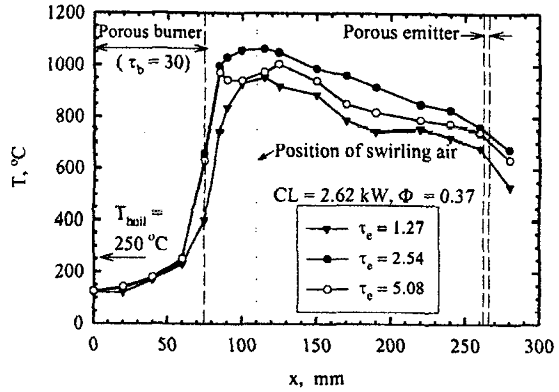

Figure 9. Effect of τ_e.

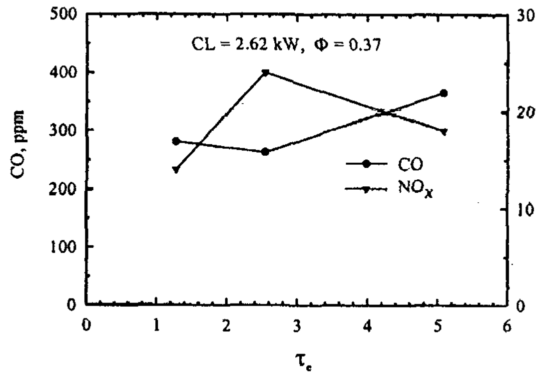

Figure 10. Effect of τ_e on CO and NO_X emissions.

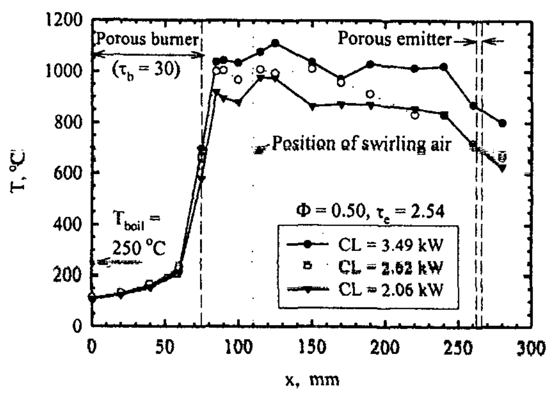

Figure 11. Effect of thermal input, CL.

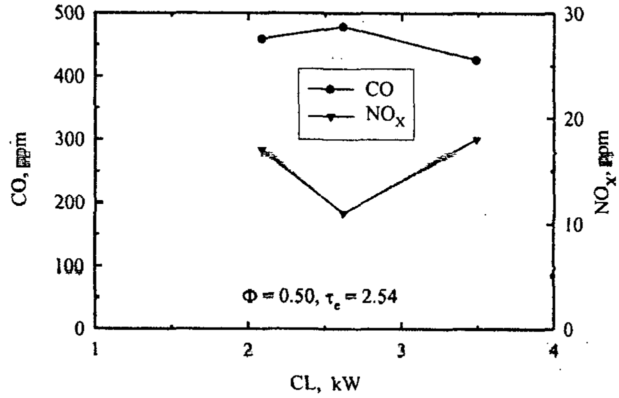

Figure 12. Effect of thermal input, CL on CO and NO_X emissions.

