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Theoretical Model for Predicting Thermal Efficiency of a Self-Aspirating Burner with Preheated Air

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Abstract

In this work, a novel method of prediction is proposed for the thermal efficiency of self-aspirating burners (high-pressure burner type i.e. KB-5 size). The objective is to present an analytical method to calculate the highly-idealized thermal efficiency ($\eta_{th,cal}$) of burners with preheated air and then compared with some experimental values ($\eta_{th,exp}$) to validate the model. In addition, the effects of firing rate and preheated air temperature on the primary aeration, secondary aeration, total equivalence ratio, adiabatic flame temperature, and thermal efficiency were examined. Moreover, these studies were also performed for both with preheated and non-preheated cases of combustion air. It is found that the model predictions agree with the experimental data. The preheated air case gives about 9.88% higher $\eta_{th,exp}$ than that of the non-preheated case. An increase in the maximum $\eta_{th,exp}$ from 44.08% to 53.96% using the heat-recirculation is observed. It is clearly seen that the thermal efficiency for the theoretical results is greater than that of the measured ones owing to the imposed assumptions such as complete combustion and no heat loss from flame to the environment. Thus, the obtained empirical formula in this work may provide helpful information for the future design, development and improvement of a high-performance, low emission, self-aspirating burner with preheated primary air.

Keywords: Self-aspirating burner, Air entrainment, Preheating effect, Thermal efficiency

1. Introduction

A self-aspirating burner is classified as a partially premixed, atmospheric-pressure Bunsen flame, wherein the combustion air is naturally entrained from the surroundings by momentum sharing between the high velocity jet of gaseous fuel and the surrounding air [1]. Since the self-aspirating burner is convenient to operate and clean to use through emission of partially premixed, blue flame, therefore it is widely adopted for use in the household sector worldwide.

In Thailand, LPG (Liquefied Petroleum Gas) is the only choice for gaseous fuel available for cooking in the household sector. According to the statistics from the years 1986 to 2011 [2], the total LPG consumption in Thailand is increasing continuously. It is found that the cooking sector

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accounts for about 50% of the country's total LPG consumption. Therefore, the higher the performance of the self-aspirating burners being used in the country, the larger the amount of LPG that can be saved by this major sector.

Much research on self-aspirating burners has been carried out to improve thermal performance and to reduce pollutant emissions by developing the burner port geometry and the combustion system [3-5]. However, research on simulate and quantify of self-aspirating burner performance have rarely been reported. Kohli et al. [6] developed an experimental setup and a computer code based on finite difference method to study a single-pan stove. Heat transfer utilization of commonly used Bangladeshi cook stoves, with emphasis placed on natural gas cook stoves, was studied by Lucky and Hossain [7]. Jugjai and Rungsimuntuchart [5] used an analytical method to simulate convection heat transfer through the burning head to the dish; they also conducted some experiments to validate their model. Jeddi et al. [8] developed a numerical code based on finite-element method to model conventional cooking tops. They represented the effect of cooking top parameters on heat transfer from flue gases to the pot.

In this work, a novel method of prediction is presented for the thermal performance of the self-aspirating burners. The objective is to present an analytical method to predict the highly-idealized thermal efficiency of the self-aspirating burners. To validate the accuracy of the proposed theoretical formulation, theoretical and experimental results were compared. These results also performed for both with preheated and non-preheated primary air. In addition, The

effects of firing rate (q) and preheated air temperature ($T_{pre.}$) on the primary aeration (PA), secondary aeration (SA), total equivalence ratio (Φ_{tot}), adiabatic flame temperature (T_b) and thermal efficiency (η_{th}) were examined.

2. Methodology

Fig. 1 shows the schematic of the experimental setup for impinging flame of the Porous Radiant Recirculated Burner (PRRB) [9]. It is composed of a self-aspirating burner (KB-5 size) and heat recirculation system. This burner has a cross-sectional area of injector A_i = 0.64 mm^2, cross-sectional area of throat A_t = 254.47 mm^2, and overall cross-sectional area of the burner port A_p = 245.44 mm^2.

