Effect of Tube Length on Combustion Characteristics of a Self-aspirating Radiant Tube Burner (SRTB)

Chanon Chuenchit [1] and Sumrerng Jugjai [2,*]

[1, 2] Combustion and Engine Research Laboratory (CERL), Department of Mechanical Engineering,
Faculty of Engineering, King Mongkut's University of Technology Thonburi, Bangkok, Thailand

Abstract

Radiant tube burners (RTBs) are widely used in heating process of industrial production such as batch annealing and heat treating process. Most of the RTBs operate at high capacity that use a lot of fuel and combustion air to yield high gas temperature. Therefore, the air compressor is needed in order to supply the sufficient air for completely combustion. The objective of this research is to study the effect of the tube length on combustion characteristics of a self-aspirating radiant tube burner (SRTB) equipped with an energy recirculation using a porous heat exchanger concept. Three different lengths of U-shaped tube of 10 cm in diameter were selected, i.e. 0.8 m, 1.0 m and 1.25 m, respectively, which are called as tube S, M and L in the study. Because the combustion air is naturally entrained through the venturi burner by the momentum of fuel jet, therefore the air compressor is not used in this method. Temperature distribution along the SRTB and emission (NO_x & CO) were measured. The results show that the temperature distribution of the tube L is higher than the other tubes at the same firing rate (*FR*). The flame can stabilize inside the SRTB for the *FR* in the range of 3.66-8.28 kW. The maximum temperature inside SRTB is 1,481°C. The NO_x and CO emission are less than 336 ppm and 450 ppm, respectively.

Keywords: Radiant tube burner; Self-aspirating; Heat recirculation, Porous heat exchanger.

1. Introduction

Radiant tube burners (RTBs) appear well suited for the industrial processes that require a high and homogeneous temperature distribution within the furnace and the exhaust gas does not contact to the product. Increasing of energy efficiency is the most important topics in order to reduce fuel consumption and pollutant emissions. Preheating of combustible mixture by recycled heat from exhaust gas has been considered as an effective method for fuel conservation. This type of combustion has been called ''excess enthalpies'' or ''super-adiabatic flame temperature'' combustion in which the reactants (or the combustion air alone) are preheated using heat ''borrowed'' from beyond the flame zone, without mixing the two streams [1]. The heat recirculation systems have been extensively applied for increasing the combustion air temperature. Moreover, it expands the combustion stability. The new type of glass melting regenerative furnace at Nippon Furnace Kogyo (NFK) makes it possible to preheat the combustion air, typically above 1,000°C, by re-circulating the heat from the hot exhaust gas resulting in fuel saving up to 50% over cold air of the conventional systems, as well as a significant decrease of greenhouse gas CO_2 and NO_x emissions [2].

Heat-recirculating combustion based on the porous medium technology has now been considered as the emerging furnace design methodology for the next generation of high performance combustion systems. Based on the prominent feature of the porous medium in effectively converting energy between flowing gas enthalpy and thermal radiation, the development of a high performance heat exchanger using a porous medium was proposed by Echigo [3]. The basic concept consists of a pair of porous mediums separated by a solid wall as shown in Fig. 1.
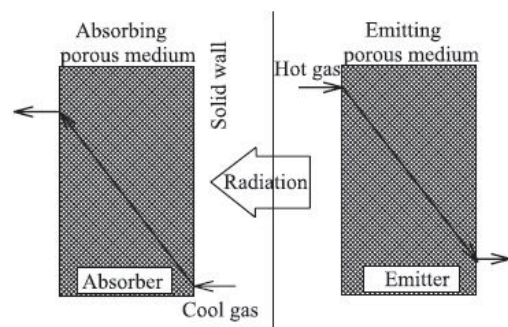

Fig. 1. Energy conversion by porous medium.

