Engineering and Applied Science Research 2022;49(3):316-326 Research Article

Engineering and Applied Science Research
https://www.tci-thaijo.org/index.php/easr/index
Published by the Faculty of Engineering, Khon Kaen University, Thailand

Design and development of natural-draft Ethanol/LPG burner for heavy-duty cooking

Ativis Sirivichaikum[1)], Usa Makmool Grare*[2)] and Sumrerng Jugjai[1)]

[1)]Department of Mechanical Engineering, Faculty of Engineering, King Mongkut's University of Technology Thonburi, Thung Kru, Bangkok 10140, Thailand
[2)]Pilot Plant Development and Training Institute (PDTI), King Mongkut's University of Technology Thonburi (Bangkuntien), Bangkuntien, Bangkok 10150, Thailand

Received 15 March 2021
Revised 15 June 2021
Accepted 30 July 2021

Abstract

A natural-draft, heavy-duty cooking burner was designed and developed as a fuel-flexible burner capable of burning both liquefied petroleum gas (LPG) and Ethanol. The burner has great potential to replace LPG with ethanol to reduce the consumption of fossil fuels. Ethanol is a renewable fuel; it can be produced domestically from agricultural products. However, its heating value is lower than that of LPG. Thanks to the advantage of combustion in a porous medium to create heat recirculation, the flame temperature is higher than that of the free flame. Using a column embedded in the burner wall, liquid ethanol is vaporized without external heating and atomization. A critical criterion for the burner is to stabilize the flame inside the porous medium so that flow velocity equals burning velocity. A common gas injector can be used to switch between LPG and ethanol while maintaining better combustion and thermal performance compared to a conventional LPG burner and previous ethanol burners. The proposed burner has a compact combustion chamber, but it yields a higher combustion temperature because it stabilizes the flame at greater amounts of primary air with a high turn-down ratio.

Keywords: Heavy duty burner, Ethanol burner, Fuel-flexible, Porous medium burner, Natural-draft burner

1. Introduction

Gas burners are popular and used worldwide for heating and cooking in households, in the food industry, and in other small and medium enterprises. Pipeline gas or natural gas burners are commonly used in the European Union (EU) and other developed countries, whereas liquefied natural gas (LNG) and liquefied petroleum gas (LPG)-which are portable sources of energy-have been used in gas burners in developing countries and in remote areas of developed countries. LPG is liquid under pressure and stored in pressurized containers. This makes it much more cost-efficient to transport over long distances. The reliability of LPG burners is generally considered high because the technology is mature and applied widely across the world. However, LPG is a fossil fuel and is non-renewable energy. In 2018, fossil fuels provide 80 % of the total energy used. Thus, they are the dominant source of local air pollution, causing an increase in CO_2 emissions and other greenhouse gases. In Thailand, the government has subsidized the price of LPG consumed by the household sector which is the highest LPG consumption sector [1].

Sustainable energy usage-not only using energy more efficiently but also using an alternative energy source-is thus the way to reduce fossil fuel consumption and CO_2 emissions. Therefore, using ethanol instead of LPG is proposed because ethanol is produced domestically, and it is renewable. In Thailand, ethanol was mainly produced from molasses and cassava, more than 1,450 million litres or 4.0 million litres/day in 2020. This is expected to reach 5.8 million litres/day by 2036 [2]. Currently, ethanol is more expensive than LPG with respect to heating value. Ethanol (99% purity) is approximately 3.43 baht/kW (0.10 US Dollar/kW), whereas LPG is 2.54 baht/kW (0.08 US Dollar/kW). However, the prices of the two fuels are not stable; they depend on market demand and supply. Also, ethanol at ambient conditions is a liquid. This makes it easy to store and transport, and it has a relatively high mass-specific energy content. Ethanol has been successfully used as a blending fuel in vehicles for both diesel engines [3, 4] and gasoline engines [5, 6]. Unfortunately, there are few studies of the direct use of ethanol in an open burner for heating and cooking in households. Ethanol has been used as clean cooking fuel by modifying an existing pressurised kerosene stove [7]. However, they are not widely used in households, rather they are generally used for remote areas and leisure activities. Because of its high flammability and low viscosity, ethanol used mostly for cooking is mixed with a thickening agent to become ethanol gel for more safety [8, 9]. One of the most documented liquid ethanol cookstoves is developed for Project Gaia in Ethiopia [10]. This stove was invented by Bengt Ebbenson in 1979. Its flame features are like those of an LPG stove. It has a maximum power of 1.8 kW, which is a small heating power purpose. Nevertheless, compared with LPG, this ethanol stove released higher emissions of CO and was slower to boil water [11]. Choi et al. [12] designed a burner with a capacity of 35 kW as a pilot scale for the combustion of bio-crude-oil (BCO) and ethanol (EtOH) blends. However, an air atomizing spray nozzle with a larger fuel orifice was adopted to keep the nozzle from clogging.

