A Study of Energy Conversion by a Porous Combustor-Heat Exchanger with Cyclic Flow Reversal Combustion

Sumrerng Jugjai and Chanon Chuenchit
Combustion and Engine Research Laboratory (CERL)
Department of Mechanical Engineering,
King Mongkut's University of Technology Thonburi (KMUTT),
91 Sukswas 48, Bangmod, Tungkru, Bangkok 10140,
THAILAND

ABSTRACT

Present observations of energy conversion from low-grade fuels aimed at producing thermal energy, strongly suggest that cyclic flow reversal combustion in a porous medium (CFRC) is a promising approach for future applications. The CFRC is very advantageous from the aspects of fuel conservation, efficiency, combustion intensity and emission pollutants because of its prominent ability in creating an efficient internal heat recirculation from the hot exhaust gases to the unburned mixture when compared with that of conventional burners. Against this background, a novel porous combustor-heat exchanger (PCHE) equipped with the CFRC was developed for abstracting heat from a typical low-grade gaseous fuel. The PCHE incorporates features of a porous medium, which can be used as an efficient compact combustor or a heat exchanger. With this combination, the PCHE in the form of a concentric cylinder is arranged in such a way that the inner cylinder is the porous combustor, which serves as a radiant burner equipped with CFRC. The outer cylinder is the porous heat exchanger, which acts as an integral function of a radiative heat absorber, a compact heat exchanger and a thermal insulator for obtaining maximum thermal shielding. The radiative heat flux converted from the heat of combustion is, therefore, effectively converted into a substantial increase in the enthalpy of the process air flowing through the porous heat exchanger. Performance of the PCHE is verified by performing parametric studies of some dominating parameters, i.e., half-period, equivalence ratio and thermal input, which affect thermal efficiencies and emission characteristics. The PCHE equipped with the CFRC effectively abstracts heat from the typical fuel with a minimum apparent heat content of 0.62 MJ/m^3 [normal] at which the maximum preheated process air temperature of 150 °C is obtained with relatively low emission pollutants. Optimum operating condition for the PCHE should be at relatively short half-period, low equivalence ratio and high thermal input.

UNITS and ABBREVIATIONS

CL	Thermal input, kW
d, D	Diameter, mm
R_a	Loading ratio (ratio of mass flow rate of the cooling air to that of the exhaust gas)
T	Temperature, °C
t	Time, s
t_{hp}	Half-period, s
u	Velocity of the combustion gas, m/s
V_a	Volume flow rate of the cooling air, m^3/s
x, y	Distance, mm
Φ	Equivalence ratio (ratio of a theoretical air to a practical air)
η_h	Thermal efficiency (ratio of net enthalpy increase of the cooling air to thermal input, CL)

Subscripts

air, in	Air at inlet
air, out	Air at outlet
av	Time-average value over one cycle
i	Inner surface of the porous heat exchanger
o	Outer surface of the porous heat exchanger

1. INTRODUCTION

With recent development in the state-of-the-art technology of the super-adiabatic combustion [1-4] (hereafter referred to as cyclic flow reversal combustion, CFRC), attention has been focussed on the application of this technique to replace the conventional one-way flow combustion (hereafter referred to as OWFC) in several types of combustion equipment and industrial furnaces [5, 6]. The CFRC, in which a ceramic honeycomb is used as a heat storage medium and a combustor, has a prominent characteristic in that it is capable of creating an effective internal recirculation or a recuperation of heat from the burned exhaust gases to the cool mixture with a substantial increase in preheating effect of the mixture before combustion. This results in the so called "excess enthalpies" or "super-adiabatic flame temperature" combustion [3, 4], which has a peak temperature considerably higher than the corresponding adiabatic flame temperature of the mixture, yielding a high combustion temperature, high combustion intensity, preferable flame stability and flammability limits when compared with those of the conventional OWFC. The CFRC is, thus, very suitable for burning mixtures normally considered incombustible, which are currently used mostly for incineration. Examples of the mixtures are, exhausts of a wide variety of industrial processes, ventilation from mines, blast furnace gases, lean methane/air mixtures from around some fermenting wastes, etc., which are potentially capable of exothermic reaction, and contain very large amounts of recoverable energy. Therefore, the CFRC can help to save a substantial proportion of energy with mitigation of the emission pollutants when used to replace the conventional burners. Moreover, at relatively high thermal input of lean combustion, the CFRC yields steep temperature gradients at both ends of the combustor, allowing a high temperature plateau to exist in the rest of the combustion chamber while mixtures of heat content considerably lower than the normal limit of flammability can be burned stably [4, 7]. These outstanding features of the CFRC are very important in terms of its utilization for augmenting combustion and enhancing the heat transfer in newly designed thermal facilities. In addition, the CFRC provides a partial exhaust gas recirculation caused by a cyclic flow reversal motion of the fresh mixture and the exhaust gas. This yields low Damkohler number combustion, where the formation of NO_x is suppressed [8].