Enthalpy of the hot flowing gas (combustion products) is effectively converted to thermal radiation emitted by the high temperature (heating) side porous medium designated as the ''emitter'' and directed to the solid wall. In thermal equilibrium, the reversed conversion from the incident thermal radiation emitted from the solid wall to the cool gas (or combustion air) enthalpy takes place in the low temperature (cooling) side porous medium designated as the ''absorber'', leading to an efficient method in preheating the combustion air. Jugjai and Rungsimuntuchart [4] applied the heat recirculation principle to a self-aspirating burner by employing a porous heat exchanger in combination with the swirling central flow burner (SB). The preheating temperature of the mixture can be increased up to 300°C without flashback. The results showed that the thermal efficiency of the proposed burner is higher than that of a conventional radial flow burner (CB) by about 30 percentage points (50% relative). Scribano et al [5] have been introduced the heat recirculation system by using of exhaust gas recirculation (EGR) techniques in order to enhance the performance of the industrial radiant tube burner. It was shown that the

NO_x emissions at the exhaust significant decrease up to 50% with respect to the original burner without the EGR. However, because almost all of the RTBs operate at high capacity that use a lot of fuel and combustion air to receive high temperature ($>1,200^o$C), therefore, the air compressor is needed in order to supply a sufficient air for completely combustion. In recent years, Chuenchit and Jugjai [6] applied the heat recirculation principle to the a self-aspirating radiant tube burner (SRTB) by employing a porous heat exchanger integrated with a self-aspirating burner and radiant tube in one unit, leading to an innovation for a radiant tube burner. The results showed that the temperature profile along the tube is higher than that of conventional radiant tube burner without the heat recirculation.

The main objective of this research is to study the effect of tube length on combustion characteristics of the SRTB equipped with the energy recirculation of a porous heat exchanger concept. The tube length significantly affects the pressure drop of radiant tube and thus optimization of the tube length is required. A unique advantage of the self-aspirating burner is that the combustion air is naturally entrained through the venturi burner by the momentum of fuel jet. Therefore, the air compressor is no longer desired by using this burner. Moreover, in order to save energy, the energy recirculation principle using porous medium technology is selected in this work. The SRTB is developed for using in a small-scale drying process and heating process at lower firing rate (lower than 10 kW), which has never been studied before.

2. Experimental Set-up

Fig. 2 shows the schematic diagram of the experimental set-up of the SRTB, whereas Fig.3 shows its photograph. The flow rates of LPG (1) (40% vol. of propane and 60% of butane) with low heating value of about 46 MJ/kg are controlled by a pressure regulator (2). The flow meter (3) is installed downstream of the pressure regulator in order to measure the firing rate (*FR*) of the fuel. A mercury manometer (4) is used to measure the fuel inlet pressure. Then the fuel emerges from an injector orifice (5). On leaving the injector, the fuel entrains the primary air by a momentum-sharing process between the emerging fuel and the preheated primary air (blue arrow). The fuel/air mixture enters the venturi burner (6) and entrains the secondary air (black arrow) followed by combustion (red arrow) in the U-shaped, stainless radiant tube (7) having 0.1 m inside diameter and 1 mm in thickness. In order to simulate and understand the heat recirculating combustion without thermal loading, the radiant tube was covered by ceramic fiber insulator of 30 mm in thickness to reduce heat lose. The oxygen concentration in the mixture and the exhaust gas compositions are measured by the sampling probe (14) and (15), respectively, and then carried to the portable emission analyzer, Messtechnik Eheinm model Visit01L (16). The measuring range is 1-10,000 ppm for CO and 0-4,000 ppm for NO_x with a measuring accuracy of about ±5 ppm and a resolution of 1 ppm for both CO and NO_x. All measurement of emissions in the experiment is those corrected to 0% excess oxygen and dry basis. All data are recorded by computer (17).