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*Corresponding author. Tel.: +66 470 7429
Email address: usa.mak@kmutt.ac.th

doi: 10.14456/easr.2022.33

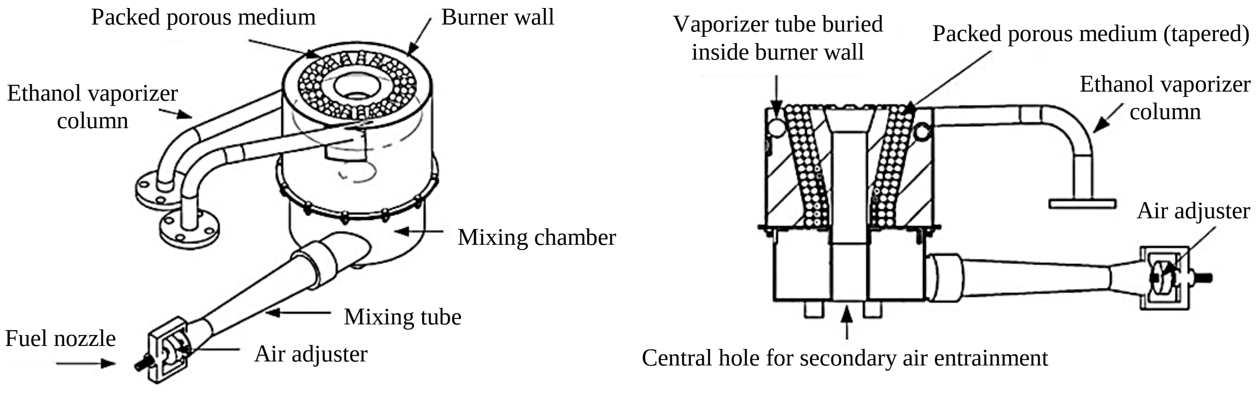
It has been shown that there are practical benefits of combustion in a porous medium, which is a promising technique for fuels with low heating value [13]. As compared with a traditional Bunsen flame burner, the operating range, turndown ratio, and pollution emissions of the porous medium burner were greater, especially for fuel-lean combustion [14, 15]. Heat is recirculated from the exhaust gas to heat fresh reactants, resulting in a higher temperature than an adiabatic temperature [16]. Porous medium combustion (PMC) has potentially low NO_X and CO emissions compared with a conventional open flame burner [17, 18]. Unburned CH and CO emissions are low because of the fuel preheating in the porous medium [19]. For the LPG cooking stove, it was reported that the burner with PMC technology, named the two-layer porous radiant burner (PRB), has a maximum thermal efficiency of 10% approximately higher than that of the conventional burner (CB) for the same power input range of 1-3 kW [20]. Moreover, it has the potential for performance improvement of the PRB by optimizing burner size. A cluster porous radiant burner consisting of 3 smaller-size individual burners is more efficient than a single-burner [21]. Besides, the results of life cycle assessment (LCA) and techno-economic assessment (TEA) of medium-scale (5-10 kW) LPG cooking stove indicated that the LPG stove with PRB is a better choice for the replacement of the CB due to its economic and environmental superiority [22]. The feasibility study of using PMC technology for combustion of raw biogas, which is a low calorific value gas, has been carried out [23]. The newly developed sideway faced porous radiant burner (SFPRB) that operates with the input powers of 5-10 kW was capable of providing efficient combustion of raw biogas in the lean fuel-air mixture range of 0.75-0.97. The ability of PRB to combust waste cooking oil (WCO) by blending with kerosene has been proven to improve the burner performances in terms of saving energy and emissions [24]. The ethanol porous medium burner capacity of 18 to 23 kW was proposed by Makmool et al. [25]. However, the thermal efficiency is less than 40% at the maximum combustion temperature of 1,200 ^{o}C approximately. The reason for low thermal efficiency might be heat loss and with the structure of the burner, which is quite large. In addition, insufficient air was supposed to be a reason for the lower combustion temperature, as compared to LPG flame. Increasing air entrainment results in a high risk of backfire or flame flashback to the burner nozzle because the burning velocity increases faster than the mixture flow velocity.

For these reasons, the aim of this project is to develop a heavy-duty cooking burner which can burn both LPG and ethanol using the same fuel orifice. This would improve flame stability and the burner's thermal performance. A compact burner that can burn well and stabilize the flames with larger amounts of air entrainment is proposed. This burner is natural-draft; it operates without an air compressor. Not only does this burner provide flexibility in terms of burning both gaseous LPG and liquid ethanol, but this innovation will also help to reduce fossil fuel consumption and reduce the impact of climate change because CO_2 emitted from biofuel is a carbon neutral.

2. Methodology

2.1 Conceptual design of the burner

The critical criterion for designing a porous-medium burner is the stabilisation of the flame inside the porous packed bed where the Peclet number (*Pe*) must greater than 65 [26], conversely, the flame cannot be propagated and maintained inside the pore because of the quenching effect. To stabilise the flame, the flow velocity of the mixture must equal the burning velocity (S_L) at the given location. Otherwise, instability, i.e., flame blowing out or flame flashback (when flow velocity is greater than burning velocity or vice versa), will occur. If the equivalence ratio (f) at the designed firing rate is known, the flow velocity of the mixture in the porous packed bed (interstitial velocity, $u_{m,in}$) can be determined.