3.4 Effect of thermal input, CL

Figure 11 shows the effect of the thermal input CL on the temperature distributions for experimental conditions of $\Phi = 0.50$, $\tau_b = 30$ and $\tau_e = 2.54$. Within the considered range of CL, increasing CL from 2.06 to 3.49 kW by increasing the fuel flow rate in proportion to the combustion air at constant Φ yields a further increase in the temperature levels throughout the combustion chamber length. Besides, noticeable increases in T_5 and temperature gradient at the lower surface of the porous burner can be observed. Even though the increase of the CL is also proportional to the increase in the gas flow, the location of the flame zone is not likely to be shifted downstream. This special feature may be attributed to the contribution of a strong energy feedback by thermal radiation to the porous burner, where evaporation and vapor preheating effect followed by combustion can be enhanced. Thermal structures deep inside the porous burner (T_1 to T_4) are almost constant, irrespective of the increase in the chamber temperature with CL. This may be due to the sufficiently large optical thickness of the utilized porous burner ($\tau_b = 30$). Figure 12 shows the effect of CL on CO and NO_X emissions. The trend of decrease in CO emission with an increase in CL is observed. For low CL, where conditions are close to the combustible limit, the high CO emissions are due to incomplete combustion. With increasing thermal input CL the maximum temperature as well as the average temperature along the combustion chamber markedly increases to assure complete combustion. Again, NO_X emission of lower than 30 ppm was observed for every run, because of the low temperature levels. However, it is due to the special features of this combustion system that stable evaporation and combustion can be achieved. Further, good mixing and possible staging combustion, assure that no significant local temperature maximums occur during combustion in the chamber. Thus, only a negligible amount of NO_X is formed during this combustion process.

4. PRACTICAL USEFULNESS, RECOMMENDATION AND FUTURE RESEARCH NEEDS

This study has led to a new design of porous combustors for liquid fuels. The porous burner has many potential applications and can replace immediately the conventional spray burner. These are:

4.1 The simple incinerator without using any kind of conventional atomizer for incinerating hazardous liquid wastes.

4.2 Industrial burners for use in areas where liquefied petroleum gas or natural gas is not available or where a liquid fuel is a by-product of an industrial/agricultural process.

4.3 As a heat source for newly designed boilers, steam-methanol reformers and more advanced thermal systems such as thermal fluid heaters for industrial applications. In the residential and commercial areas, this concept may find applications in the development of a highly compact, efficient air heater.

Much work remains to be done to further investigate the combustion regime and the heat transfer characteristics within the new version of the porous medium burner for liquid fuels. In particular, a better understanding is needed of thermal structure with regard to radiative properties and physical properties of the porous burner used. Theoretical study is also worth doing for fully understanding the complex phenomena of heat and mass transfer, phase change and chemical kinetics which simultaneously took place inside the system. This method of heat utilization is of interest in engineering applications and needs to be examined further, as well as the ability and characteristics of the burner to burn other fuels such as diesel oil and oil palm waste etc. The ability of the burner to be scaled-up in capacity is also studied further.

5. CONCLUSIONS

An experimental study of combustion of kerosene using a porous medium burner has been carried out. Evaporation mechanism and combustion characteristics were elucidated through measured thermal structures in terms of temperature profiles along the combustor length. The effect of the main parameters on burner performance has been investigated to this end. The following conclusions can be drawn from the experimental results:

5.1 The porous medium burner plays a very important role in enhancing evaporation followed by ignition and combustion through the contribution of the thermal radiation and heat conduction through the porous burner. Hence, the porous medium burner can successfully serve as an integral function of the liquid fuel distributor, the thermal radiation absorber, the liquid fuel evaporator and the fuel vapor pre-heater.

5.2 The effect of the equivalence ratio, the optical thickness of the porous emitter, and the thermal input on the combustion characteristics was elucidated. Optimum value for the optical thickness of the porous emitter of 2.54 was found which led to favorable results. Relatively low NO_X emission was observed for the whole operating range under investigation. CO emissions were found to be strongly dependent on the operating conditions and optical thickness of the porous emitter used.

6. NOMENCLATURE

CL	=	Thermal input, kW
D	=	Diameter, mm
L	=	Length, mm
r	=	Radius, mm
T	=	Temperature, °C
x	=	Distance, mm
Φ	=	Equivalence ratio (ratio of a theoretical air to a practical air supplied)
τ	=	Optical thickness

Subscripts

adi	=	Adiabatic
b	=	Burner
boil	=	Boiling
e	=	Emitter

7. ACKNOWLEDGEMENT

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