Heat recirculation in the PRRB is applied by porous technology, which have been made on the arrangement of a pair of the porous medium; the emitting porous medium (EP) ⑨ and the absorbing porous medium (AP) ⑩, through which the exhaust gas enthalpy part is recirculated to the primary air (combustion air) by thermal radiation. EP is made by a perforated stainless steel (3 mm-thickness). AP is formed by a stack of four pieces of stainless steel wire net having 100 mesh per inch. An outer and inner housing of heat recirculation system are constructed by a steel plate of thickness 2 mm.

Liquefied petroleum gas (LPG) was used as a fuel in the experiment. The composition of the LPG was 40% (by vol.) propane (C_3H_8) and 60% butane (C_4H_{10}). LPG ① is controlled by a high pressure regulator ② with calibrated by a high pressure flow meter ③. A mercury manometer ④ is used to measure the fuel pressure. Then, the
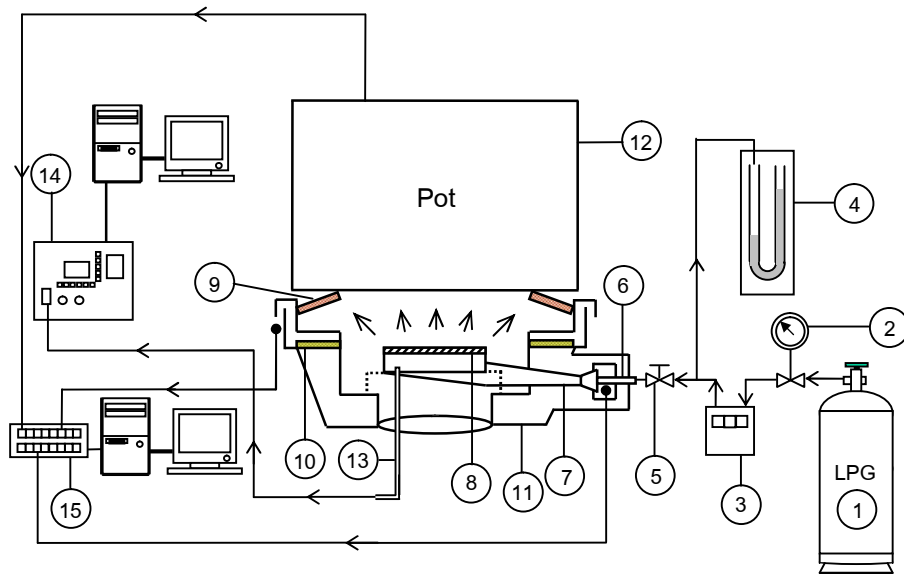
1. Fuel (LPG)
2. Pressure regulator
3. Flow meter
4. Mercury manometer
5. Ball valve
6. Nozzle

7. Mixing tube
8. Multiple port burner
9. Emitting porous medium (EP)
10. Absorbing porous medium (AP)
11. Housing of Porous Radiant Recirculated Burner (PRRB)

12. Flat bottom vessel (Pot)
13. Sampling tube
14. Oxygen sensor
15. Data logger

Fig. 1 Schematic diagram of PRRB (KB-5)

fuel gas emerges from the fuel nozzle ⑥ which connected with a ball valve ⑤ . On leaving the fuel nozzle, the fuel gas entrains the primary air by a momentum-sharing process between the emerging gas and ambient air. This primary ambient air is induced through a gap between outer and inner housing ⑪ that likes an air jacket cooling. After that a cool primary air is heated up by hot AP ⑩ and then preheated primary air is entrained with fuel into the mixing tube ⑦. This mixture is distributed uniformly to the multiple port burner ⑧ with premixed flame. A hot combustion product impinges at a stagnation point of a flat bottom vessel ⑫ . Some of the combustion product flow through EP ⑨ that absorbs a heat from hot combustion product and radiates heat to