(a) (b)

Figure 1 Design structure of the burner (a) full view (b) sectioned view

To design the burner, the initial f and S_L were evaluated based on a previous experiment on an LPG porous-medium burner [27], where the burning velocity was greater than a conventional LPG burner. The gas nozzle diameter is reduced from 1.2 mm to 1.0 mm to improve air entrainment by increasing the fuel jet velocity at the same volumetric gas flow rate. For an atmospheric porous-medium burner (PMB) using LPG (C_3H_8:C_4H_{10} = 40:60) as fuel, f can be predicted using a quadratic equation adapted from Addamane et al. [28] to become as Eq. (1) to Eq. (7). Note that for the atmospheric gas burner, where air entrainment is a natural draft, the primary air entrainment is only 50-60% of the air required for complete combustion. Therefore, to optimise air entrainment, LPG was used in the design of the proposed burner because it has a natural rich burn (f > 1). Thus, to stabilise the flame of ethanol (which is an oxygenate and requires less air than LPG for complete combustion), an air adjustor is installed at the entrance of the mixing tube (see Figure 1). It controls the amount of primary air needed to balance the burning velocity and the mixture flow velocity.

$$\phi^2 X + \phi Y + Z = 0 \tag{1}$$

Thus,
$$\phi = \frac{-Y \pm \sqrt{Y^2 + XY}}{2X} \tag{2}$$

where *X*, *Y*, and *Z* are given by, respectively:

$$X = \left[\rho_s \left[\frac{1}{A_t A_i} - \frac{1}{2A_b^2} \left(\frac{T_h}{T_l} \right)^2 - \frac{(1+K_{eff})}{2A_t^2} - N \right] - \frac{M}{Q_g} \right] \tag{3}$$

$$Y = -28.08 \left[(\rho_a + \rho_g) \left[\frac{(1+K_{eff})}{2A_t^2} + \frac{1}{2A_b^2} \left(\frac{T_h}{T_l} \right)^2 + N \right] - \frac{M}{Q_g} \right] \tag{4}$$

$$Z = \left(-28.08\right)^2 \left[\rho_a \left[\frac{1}{2A_b^2} \left(\frac{T_h}{T_l} \right)^2 + \frac{(1+K_{eff})}{2A_t^2} + N \right] \right] \tag{5}$$

and M and N are determined by, respectively:

$$M = \frac{150 \mu L C}{d_p^2} \left[\frac{\left(1-\bar{\varepsilon}\right)^2}{\bar{\varepsilon}^3} \right] \left(\frac{T_m}{T_l} \right)^2 \frac{1}{A_b} \tag{6}$$

$$N = \frac{1.75 L C^2}{d_p A_b^2} \left[\frac{\left(1-\bar{\varepsilon}\right)}{\bar{\varepsilon}^3} \right] \left(\frac{T_m}{T_l} \right)^2 \tag{7}$$

It is shown that *f* is a function of several important parameters: the geometry of the packed bed and mixing tube, temperature, firing rate, and type of gaseous fuel. Once *f* is known, $u_{m,in}$ can be calculated as follows:

$$\dot{V}_m = \dot{V}_f + \dot{V}_a \tag{8}$$

where,

$$\dot{V}_a = \frac{\left(\dfrac{\dot{V}_a}{\dot{V}_f} \right)_{stoi} \times \dot{V}_f}{\phi} \tag{9}$$

$$u_{m,in} = \frac{u_{m,\mathrm{sup}}}{\bar{\varepsilon}} = \frac{\dot{V}_m}{A_b \times \bar{\varepsilon}} \tag{10}$$

At the flame location, $S_L = u_{m,in}$

Conversely, as S_L is known, A_b, which is a cross-sectional area of the packed bed at the designed *FR* and the predicted, *f* is determined by

$$A_b \frac{\dot{V}_m}{\bar{\varepsilon} \times u_{m,in}} \tag{10.1}$$

In addition, to improve flame stabilisation and prevent either flashback or blow off owing to a variation of mixture flow velocity (because of the variation of either *f* or *FR*), a conical packed bed was used. A more detailed explanation of the calculation is given in Chaelek et al. [27].

Figure1 shows the overall structure (Figure 1(a)) and cross-section (Figure 1(b)) of the burner. The porous packed bed with an annular cone shape was made of 10 mm diameter aluminium oxide (Al_2O_3) spheres. The packed bed length, *L*, is 143 mm, the average outer and inner diameters of the burner are 135 mm and 75 mm, respectively. The central hole was made to enhance secondary air entrainment for more complete combustion outside the porous packed bed. The combustion chamber wall is made of 3 mm steel, and it is surrounded by refractory bonding mortar resistant to high temperature, i.e., to 1700 °C. A commercial mixing tube, 251 mm long with a throat diameter of 27.5 mm, and a gas injector with 1.0 mm diameter were used. These are the dimensions of a conventional gas burner (CB).