All these characteristics are suitable for the application of the CFRC technology to abstracting heat from low-grade fuels. To serve this purpose, a newly developed porous combustor-heat exchanger (PCHE) equipped with the CFRC was proposed to demonstrate its advantages. Figure 1 shows the schematic diagram of the PCHE, wherein a relatively lean super-adiabatic combustion of a typical low-grade fuel takes place inside the combustor. Utilization of the combustion heat can be done by capturing

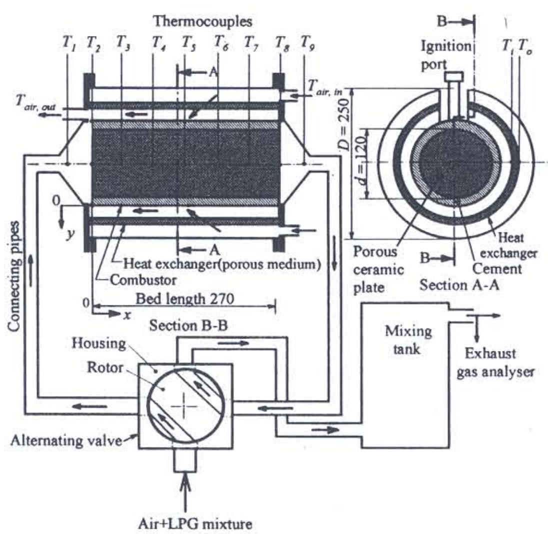

Fig. 1. Pororus combustor-heat exchanger (PCHE).

radiative heat flux emitted from the external surface of the combustor. This is done by enclosing the lateral surface of the combustor with a porous medium, through which the air is flowing, having high surface area to volume ratio and sufficient optical thickness. By these means, the surrounding porous medium acts as an integral function of a radiative heat absorber, a compact heat exchanger and a thermal insulator [9, 10] for the combustor, whereas the combustor serves as a radiant burner equipped with the CFRC, the function of which is to convert as much as possible the heat of combustion to thermal radiation. The efficiency of the radiant burner can approach 100 % provided that the maximum working temperature tends to infinity. At high temperatures, thermal radiation becomes a dominant mode of heat transfer because the inevitable convection and conduction losses when the radiant surface is surrounded by gas become very small when compared with the T^4 dependence of radiant emission [3]. The radiative heat flux is effectively converted into an increase in the enthalpy of the process air that flows through the porous medium, leading to a substantial increase in the air temperature.

The present study proposes a porous combustor-heat exchanger (PCHE) equipped with the CFRC for abstracting heat from a typical low-grade gaseous fuel. Combustion phenomena and heat transfer characteristics, respectively, taking place inside the PCHE are experimentally elucidated. Performance of the PCHE would be assessed through thermal structure of the process air in the porous heat exchanger. Effects of various parameters that are expected to control the heat transfer performance, combustion phenomena and emission characteristics of the PCHE such as half-period (t_{hp}), equivalence ratio (Φ) and thermal input (CL) are elucidated.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows the proposed experimental apparatus of the PCHE. It consists of three main components; a combustor, a heat exchanger and an alternating valve connecting with the inlet manifold. The combustor and the alternating valve are connected by pipes for carrying the combustible mixture and the exhaust gases, respectively, into and out of the combustor. The alternating valve permits a variable combination of inlet and outlet to be selected from the combustor. The exhaust gas, of which its emission level is varied with time when operating the CFRC, is directed to a large mixing tank to stabilize the emission level before measuring. The inner surface of the combustor is lined with high temperature cement 12 mm thick, allowing high temperature combustion. The space inside the combustor is filled with a stack of circular honeycomb porous ceramic plate (d = 120 mm) with each plate having 6 pores per cm (ppcm) and is 15 mm thick. The component of the ceramic is magnesia-stabilized zirconia. The external surface of the combustor is surrounded by a concentric porous medium to serve as a heat exchanger for preheating the flowing air. The heat exchanger with its thickness of 10 mm is formed by a stack of pieces of stainless steel wire net having 32 mesh per inch. Heat liberated from the high temperature combustor wall by thermal radiation is absorbed by the porous heat exchanger, wherein the air is preheated by the interaction between the gas and the solid phase [9]. An ignition port is installed near the middle of the combustor wall for inserting a pilot flame into the combustor.