AP ⑩ and bottom vessel ⑫ . Water temperatures, T_{water}, were monitored by a K-type sheath thermocouple with a wire diameter of 0.5 mm. Meanwhile, T-type sheath thermocouples are used to measure the ambient temperature T_o and preheated air temperature T_{pre}. The thermocouple signals were digitized by a data logger (Testo model 175-T3) ⑮ , and then transmitted to a personal computer. The oxygen concentration in the mixture is measured by using a sampling gas line ⑬, which is located at the burner head, and then carried to the oxygen sensor ⑭ with an accuracy of about 0.05%. The primary aeration (*PA*) can be calculated using equation (1) [10]

$$PA = \frac{\%O_2}{(A/F)_{stoi.} \times (21 - \%O_2)} \times 100 \quad (1)$$

Thermal efficiency is determined according to the German Standard [11], owing to the former's applicability to varying capacities of the burner thermal output. The efficiency, η_{th}, is defined as the ratio of the sensible heat absorbed by the specified mass of water m_{water}, to raised its temperature from an initial value $T_{water,ini}$ to 90°C, to the combustion heat of the burned gaseous fuel, as expressed by equation (2)

$$\eta_{th,exp} = \frac{m_{water} C_{p,water} (90 - T_{water,ini})}{\dot{V}_g \times LHV \times t} \times 100 \quad (2)$$

Where $C_{p,water}$, $\dot{V}_g$, LHV, and t are specific heat of water at constant pressure, volume flow rate of fuel, low heating value of gas fuel, and time, respectively. The primary aeration and thermal efficiency of self-aspirating burner for both non-preheated and with preheated (PRRB) were compared at various firing rate q.

3. Results and discussion

3.1 Preheated primary air temperature

A preheated air temperature, $T_{pre.}$, is an important factor of the heat recirculation performance [5]. Fig. 2 shows $T_{pre.}$ as a function of firing rate q with preheated and non-preheated
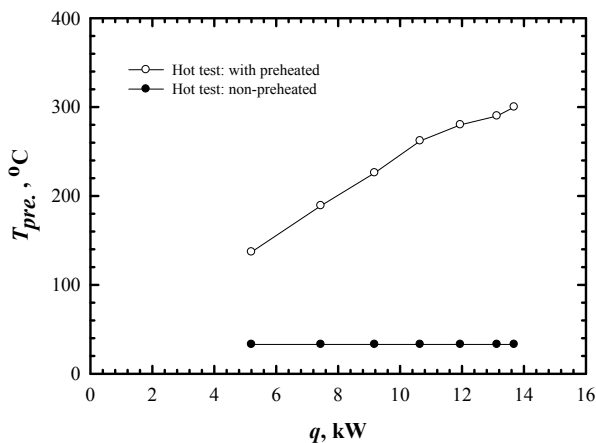
cases. At non-preheated case, $T_{pre.}$ reached certain values as ambient temperature. Whereas, preheated case results provide higher values for $T_{pre.}$ than non-preheated ones. $T_{pre.}$ is increased by increasing of q because of a high heat recirculation performance by high exhaust gas temperature.

3.2 Correlation for primary aeration

The primary aeration for the hot test PA_h can be estimated from the cold test PA_c [12]. Best-fit empirical correlation for PA_h as a function of PA_c and temperature has been determined and proposed with the formula:

$$PA_h = PA_c \left[\frac{T_f}{T_c} \right]^m e^{\lambda b} \quad (3)$$

where T_f and T_c respectively, are adiabatic flame temperature and ambient temperature. In addition, T_f is the flame temperature based on the primary equivalence ratio of the cold test with the primary air preheated. λ is a stage of completion. $\lambda = 0$ means without combustion, whilst with combustion referring $\lambda = 1$. The variable m and b in equation (3) depend on the preheated primary air temperature and can be expressed in the form