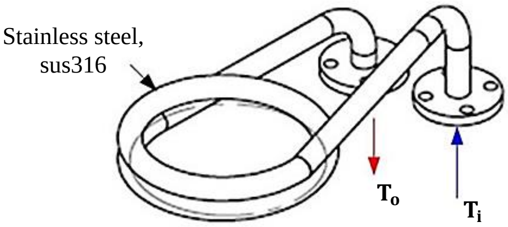


Figure 2 Ethanol vaporizer column

2.2 Ethanol vaporizer

In the proposed burner, ethanol is self-vaporized without atomisation or an external heating system. The liquid ethanol is heated inside the burner to become a superheated vapour before being injected through a conventional gas nozzle. A part of the vaporizer column (Figure 2) is twisted around the burner and embedded in the burner wall. The vaporizer tube touches the combustion zone (see Figure 1(b)) for effective heat transfer from the combustion zone directly to the tube. The vaporizer tube is made of Stainless steel sus316 with an internal diameter (D_i) of 15.8 mm and an external diameter (D_o) of 21.9 mm. The tube length is 1.0 m, which was determined from energy balance equations to ensure vaporization is complete before the ethanol is injected through the gas injector. The minimum heat required to evaporate the liquid ethanol can be determined by:

$$\dot{Q}_{\text{EtOH,min}} = \dot{Q}_{\text{EtOH,sensible heat}} + \dot{Q}_{\text{EtOH,latent heat}} \tag{11}$$

$$= \left[\dot{m}_{\text{EtOH}} \times C_{P,T_{\text{avg}},\text{EtOH}_{\text{liq}}} \times (T_{\text{boil}} - T_{\text{in}})\right] + \left[\dot{m}_{\text{EtOH}} \times L_{T_{\text{avg}}}\right] \tag{11.1}$$

The amount of heat conducted through the burner wall to heat the vaporizer can be expressed as

$$\dot{Q} = sk(T_b - T_s) \tag{12}$$

where T_b is wall temperature of the burner, estimated from the experimental data, T_s is the temperature of the outer surface of vaporizer, while s is the conduction shape factor for a horizontal isothermal cylinder of length L embedded in a semi-finite medium and it is determined by:

$$s = \frac{2\pi L}{\cosh^{-1}\left(\frac{2z}{D_o}\right)} \tag{13}$$

For T_s constant, the heat transfer rate to a fluid flowing through a tube can be expressed as:

$$\dot{Q} = UA_s \Delta T_{lm} \tag{14}$$

where U is the overall heat transfer coefficient and A_S is the surface area that is equal to $\pi D_o L$. For a circular pipe of length L, and ΔT_{lm} is expressed as:

$$\Delta T_{lm} = \frac{(T_i - T_o)}{\ln\left[(T_s - T_o)/(T_s - T_i)\right]} \tag{15}$$

and thus,

$$\dot{Q} = \pi h D_o L \left[\frac{(T_i - T_o)}{\ln\left[(T_s - T_o)/(T_s - T_i)\right]}\right] \tag{16}$$

T_i and T_o, respectively are the inlet and the outlet temperature of ethanol, which is restricted to not exceed 400 °C to avoid fuel pyrolysis clogging the vaporizer tube. Because the amounts of heat from Eq. 12 and Eq. 14 are balanced as:

$$\dot{Q} = sk(T_b - T_s) = UA_s \Delta T_{lm} \tag{17}$$

Therefore, T_s and L can be determined. This ensures that the heat received through the vaporizer is enough for complete vaporization (converting liquid ethanol to superheated vapour) before being injected into the burner at the designated firing rate.