The temperature *T* at different locations on the axial axis of the SRTB are also recorded as shown in Fig. 2. There are 9, 10, and 8 locations of thermocouples from the venturi burner outlet (*x* =0) to the radiant tube exit for three different tube length as specified by S, M and L, respectively. $T_1 - T_3$ are measured by B-type and the other locations are measured by N-type thermocouples. End of the U-tube is connected to the porous heat exchanger (13), the structure of which is composed of the stainless steel porous emitter (8) having a conical shape with several holes of 2 mm in diameter drilled on its surface for converting a part of gas enthalpy to thermal radiation to the separating wall (9) and then reradiated to the porous absorber (10) for preheating the primary air [3].
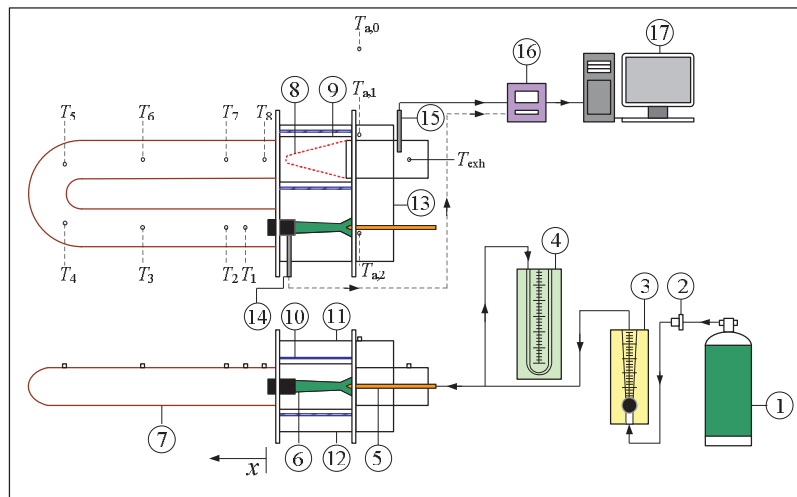
1. Fuel (Liquefied Petroleum Gas, LPG)
2. Pressure regulator
3. Flow meter
4. Mercury manometer
5. Injector orifice
6. Venturi burner
7. Radiant tube
8. Porous emitter
9. Separating wall
10. Porous absorber
11. Upper cover
12. Lower cover
13. Porous heat exchanger
14. Mixture sampling probe
15. Exhaust gas sampling probe
16. Gas analyzer
17. Computer

Fig. 2. Schematic diagram of the SRTB.

The porous absorber (10) was made of a layer of stainless steel wire mesh with 34 meshes per inch. It is cylindrical in shape with 17.7 cm in diameter and 24.7 cm in length. The energy that absorbs at the porous absorber is transferred to the in-coming air (black arrow). $T_{a,o}$, $T_{a,1}$, $T_{a,2}$ and T_{exh} are N-type thermocouple to measure the ambient temperature, air temperature at the porous absorber outlet, the preheated primary air temperature, and the exhaust gas temperature, respectively.

Fig. 3. Photograph of the SRTB.

The upper cover (11) and the lower cover (12) are made of steel sheath of 1 mm in thickness, which protect the heat loss from the heat recirculation section. The porous heat exchanger (13) was constructed such that the ambient air is induced from below to the inlet of the venturi burner (in case of with heat recirculation). In case of without heat recirculation, the porous emitter (8) and the porous absorber (10) are removed.

3. Experimental Results
3.1 Temperature distributions
Fig. 4 shows effect of the tube length on temperature distributions of the SRTB with porous heat exchanger at typical FR = 7.52 kW. Among the three tube lengths, tube L gives the highest combustion
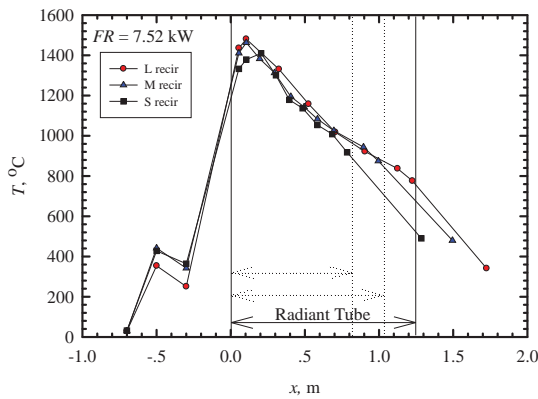

Fig. 4. Temperature distribution.

temperature despite its lowest preheated temperature. This may be attributed to a near stoichiometric combustion for the tube L as will be shown later. The longer the tube length, the more heat loss to the environment, leading to a low exit temperature of the exhaust gas with a decrease in the preheat temperature for tube L. The temperature drop of the last two temperatures at the exit as shown in Fig. 4 for each tube length means part of an enthalpy of the exhaust gas is re-circulated to the primary air by the porous heat exchanger (13), resulting in an increase in the inlet temperature $T_{a,2}$ of the venturi burner.