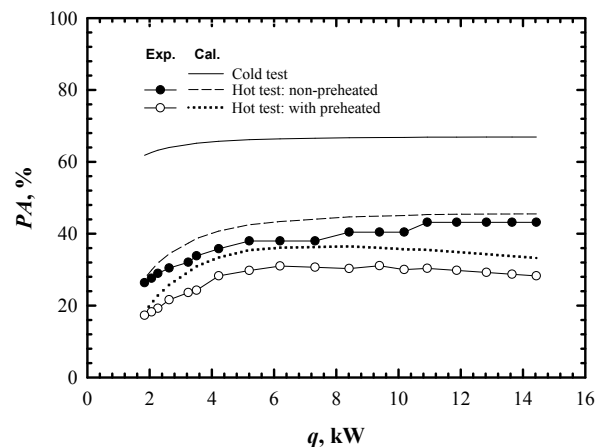Fig. 2 Temperature of primary air preheated

Fig. 3 Primary aeration of a KB-5 burner

$$m = -33.9007 + 2.5498 \left[1 - e^{\left\{ -0.1077 \left(\frac{T_{pre.}}{T_c} \right) \right\}} \right] + 47.7570 \left[1 - e^{\left\{ -2.0468 \left(\frac{T_{pre.}}{T_c} \right) \right\}} \right]$$

$$b = -71.3366 + 16.2385 \left[e^{\left\{ -0.0739 \left(\frac{T_{pre.}}{T_c} \right) \right\}} \right] + 188.1011 \left[e^{\left\{ -2.0392 \left(\frac{T_{pre.}}{T_c} \right) \right\}} \right]$$

Fig. 3 shows a comparison of the corresponding *PA* between theoretical results and measured values, which were performed both the cold test and hot test (with preheat and non-preheat) as a function of *q*. It is observed that the maximum level of *PA* in the cold test is about 66%. On the other hand, the maximum level of *PA* in the hot test is about 45%, which occurred in the non-preheated case. Moreover, the *PA* further decreases with the preheated air case. The experimental results provide slightly lower values for *PA* than theoretical ones. It is important to note that the *PA* in the hot test with preheated air is decreased by increasing of *q* because of a high preheating effect. For complete combustion, a sufficient supply of the secondary air from the surroundings into the flame is of importance to compensate for the reduction in the *PA*.

3.3 Correlation for secondary aeration

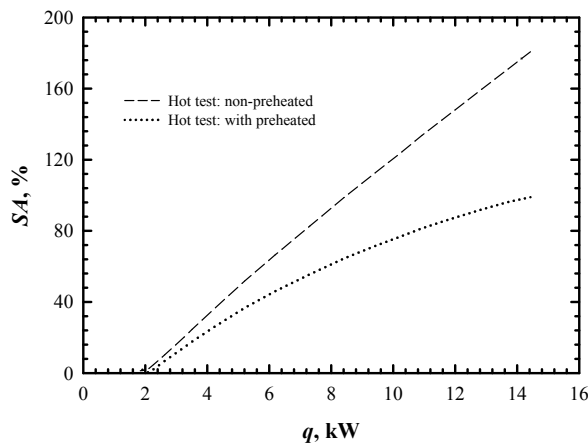Ricou and Spalding [13] presented an experimental technique for measuring the

Fig. 4 Secondary aeration of a KB-5 burner

secondary air entrainment in the turbulent free jets. A correlation for secondary air entrainment has been determined and proposed with the formula:

$$\frac{\dot{m}_m + \dot{m}_{sa}}{\dot{m}_m} = 0.32 \left(\frac{\rho_{sa}}{\rho_m} \right)^{0.5} \frac{h_f}{D_p} \qquad (4)$$

As shown in equation (4), the mass flow rate of secondary air $\dot{m}_{sa}$ is a function of the mass flow rate of mixture ($\dot{m}_m$), density of secondary air (ρ_{sa}), density of mixture (ρ_m), flame height (h_f), and diameter of burner port (D_p).

Since $\dot{V}_{sa} = \dot{m}_{sa} / \rho_{sa}$, from equation (4), the volume flow rate of secondary air is given by

$$\dot{V}_{sa} = \frac{\dot{m}_m}{\rho_{sa}} \left[\left(0.32 \left(\frac{\rho_{sa}}{\rho_m} \right)^{0.5} \frac{h_f}{D_p} \right) - 1 \right] \qquad (5)$$

As a result, the secondary air entrainment (or secondary aeration) is expressed in the form

$$SA = \frac{\dot{V}_{sa} / \dot{V}_g}{(A/F)_{stoi.}} \times 100 \qquad (6)$$

where $\dot{V}_g$ and $(A/F)_{stoi.}$ respectively, are the volume flow rate of gas fuel and stoichiometric air-fuel ratio.