The alternating valve consists of a rotor and a housing. The rotor is a solid steel cylinder having two drilled holes for conveying the mixture and the exhaust gas, respectively, into and out of the combustor via the connecting pipes. The rotor alternates back and forth by hand or by an external driving device comprised of an AC motor and a reciprocating cam mechanism to periodically change the flow direction of the mixture (forward flow or backward flow) into the combustor. With this design, the PCHE can be operated in a manner as the cyclic flow reversal combustion (CFRC) or the one-way flow combustion (OWFC), depending on the operation of the alternating valve. For the CFRC or the OWFC, a forward flow is defined as a clockwise flow of the gas through the connecting pipes, while a backward flow is defined as a counter-clockwise flow. The time interval (or half-period, t_{hp}) for each flow direction can be independently adjusted by clock or by automatic time switches.

N-type sheath thermocouples of 0.5 mm diameter wire, were used in the experiment at all measuring locations for both the combustor and the heat exchanger. T_1 and T_9 indicate the gas temperature at about 55 mm from both ends of the combustor, whereas T_2 to T_8, which are equally spaced along the axial direction of the combustor, represent the solid phase temperatures (porous ceramic) inside the combustor. T_i and T_o, respectively, represent the inside surface temperature and the outside surface temperature of the porous heat exchanger, which serves as a radiative heat absorber for preheating the flowing air. $T_{air,\,in}$ and $T_{air,\,out}$, respectively, represent the inlet and the outlet temperature of the air flowing through the porous heat exchanger. The thermocouple signals are digitized by a general-purpose data logger, and then transmitted to a personal computer.

Liquefied petroleum gas (LPG) diluted with air was used as a typical low-grade fuel in the experiment. The composition of the LPG was propane (C_3H_8) 40% (by vol.) and butane (C_4H_{10}) 60%, respectively, with a low heating value of about 115 MJ/m³[normal]. An air compressor with pressure regulator was used for supplying the cooling air to the porous heat exchanger and for the combustion air to homogeneously mix with the LPG at the inlet manifold prior to entering the alternating valve. The combustion air, the cooling air and the LPG were metered by calibrated rotameters.

Operating the PCHE was first started by setting the position of the rotor at either forward flow or backward flow, i.e., OWFC. Then the mixture, with an initial equivalence ratio Φ close to 1, was supplied into the combustor. The mixture was then ignited by a pilot flame inserted through the ignition port. The combustion flame was stabilized near the middle portion of the combustor with an initial equivalence ratio Φ of about 0.8. After the thermal equilibrium was reached, air was then supplied through the porous heat exchanger with a typical constant loading ratio R_a of 0.81. Then, the OWFC was switched to the CFRC by operating the alternating rotor with an initial half-period t_{hp} = 60 seconds. A steady-state condition of the CFRC was reached once the constant amplitude and the constant average over a half-period t_{hp} of the fluctuation temperatures T_1 to T_9 were obtained. The emission measurement of the dry combustion products was carried out by using a standard combustion analyzer.

3. RESULTS AND DISCUSSIONS

3.1. Combustion Characteristics and Heat Transfer Performance

Steady-state self-sustained combustion conditions for the CFRC were realized in the experiment. Figure 2 illustrates a typical transient change of the measured combustion gas temperatures T_1 to T_9 for the CFRC. The corresponding transient temperature distribution along the combustor axis is depicted in Fig. 3. The fluctuation in temperatures in Fig. 2 during a half-period t_{hp} is caused by reversal of the flow direction of the cool mixture at a regular time interval (t_{hp} = 60 s). A hot exit from the previous half-period t_{hp} (T_1 at t = 0 s in Fig. 2 and Fig. 3) becomes a cool inlet for the present half-period, and vice-versa, while the alternating valve is operating. Therefore, amplitude of the fluctuation depends on the location of the temperature measured. A typical large amplitude of about 440 °C for T_2 was observed at the left end of the combustor, whereas at the near middle point of the combustor an amplitude of T_5 in Fig. 2 was nearly zero irrespective of the flow direction of the mixture. A large amplitude of T_2 and T_8 in Fig. 2 shows efficient heat transfer between the solid phase and the gas phase leading to an efficient preheating of the mixture and augmentation of combustion. T_3 and T_6 in Fig. 3 were the maximum temperatures, implying that two reaction zones had occurred and stabilized there. The flame location has been alternately changed from the location of T_3 to T_6 depending on the flow direction. This yielded a near symmetrical temperature distribution along the axis of the combustor. The temperature profile at the end of the previous half-period (t = 0 s) is nearly symmetrical with that of at the end of the present half-period (t = 60 s).