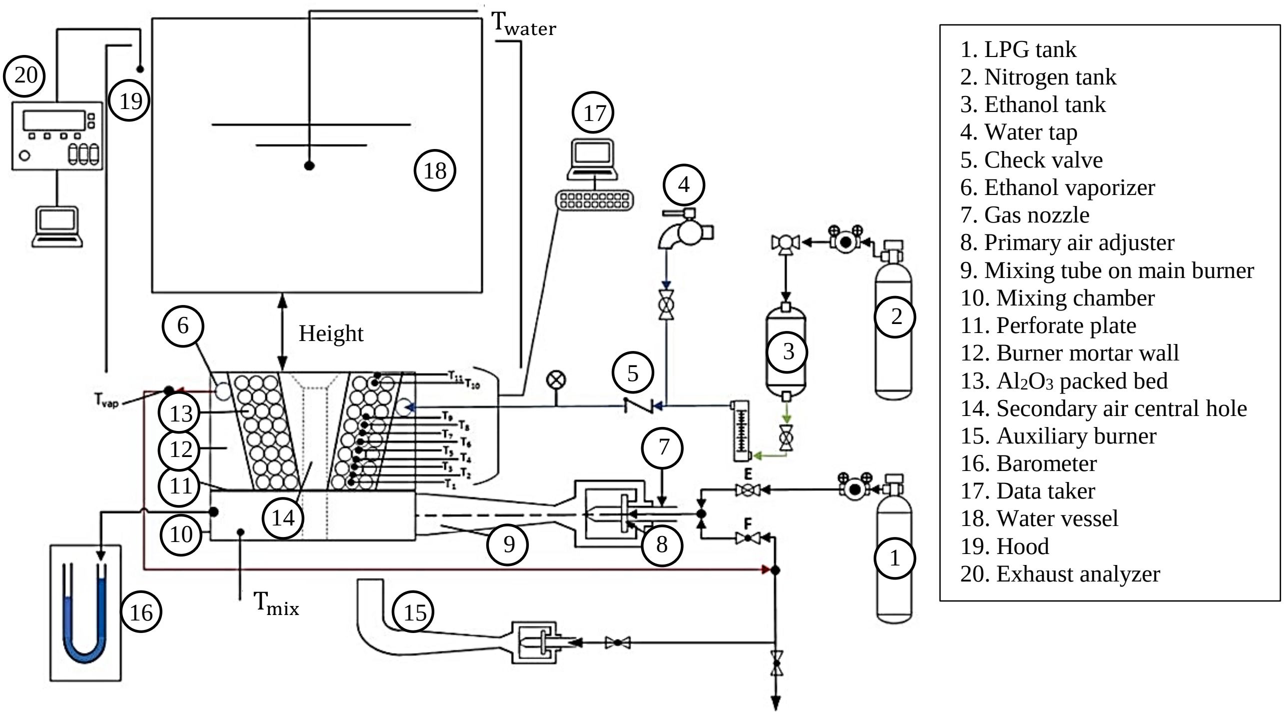


Figure 3 Schematic diagram of the experimental set up

2.3 Experimental setup and testing procedure

Figure 3 shows the experimental setup and the details of the system of the burner. The fuel supply consists of LPG (1) and ethanol supply systems (3). They are independently controlled, but they are injected through a common gas injector (7). At the outlet of the vaporizer, an auxiliary burner (15) was installed to ensure complete vaporization is achieved before being switched to the main burner (9). The liquid ethanol was pressurized using nitrogen (2) at the maximum gauge pressure of 4 bar and then supplied to be heated inside the vaporizer column (6). A check valve or non-return valve (5) was mounted after the flow meter to prevent the ethanol from reversing flow. Thus, the ethanol vapor is injected through the gas injector which is a common nozzle as used for LPG. Note that, without the flow of ethanol during the burning of LPG, the tap water (4) at minimum flow rate was supplied for cooling the vaporizer column to prevent the vaporizer tube from melting. This also helps keep the flames located at the upstream of the vaporizer tube because of the quenching effect. The burner was ignited at the upper surface of the porous medium packed bed (13) using a diffusion flame of LPG at a specific firing rate, and the primary air adjustor (8) was slowly opened to allow flame flashback and stabilized inside the packed bed. The fuel supply was switched to liquid ethanol at the same firing rate; ethanol combustion took place at the auxiliary burner. Once a steady combustion of ethanol was observed at the auxiliary burner, the LPG at the main burner was turned off and switched to the ethanol vapor.

Thermocouples N-type (measuring range 0-1,260 °C ±2.2 °C) were installed inside the burner to measure temperature of the pre-flame zone denoted by T_1-T_3, While thermocouples B-type (measuring range 600-1,650 °C ±0.5°C) were used to measure temperature at the combustion zone and the post-flame location denoted by T_4-T_{11}. T_1 is located 10 cm above the entrance of the packed bed (i.e., X= 0.01 m.), and T_2 to T_{11} were placed at increments of 10 mm along the burner height. The tips of thermocouples were located in the middle of the packed bed between the inner combustion chamber wall and the secondary air inlet wall. In addition, to measure the temperature inside the mixing chamber (T_{mix}) and at the exit of the vaporizer tube (T_{vap}), N-type thermocouples were used. The DATA LOGGER (EQ 600) was used to acquire the temperatures.

A water-boiling test rig and pollutant emission measurement system were set up for testing the burner's performance. The effect of the firing rate (*FR*) on temperature distribution inside the porous burner and thermal efficiency and pollutants emission were investigated and compared. Thermal efficiency and CO and NOx emissions were performed based on the Thailand industry standard for a gas cooking stove [29]. The amount of heat transferring from flames to raise the temperature of water in the pot from ambient temperature to 90 °C was determined. The 100 kg water was filled in a stainless-steel vessel with a diameter of 920 mm. The thermal efficiency, η_{th}, is given by:

$$\eta_{th} = \frac{m_{\text{water}} c_{\text{p,water}} (90 - T_{\text{water,i}})}{\dot{m}_{fuel} LHV \times t} \tag{18}$$

Note that in the case of LPG, the heat lost to the cooling water flowing through the vaporizer was taken into account in the evaluation of thermal efficiency.

For measurements of CO and NO_X emissions, a hood for collecting a flue gas covered the vessel filled with water. The concentrations of CO, CO_2, and NO_X were measured at 15 minutes after ignition by using a portable exhaust analyser (Messtechnik Eheim model Visit01L). The measuring range is 0-10,000 ppm for CO and 0-4,000 ppm for NO_X with a measuring accuracy of ±5 ppm and a resolution of 1 ppm. The emissions reported in this paper are corrected to 0% excess oxygen and dry basis.

3. Results and discussion

3.1 Combustion characteristics of LPG and ethanol

An investigation of combustion within the burner was started using LPG as the fuel. Temperature profiles inside the burner as a function of firing rate (*FR*) are shown in Figure 4. To study flame stabilisation of LPG, *FR* was varied from 21.6 kW to 32.7 kW. As the *FR* increased, flame location was moved downstream, the maximum temperatures and the temperature at the exit of the porous packed bed increased.