Fig. 4 shows typical secondary aeration (*SA*) of a self-aspirating burner obtained using equation (6). It was found that the levels of SA for both with preheated and non-preheated cases increase with firing rate q, because of a strong momentum effect caused by the high volume flow rate of the mixture as q increases. It is clearly seen that the *SA* for the non-preheated case is greater than that of the preheated case. Moreover, the difference *SA* values between in both cases at maximum firing rate is increased because the high preheating effect causes low

primary aeration and a decrease in the momentum force of mixture.

3.4 Total equivalence ratio

The total equivalence ratio is commonly used to indicate quantitatively whether a fuel-oxidizer mixture is rich, lean, or stoichiometric. The total equivalence ratio is defined as

$$\Phi_{tot} = \frac{100}{(PA + SA)} \qquad (7)$$

Fig. 5 shows typical total equivalence ratio (Φ_{tot}) of a self-aspirating burner obtained using equation (7). It was found that the levels of total equivalence ratio are in the range between rich to lean conditions. The rich condition is occurred at lower firing rate q because of the low total air entrainment. At higher firing rate q, the high momentum jet provides an excess air. Therefore, the total equivalence ratio becomes very low under lean conditions. The optimum operation of this burner occurs at firing rate which is indicative of stoichiometric combustion.

3.5 Adiabatic flame temperature

The obtained adiabatic flame temperature is calculated for LPG using constant pressure by chemical equilibrium principle [14].
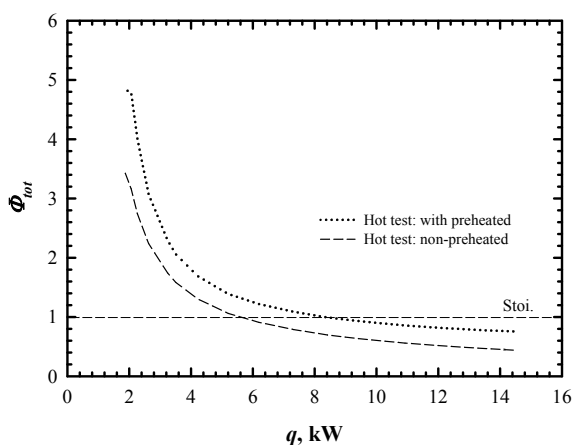
Fig. 6 shows typical adiabatic flame temperature (T_b) of a self-aspirating burner for both with preheated and non-preheated cases as a function of firing rate q. Both cases show similar trends. It was found that the levels of T_b are the maximum at firing rate which is indicative of stoichiometric combustion. The maximum level of T_b in the non-preheated case is about 2283 K at q = 5.2 kW. Whereas, the preheated air case gives the maximum T_b value is about 2304 K at q = 7.31 kW. As is clearly seen, the preheating effect causes the level of total equivalence ratio and initial temperature of mixture. The difference T_b values between in both cases at maximum firing rate as $q = 14$ kW is increased because of the high preheating effect as q increases. Thus, the T_b for the preheated case is greater than that of the non-preheated case at high q.

3.6 Thermal efficiency

The thermal efficiency of the combustion system is studied based on the Carnot efficiency and heat exchanger principles. Assuming that the top temperature is not liked the Carnot system because the temperature gradient is not constant. Therefore, the maximum thermal efficiency of
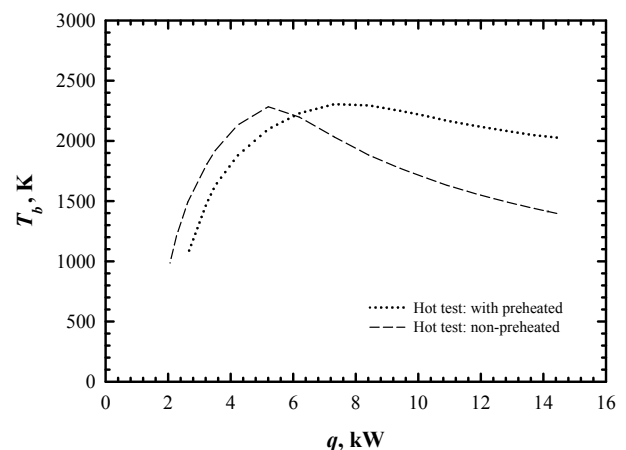