Figure 4 shows a comparison of axial temperature distributions of the combustion gas between the OWFC and the CFRC at the same experimental conditions. The corresponding adiabatic flame temperature T_{ad} = 1,114 °C is also included for comparison. Since the gas temperatures of the CFRC were varied with time, time-averaged temperature profiles during one cycle ($2t_{hp}$) of the CFRC were plotted throughout the experiment. The OWFC provides a steady, single reaction zone with maximum temperature higher than T_{ad}, allowing a relatively narrow region of high temperature combustion to be stabilized downstream of the combustor. Unlike the OWFC, the CFRC flame yields transient, two-reaction zone combustion near the combustor ends, allowing a more uniform temperature distribution along the bed length to exist with a steep temperature drop at both ends of the combustor, and this fact suppresses the convective heat loss with the exhaust gas. This may be attributable to a greater heat transfer to the air for the CFRC than that of the OWFC as shown in Fig. 5, even though the maximum temperature of the CFRC was relatively less than the corresponding T_{ad}. This is because more radiative heat losses from the combustor end by which the flames were stabilized at this relatively high equivalence ratio $\Phi = 0.44$. A higher temperature in T_i and T_o of the CFRC than that of the OWFC was clearly observed across the porous heat exchanger. This implies that a more radiative heat flux emitted from the combustor wall of the CFRC is effectively absorbed by the porous heat exchanger before converting into an increase in enthalpy of the flowing air.

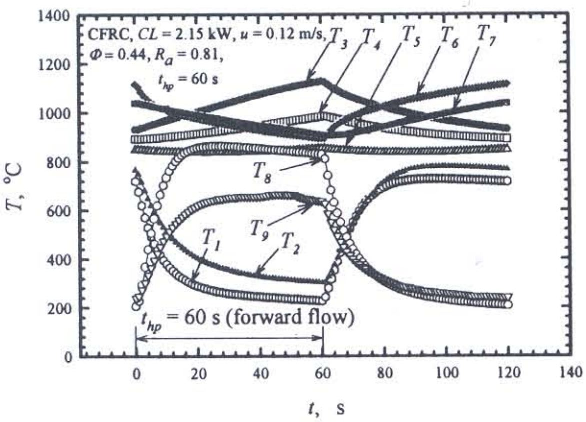

Fig. 2. Typical transient change of combustion gas temperature for the CFRC.

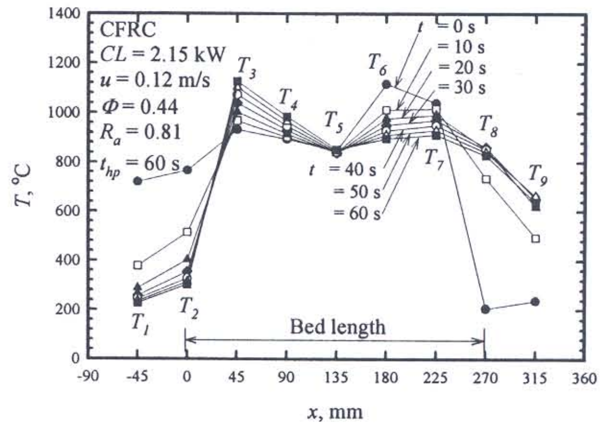

Fig. 3. Transient combustion gas temperature distribution for the CFRC.

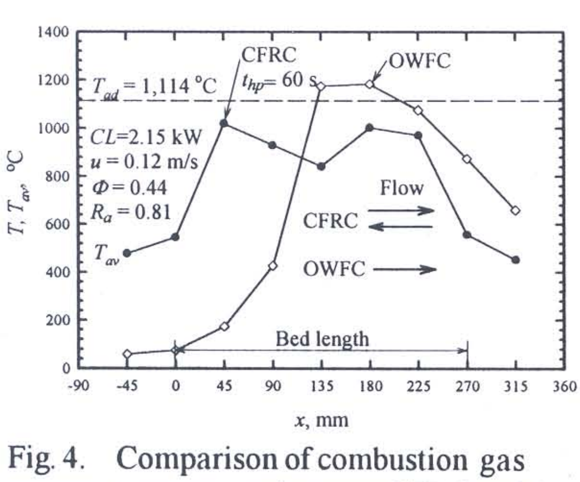

Fig. 4. Comparison of combustion gas temperature between CFRC and OWFC.