3.2 Total equivalence ratio ϕ
The total equivalence ratio is defined as the ratio of the fuel-to-oxidizer ratio to the stoichiometric fuel-to-oxidizer ratio and can be calculated from Eq. (1)

$$\phi = \frac{21 - \%O_2}{21} \qquad (1)$$

where $\%O_2$ is the oxygen concentration in the exhaust gas at the exit. The total equivalence ratio ϕ, is used to indicate quantitatively whether a fuel-oxidizer mixture in the combustion zone of the SRTB is rich, lean, or stoichiometric with the secondary air is taken into consideration.

Fig. 5 shows effect of FR on ϕ at the three tube lengths and with or without heat recirculation by the porous heat exchanger. Increasing FR causes to the total equivalence ratio increased in every case, because the higher temperature causes thermal expansion of mixture and increase in its viscosity [7]. In case of with heat recirculation (solid line), the total equivalence ratio for FR of tube L varying from 3.66 to 7.52 kW is near stoichiometry (ϕ =1) as compared to the other tubes. This is a desirable condition for the design of the SRTB where a near stoichiometric combustion is achieved whilst an acceptable preheating effect is maintained. According to this reason, the combustion temperature of tube L is the highest despite its relatively low preheat temperature (Fig. 4). In case of without heat
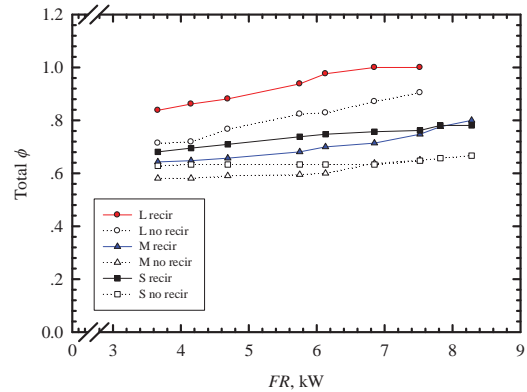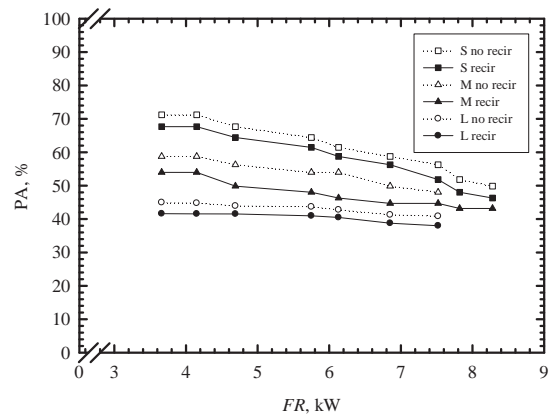

Fig. 5. Total equivalence ratio.

Fig. 6. Primary aeration.

3.3 Primary aeration *PA*

The primary aeration *PA* is used to indicate quantitatively whether a fuel-oxidizer mixture in the venturi burner of the SRTB is rich, lean, or stoichiometric prior to combustion without taking the secondary air into consideration. It is defined as the ratio of the oxidizer-to-fuel ratio to the stoichiometric oxidizer-to-fuel ratio, which could be expressed in the form

$$ PA = \frac{\%O_2}{(A/F)_{stoi} \times (21 - \%O_2)} \times 100 \qquad (2) $$

where $\%O_2$ is the oxygen concentration of the mixture in the venturi burner, $(A/F)_{stoi}$ is the air-fuel ratio of LPG at stoichiometric condition.

Fig. 6 shows variation of *PA* of SRTB in case of with heat recirculation (solid line) and without heat recirculation (dash line). Within the range of *FR* studied, *PA* in all cases is almost linearly decreased with increasing *FR*, due to an increasing of air viscosity [7]. Therefore, the primary air is difficult to be entrained into the venturi burner. In case of with heat recirculation, therefore, all *PA* values are even lower than that of without heat recirculation, because the preheating effect causes expansion of the mixture and an increase in its viscosity [7]. Among the three tubes, tube S gives the highest *PA* because of good entrainment of the secondary air into the relatively short radiant tube with relatively low pressure drop. On the other hand, tube L yields the lowest *PA* because of relatively high viscous flow caused by relatively high combustion temperature with a relatively high pressure drop of the relatively long tube length.