Fig. 5 Total equivalence ratio of a KB-5 burner

Fig. 6 Adiabatic flame temperature of a burner

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systems without and with heat recirculation in extremely case can be calculated using equations (8) and (9), respectively [15].

$$\eta_{th} = 1 - \left[\frac{T_o}{(T_b - T_o)} \right] \ln \frac{T_b}{T_o} \qquad (8)$$

$$\eta'_{th} = 1 - \left[\frac{T_o}{(T'_b - T_{pre})} \right] \ln \frac{T'_b}{T_{pre}} \qquad (9)$$

From equation (9), an expression can be written as

$$\eta'_{th,cal} = 1 - \left[\frac{T_o}{(T'_b - T_o)(1-r)} \right] \ln \frac{T'_b}{T_o + r(T'_b - T_o)} \qquad (10)$$

Where r is the fraction of heat recirculated which defined as the ratio of the heat of preheated to the total heat to used, as given by

$$r = \frac{T_{pre} - T_o}{T'_b - T_o} \qquad (11)$$

As shown in equation (11), at $r = 0$ means without heat recirculation ($T'_b = T_b$) and then equation (10) is expressed in the form of equation (8). Whereas at $r > 0$ refers to with heat recirculation. If we recover the total heat to used

Fig. 7 Thermal efficiency of a self-aspirating burner (KB-5)

and feed it back to preheat the reactants, it obtained the maximum fraction of heat recirculated ($r = 1$).

Fig. 7 shows a comparison of thermal efficiency η_{th} between theoretical results and measured values for both with preheated and non-preheated cases. It was found that the preheated air case gives about 9.88% higher $\eta_{th,exp}$ than that of the non-preheated cases. An increase in the maximum $\eta_{th,exp}$ from 44.08% to 53.96% using the heat-recirculation is observed. It is clearly seen that the η_{th} for the theoretical results is greater than that of the measured ones owing to the imposed assumptions such as complete combustion and no heat loss from flame to the environment. The difference in the measured η_{th} value and the theoretically estimated one is significance. As a result, it was decreased by the utmost requirement in improving thermal efficiency of burner in the future.

4. Conclusions

1. The preheated case gives a lower PA and SA values than that of the without preheat case, because the preheating effect causes expansion of the mixture and an increase in its viscosity.

2. The thermal efficiency for the preheated case is greater than that of the non-preheated case because of the preheating effect.

3. The obtained empirical formula may provide helpful information for the future design, development and improvement of high-performance, low emission, self-aspirating burners with preheated primary air.
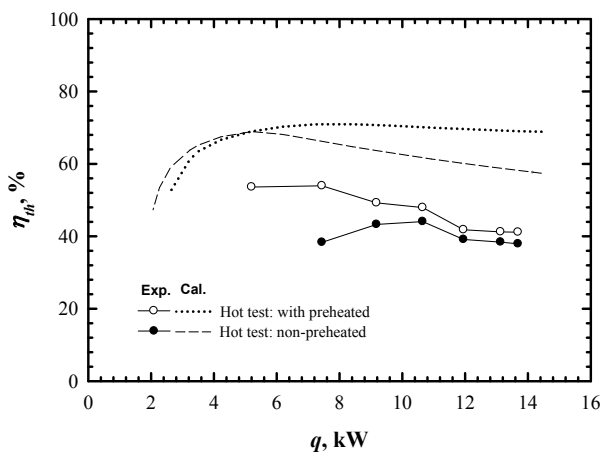
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