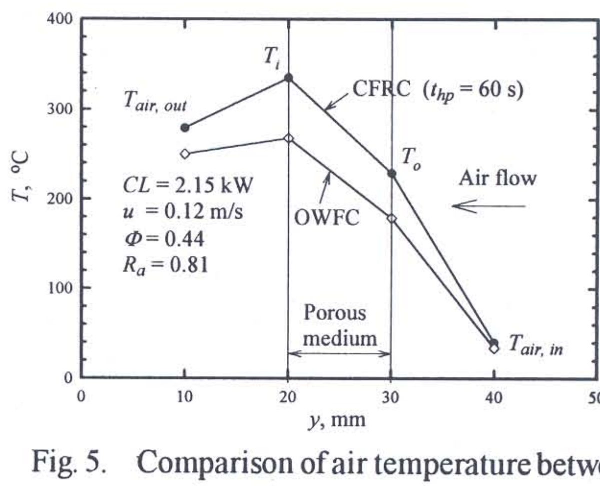

Fig. 5. Comparison of air temperature between CFRC and OWFC.

Figure 6 shows comparison in flammability limits between the CFRC and the OWFC under the constant air volume flow rate $V_a = 0.06$ m³/min, which serves a thermal load in the heat exchanger. It is clear that the CFRC yields exceptional lean combustion condition at minimum $\Phi = 0.15$, which is relatively small when compared with that of the OWFC. The corresponding apparent heat content of the mixture at this equivalence ratio is about 0.62 MJ/m³ (normal). Thus, possible heat abstraction from low-grade fuels or lean mixture may be accomplished by the proposed PCHE equipped with the CFRC. In the subsequent section, focus has been made on the parametric studies of the PCHE equipped with the CFRC only so as to understand the effects of some dominating parameters, i.e., equivalence ratio, half-period and thermal input on the combustion regime and the heat transfer characteristic of the PCHE.

3.2 Effect of Half-Period, t_{hp}

Figure 7 shows the effects of the half-period t_{hp} on thermal structures in terms of profiles of the time-average temperatures for the CFRC. Here the experiment was conducted at a relatively low equivalence ratio $\Phi = 0.23$ at which the CFRC has shown its prominent combustion characteristic by providing a maximum combustion temperature of about 1,000 °C. This value is considerably higher than the corresponding T_{ad}, which is equal to 645 °C. The temperature profiles have shown a trapezoid-like shape with steep temperature gradients and low temperatures of T_2 and T_8 at both ends of the combustor. Thus, the convective heat losses with the exhaust gas is significantly suppressed at this relatively low equivalence ratio $\Phi = 0.23$. As t_{hp} increases, T_2 and T_8 are nearly constant irrespective of the t_{hp}. Despite the considerable change in the t_{hp} from 60 s to 180 s, there was a moderate change in the maximum temperature of the combustion gas. This may be attributed to the equivalence between the total released heat and the total stored heat in every considered half-period. The total released heat is defined as the heat released by the porous ceramic plate in the combustor to preheat the incoming premixed gases on the inlet side of the porous combutor during a considered half-period, whereas the total stored heat is the heat transferred from the exhaust gases to the porous combustor when they are cooled down prior to leaving the porous combustor during the same half-period [2]. If the half-period t_{hp} becomes too long, flame extinction may occur due to an out moving of the flame from the combustor. The longer the half-period t_{hp}, the deeper the flame moves inside the combustor, resulting in a less uniform temperature distribution with less area for high radiative heat flux emitted to the porous medium surrounding the combustor. This may be attributable to a moderate decrease in the air temperature and the corresponding thermal efficiency, respectively, as shown in Fig. 8 and Fig. 9.

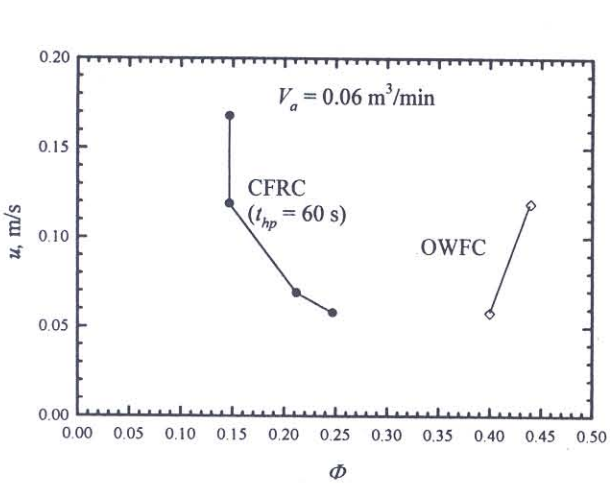

Fig. 6. Comparison of flammability limit between CFRC and OWFC.