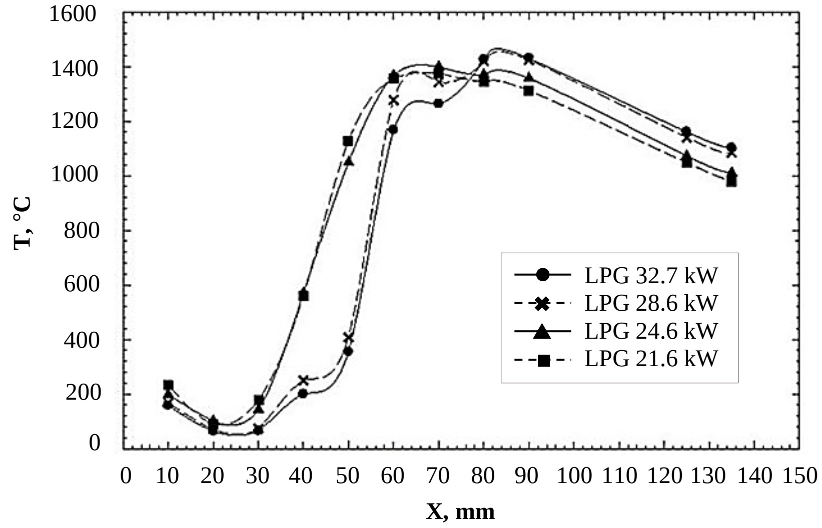


Figure 4 Temperature profiles of LPG burned in the burner

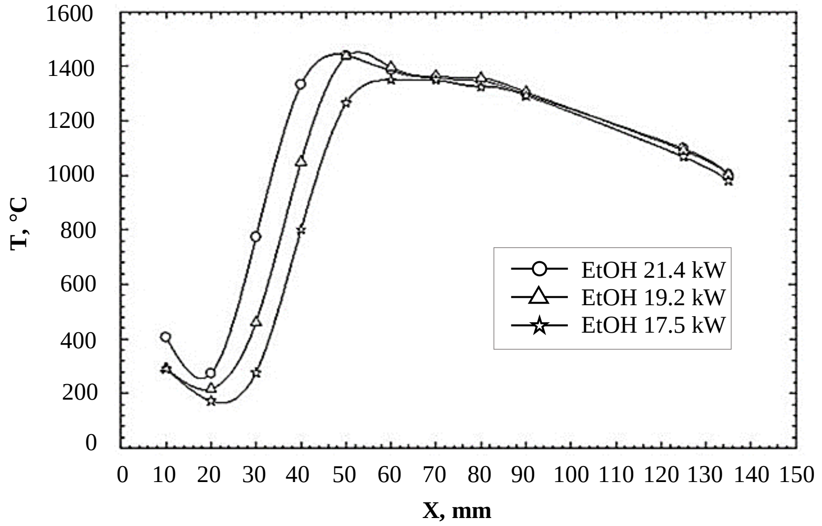


Figure 5 Temperature profiles of Ethanol burned in the burner

The flame apparently shifts downstream when increasing *FR* from 24 kW to 28.6 kW. However, the temperatures peak in the middle region of the porous packed bed (i.e., *X* = 60-90 cm or X/L = 0.42-0.63). This shows that the burner is well designed for high combustion stability at the considered *FR*s. The maximum temperature of 1,430 °C was achieved at *FR* of 32.7 kW. In addition, to prevent the burner from damage due to excessive heat absorption at the high operating temperature, experiments at *FR* higher than 32.7 kW were not performed.

Figure 5 shows temperature profiles of the ethanol combustion at *FRs* varying from 17.5 to 21.4 kW. Note that because the ethanol vapour was injected through the same nozzle as that used for LPG, therefore, at the same fuel mass flow rate, a lower firing rate was supplied compared with that of LPG (Figure 6). This is because ethanol has a heating value (i.e., approx. 27 MJ/kg) lower than that of LPG (i.e., approx. 46 MJ/kg). In addition, the corresponding pressure inside the vaporizer column of ethanol was quite high, and it increased as *FR* increased (i.e., varied from 2.5 bar to 3.7 bar as *FR* increased from 17.5 kW to 21.4 kW). Therefore, for safety reasons, experiments at *FR* higher than 21.4 kW were not performed.

Unlike LPG combustion where combustion was best at the middle of the porous packed bed, the ethanol flames took place earlier at X ~ 50 cm or X/L ~ 0.34. Moreover, increasing the *FR* of ethanol affected flame stabilisation by moving the flame location (i.e., peak temperature location) upstream. Because ethanol is an alcohol in which molecules contain oxygen (C_2H_5OH), increasing *FR* increases the number of oxygen molecules and it increases air entrainment because of an increase in fuel jet momentum. This makes ethanol combust at leaner fuel conditions (at the same amount of entrained air) as compared with that of LPG, as shown in Figure 7.

Moreover, the maximum temperature ($T_{max.}$) of ethanol combustion reached as high as 1,440 ^oC even though it was burning at lower *FR* compared with that of LPG (Figure 8). In other words, because ethanol contains oxygen atoms, which join with oxygen molecules in the air to help ethanol burn more completely. The maximum combustion temperatures of both LPG and ethanol, as shown in Figure 8, correspond well to the variation of the equivalence ratio as shown in Figure 7. Increasing *FR*, *f* decreases to near stoichiometric condition (*f* = 1), and thus increases the maximum combustion temperature.