3.4 Emission characteristics

Fig. 7 shows the effect of *FR* on CO emission for the three tube lengths and for the cases with and without heat recirculation. In any cases and tube lengths, increasing *FR* increases the CO emission due probably to a decrease in *PA* that results in incomplete combustion (Fig. 6). It is clear, however, that the CO emissions for the case with heat recirculation are higher than that without heat recirculation, because of relatively low *PA* for the cases with heat recirculation. At any particular *FR*, the CO emission for the case without heat recirculation shows insensitive to the tube length,
whereas an opposite result occurs for the case with heat recirculation. Among the three tubes, it is interesting to note that for the case with
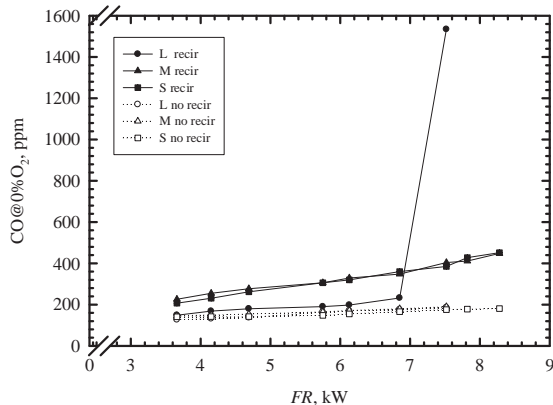
heat recirculation and *FR* of lower than 7.5kW, tube L yields the lowest CO emission despite its relatively low *PA*, because of a long residual time with relatively high temperature for complete combustion. If *FR* is larger than 7.5kW, CO abruptly increases. This may be attributed to a significant decrease in *PA* as explained before.

Fig. 8 shows the corresponding NO_x emission of the SRTB. In any cases and tube lengths, increasing *FR* yields an increase in NO_x emission due probably an increase in the combustion temperature and the reduction in the combustion air. It is considered that, NO_x emission in the present SRTB may arise from the combination of thermal NO_x and prompt NO_x, which are expected to be a major contribution of the measured value according to Sullivan and Kendall [8]. They reported that the thermal NO_x is sensitive to the firing rate whereas the prompt NO_x is much more sensitive to the theoretical air percent. This statement is justifiable for the present SRTB where the firing rate (*FR*) and the theoretical air percent (or total equivalence ratio ϕ) are dependent on each other and are the dominating parameters controlling the combustion characteristics.

At relatively low *FR*, relatively low NO_x emission is observed due probably to relatively low thermal NO_x and prompt NO_x caused by relatively low combustion temperature at relatively lean mixture. On the other hand, a relatively high NO_x emission appears at a relatively high *FR*, due probably to relatively high thermal NO_x and prompt NO_x caused by relatively high combustion temperature at relative rich mixture.

It is also observed that NO_x emission for the case with heat recirculation is higher than that without the heat recirculation particularly at the relatively high *FR*, the same reason as stated above. Among the three tubes, it is interesting to note that, for the case with the heat recirculation and *FR* of lower than 7.5 kW, tube L yields the lowest NO_x emission, the value of which is comparable to that of the cases without the heat recirculation. This may be attributed to the significant reduction in the thermal NO_x caused by less N_2 available at a relatively rich mixture for a relatively large tube length despite its relatively high combustion temperature.
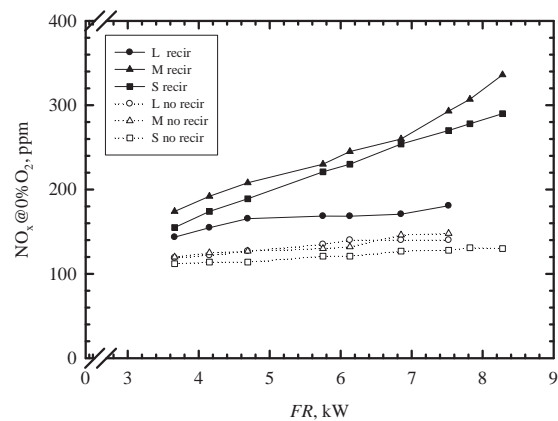

Fig. 8. NO_x emission

3.5 Carnot efficiency

Carnot efficiency is the highest efficiency a heat engine operating between the two thermal energy reservoirs at temperatures T_L and T_H can have, which could be expressed in the form

$$ \eta_{th} = 1 - \frac{T_L}{T_H} \qquad (3) $$

Fig. 7. CO emission.

where T_L and T_H, respectively, are absolute temperatures of exhaust gas temperature (low temperature reservoir) and log-mean temperature of gas temperature at the positions that transfer heat from radiant tube (high temperature reservoir) to the surrounding. The log-mean temperature could be expressed in the form

$$T_H = \frac{T_{max} - T_{ex}}{\ln\left(\dfrac{T_{max}}{T_{ex}}\right)} \qquad (4)$$

where T_{max} is the maximum gas temperature inside the SRTB and T_{ex} is the exhaust gas temperature (for the case without heat recirculation) or the gas temperature at the radiant tube exit before the gas enters the porous emitter (for the case with heat recirculation).