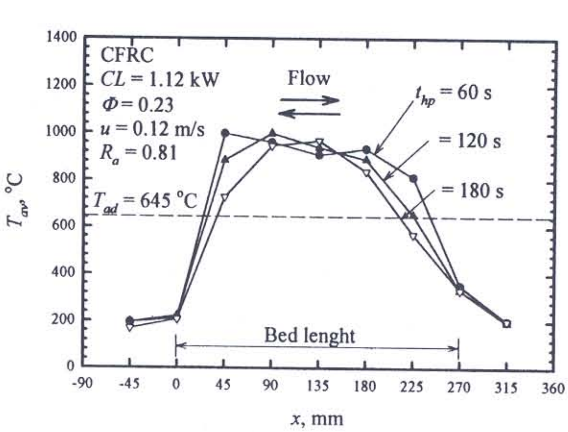

Fig. 7. Effect of t_{hp} and t_{av}.

3.3 Effect of Equivalence Ratio Φ

Figure 10 shows the effect of the equivalence ratio Φ on the thermal structure in terms of the time-average temperature distributions for the PCHE. The equivalence ratio is expected to have a dominating effect on the thermal structure, since the thermal input CL is proportional to it. When Φ was decreased from 0.44 to 0.15 by decreasing the fuel flow rate, the maximum combustion temperatures were hardly decreased but the flame location (defined by the location of the maximum temperature) shifted from the near inlet ends to deep inside of the combustor. At a small equivalence ratio $\Phi = 0.15$, which is the flammability limit of this PCHE, the temperature gradients at both ends of the combustor come together with almost no high temperature plateau, resulting in a change of temperature profile from the trapezoidal shape to the triangular one. If Φ is further decreased, the maximum temperature drops considerably, leading to an incomplete combustion and to an extinction of the combustion flame. An interesting feature of this PCHE can be seen from the fact that at $\Phi = 0.15$ the maximum combustion temperature of about 1,000 °C was obtained, the value of which is far higher than the corresponding $T_{ad} = 445$ °C. The air temperature in the porous heat exchanger was considerably decreased with Φ as shown in Fig. 11, whereas, thermal efficiency h_{th} is showing a tendency to improve as shown in Fig. 12. This implies that the PCHE with the CFRC technique is very suitable for abstracting heat from low-grade fuels.

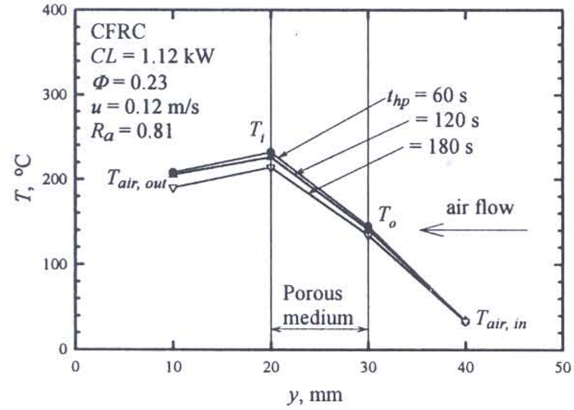

Fig. 8. Effect of t_{hp} on air temperature.

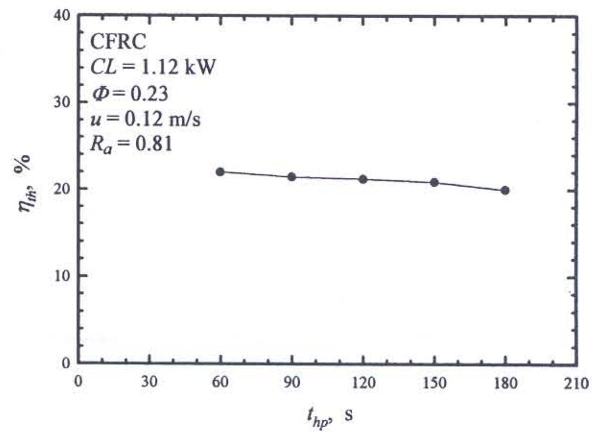

Fig. 9. Effect of t_{hp} on η_{th}.