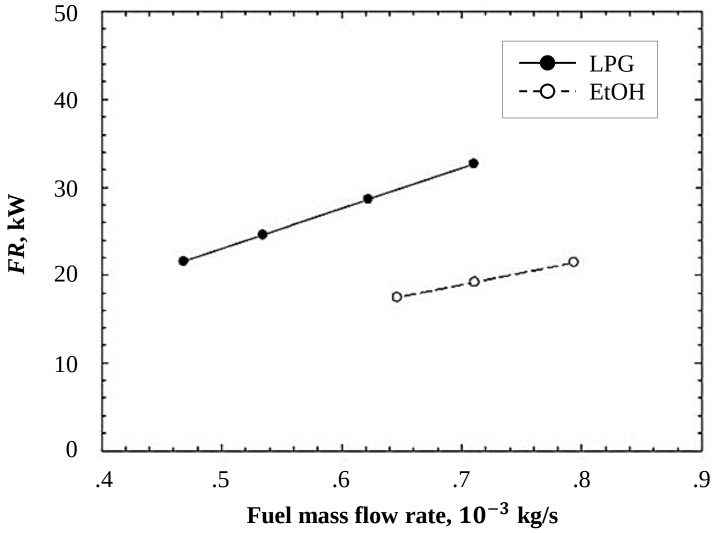


Figure 6 Firing rates of LPG and Ethanol as a function of fuel mass flow rate

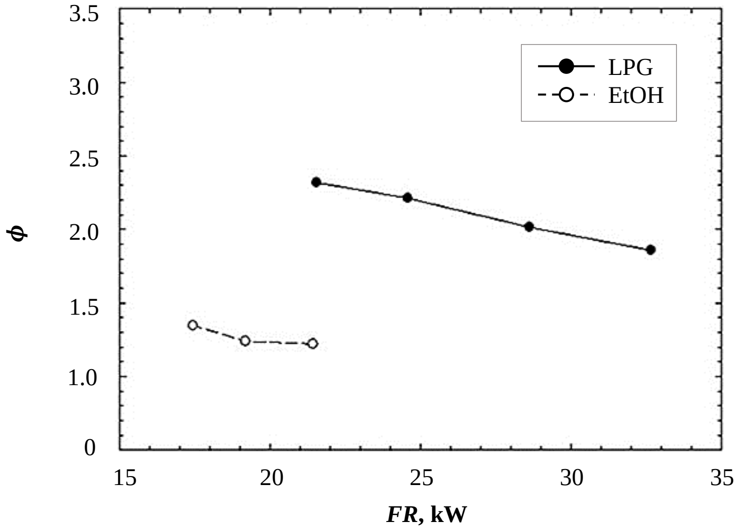


Figure 7 Predictions of the primary equivalence ratio of LPG and Ethanol

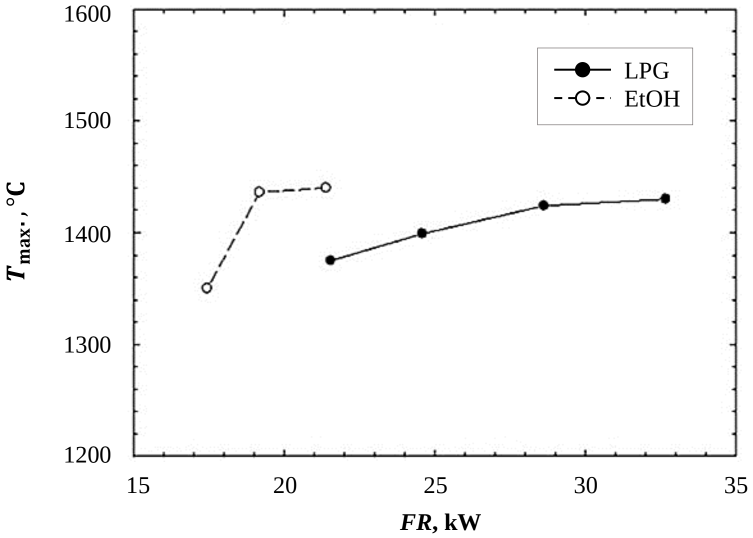


Figure 8 The maximum combustion temperature of LPG and Ethanol

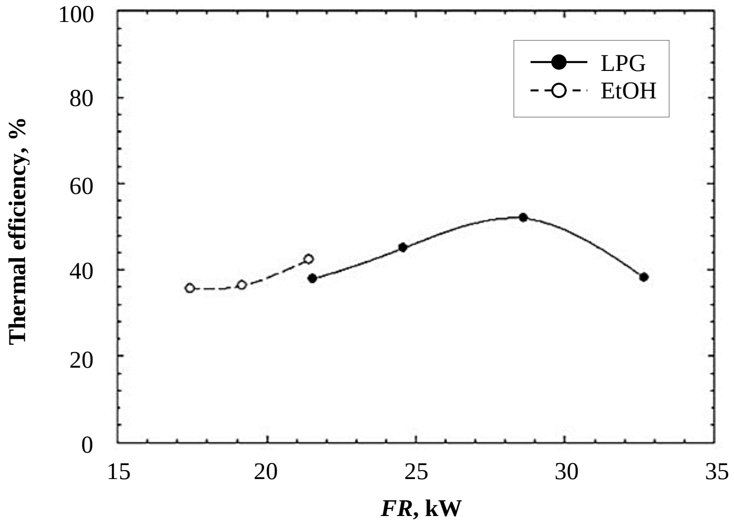


Figure 9 Thermal efficiency of the burner as a function of *FR*

3.2 Effect of fuel type on burner performance

A comparison of thermal efficiency between the two fuels is shown in Figure 9. As mentioned earlier and shown in Figure 6, the heating value of ethanol is lower than that of LPG. If the two fuels are injected at the same mass flow rate, the corresponding *FR*s of ethanol are lower than those of LPG. Thermal efficiency of the burner is quite sensitive to the variation of *FR*. At the given range of *FR*, increasing *FR* of ethanol from 17.5 kW to 21.4 kW, thermal efficiency increases from 38% to 42.3%. Moreover, as compared with LPG at approximately the same *FR* of 21.6 kW, the thermal efficiency of ethanol is higher than that of LPG. This might be a result of the higher convective heat transfer of the ethanol flame owing to a higher mass flow rate injected (Figure 6) and a higher combustion temperature, as compared with those of the LPG flames (Figure 8). For LPG, a bell-shaped thermal efficiency was achieved. Increasing *FR* from 21.6 kW to 28.6 kW, thermal efficiency increased and reached a maximum of 52% at 28.6 kW. This is higher than the previous burner [27], in which the maximum thermal efficiency was 