Fig. 9 shows the effect of *FR* on the Carnot efficiency. The result shows that Carnot efficiency in case of with heat recirculation is higher than that without heat recirculation because of heat re-circulating combustion, leading to an increase in the maximum gas temperature. In addition, the exhaust gas temperature decreased due to part of heat from exhaust gas is used to preheat the primary air. Among the three tubes, tube L gives the highest value of Carnot efficiency because it has the highest T_H and the lowest T_L.
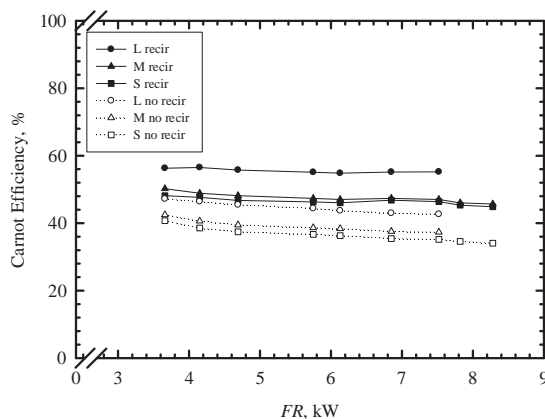

Fig. 9. Carnot efficiency.

4. Conclusions

An experimental study of the effect of the tube length on combustion characteristics of a self-aspirating radiant tube burner with and without heat recirculation has been carried out. Burner performance, combustion characteristic and effect of tube length of SRTB were clarified through measured thermal structure in terms of temperature distribution along the centerline of the SRTB and the emissions at the exit pipeline. The following conclusions can be drawn from the experimental results;

1. An innovative SRTB equipped with the energy recirculation of a porous heat exchanger is successfully developed. It can operate with *FR* in range of 3.66 to 8.28 kW and without flame flashback into the venturi burner.
2. According to the results of temperature distribution, the NO_x and CO emission and the Carnot efficiency, it can be concluded that the tube L is the most suitable length to apply and use in practical applications.
3. The SRTB with the energy recirculation in case of having external thermal load is the topic of the study in the near future.

5.References

[1] Weinberg, F.J. (1971). Combustion temperatures: the future, *Nature*, 233, pp. 239–241.

[2] Katsuki, M. and Hasegawa, T. (1998). The science and technology of combustion in highly preheated air, in: *27th Symposium (International) on Combustion/ The Combustion Institute*, pp. 3135–3146.

[3] Echigo, R. (1982). Effective energy conversion method between enthalpy and thermal radiation and application to industrial furnace, *Proceedings of the 7th Heat Transfer Conference*, Munich, 6, pp. 361–366.

[4] Jugjai, S. and Rungsimuntuchart, N. (2002). High efficiency heat-recirculating domestic gas burners, *Experimental Thermal and Fluid Science*; 26(5), pp. 581–592.

[5] Scribano, G., Solero, G., and Coghe, A. (2006). Pollutant emissions reduction and performance optimization of an industrial radiant tube burner, *Experimental Thermal and Fluid Science*; 30, pp. 605–612.

[6] Chuenchit, C. and Jugjai, S. (2011). Self - aspirating radiant tube burner, paper present *in the 4th International Conference on Sustainable Energy and Environment (SEE 2011)*, Bangkok, Thailand.

[7] Namkhat, A. and Jugjai, S. (2010). Primary air entrainment characteristics for a self-aspirating burner: model and experiments, *Energy*, 35(4), pp. 1701-1708.

[8] Sullivan, J.D. and Kendall, R.M. (1992). Thermal performance and NO_x emissions from porous surface radiant burners, *International Gas Research conference*, pp.1626-1634