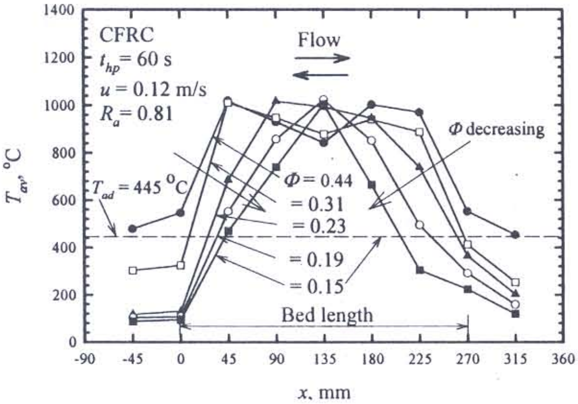

Fig. 10. Effect of Φ on T_{av}

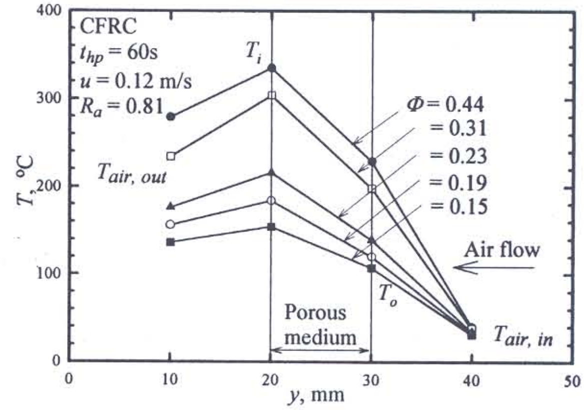

Fig. 11. Effect of Φ on air temperature.

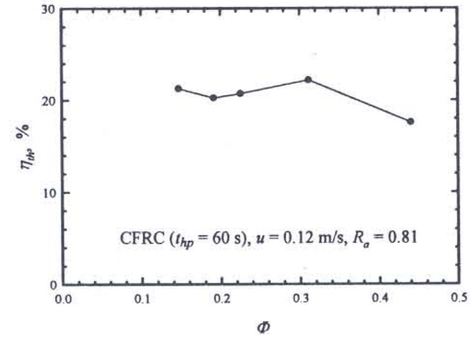

Fig. 10. Effect of Φ on η_{th}.

3.4. Effect of Thermal Input

Figure 13 shows the effect of thermal input CL on time-average temperature distributions of the PCHE for an experimental conditions of Φ = 0.24 and t_{hp} = 60 s. Within the considered flow range, increasing the thermal input from CL = 0.56 to 2.15 kW, by increasing the fuel flow rate in proportion to the combustion air at constant equivalence ratio Φ, yields higher and wider temperature plateaus. These result in a marked increase in the air temperature as shown in Fig. 14, even though the temperature at both ends of the combustor (T_2 and T_8) tend to increase with CL. However, the thermal efficiency decreases as the thermal input increases as shown in Fig. 15, owing to the increase in the convective and radiative heat losses at both ends of the combustor.

3.5. Emission Characteristics

Figure 16 shows a typical emission characteristic in relation to the equivalence ratio Φ. Undoubtedly, within the range of the experimental condition, the NO_x emission is in the order of less than 5 ppm, owing to the relatively lean combustion at relatively low combustion temperatures of the CFRC. At low equivalence ratio (Φ < 0.2) the NO_x emission is negligibly small, whereas the CO emission is relatively high due to a reduction in the combustion temperature.

4. PRACTICAL USEFULNESS / SIGNIFICANCE

This study has led to a new design of porous combustor-heat exchanger (PCHE) for abstracting heat from low-grade fuel. The PCHE has many potential applications to replace the conventional heat exchanger. These may include:

- An integrated combustor and heat exchanger for burning low calorific fuels or ultra-lean mixtures such as blast furnace gas and toxic gas (e.g., volatile organic compounds, VOCs) for the purpose of fuel saving and mitigation of the emission pollutants. Although the gases represent very low energy concentration and potential conversion, they may be of practical use because their heat is sufficient to sterilize, humidify and heat air for direct use in buildings. They are able to pre-gasify premix and pre-heat solid or to pre-vaporize liquid fuels prior to burning so that a blue "Bunsen burner type" of flame can result from burning very unpromising feedstock.
- A state-of-the-art technology for a new version of a steam-methanol reformer 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 highly compact, efficient air heater.

The research presented here elucidates several characteristics of the proposed PCHE equipped with the CFRC technique. With further development a commercial PCHE of this type may be possible.

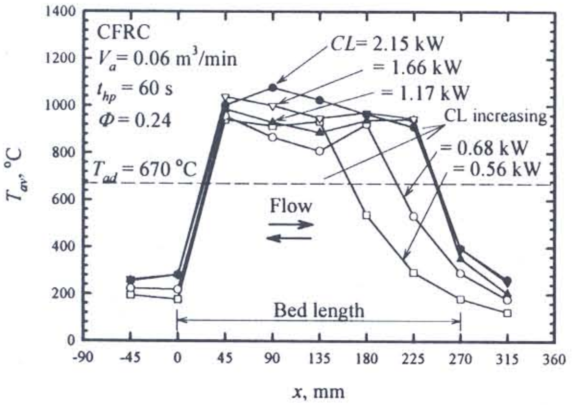

Fig. 13. Effect of CL on T_{av}.

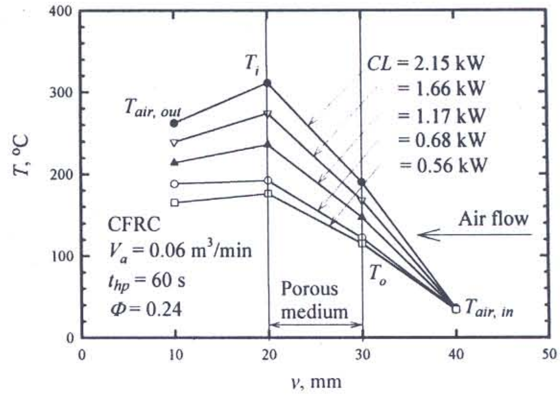

Fig. 14. Effect of CL on air temperature.

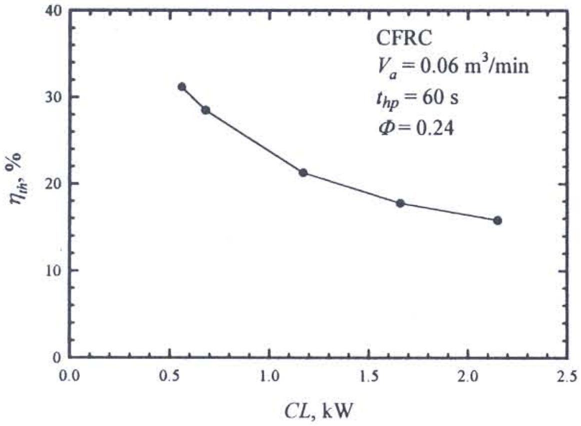

Fig. 15. Effect of CL on η_{th}.

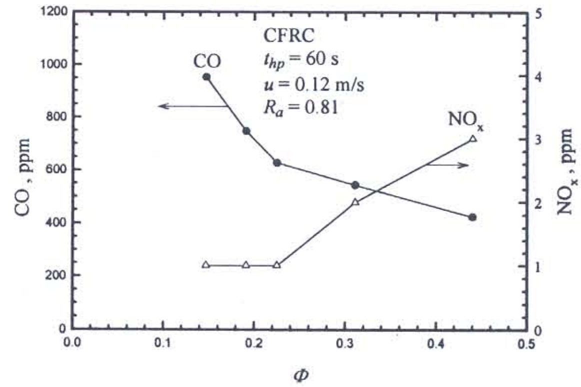

Fig. 16. Emission characteristics.

5. CONCLUSIONS

This research was undertaken to explore the potential of the PCHE equipped with the concept of the cyclic flow reversal combustion (CFRC) in burning a typical low-grade gaseous fuel for the purpose of energy saving. The combustion phenomena and the heat transfer characteristics of the proposed PCHE were evaluated by comparing its heat transfer performances and the combustion characteristics with those of conventional one-way flow combustion (OWFC). Operating parameters such as equivalence ratio, half-period and thermal input expected to control the performance of the new PCHE equipped with the CFRC were clarified. The following conclusions can be drawn from the experimental results:

- The proposed PCHE has proven to be suitable for energy conversion and utilization of low-grade gaseous fuels. The PCHE yielded a self-sustained combustion with a minimum equivalence ratio of $\Phi = 0.15$ for a typical fuel used, having the apparent heating value and the corresponding adiabatic flame temperature, respectively, as small as 0.62 MJ/m³ (normal) and 445°C. At this combustion condition, the maximum temperature of about 150°C of the preheated process air at loading ratio of 0.81 was obtained. The PCHE can be used to save fuel and enable lean mixture to be burned at higher rates, thus extending the definition of what is a fuel.
- Optimum operating conditions for the PCHE equipped with the CFRC technique should be at relatively short half-period, low equivalence ratio and high thermal input.

6. RECOMMENDATION AND FUTURE RESEARCH NEEDS

Much work remains to be done to further investigate the combustion regime and the heat transfer characteristics within the new version of the PCHE equipped with the concept of the CFRC. In particular, a better understanding is needed of thermal structure with regard to radiative properties and physical properties of the porous ceramic used. Also the relatively low thermal efficiency obtained could be alleviated by not lining the inner surface of the combustor wall with the high temperature cement, allowing the wall temperature together with the thermal radiative heat flux directed to the heat exchanger to be substantially increased. The ability of the new PCHE to be scaled up in capacity also needs to be examined further.

7. ACKNOWLEDGEMENT

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