51% and 41%, with and without a preheating unit, respectively. However, increasing *FR* higher than 28.6 kW, thermal efficiency had a tendency to decrease. At *FR* of 32.7 kW, thermal efficiency declines to 40% because of greater heat loss. However, it is higher than that of the conventional gas burner compared at the same range of firing rates [30].

In addition, the uncertainty analysis of thermal efficiency has been estimated using the propagation of uncertainties method [31]. Using Eq. (19), by considering the uncertainties of mass of water (m_{water}), temperature differentials (ΔT), and fuel flow rates ($\dot{m}_{\mathrm{fuel}}$), it was found that the maximum uncertainty in the estimation of thermal efficiency has been found within±1.6%.

$$\delta\eta_{th} = \left[\left(\frac{\partial\eta_{th}}{\partial m_{water}}\delta m_{water}\right)^2 + \left(\frac{\partial\eta_{th}}{\partial\Delta_{water,i}}\delta(\Delta_{water,i})\right)^2 + \left(\frac{\partial\eta_{th}}{\partial\dot{m}_{fuel}}\delta\dot{m}_{fuel}\right)^2\right]^{1/2} \tag{19}$$

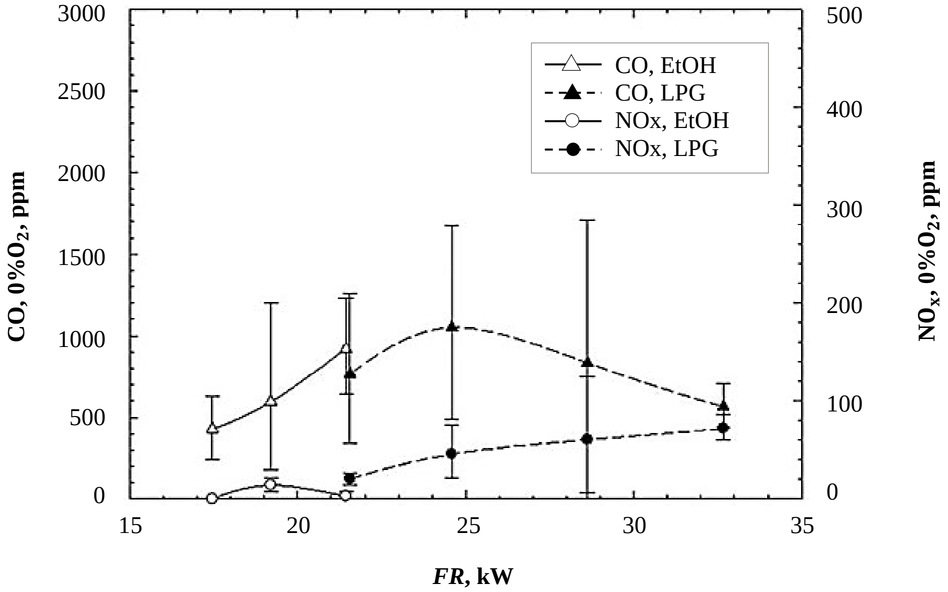


Figure 10 CO and NO$_X$ emissions of the burner as a function of firing rate

In Figure 10, CO emissions showed the same tendency as that of thermal efficiency in the relationship with *FR*. This trend in the CO emissions seems to be inverted compared to a conventional burner, which is a U-shaped curve. This might be because the given range of *FR* did not meet optimum conditions in terms of CO emissions. However, the average CO emission was lower than the emission limits of both the European standard requirement [32] (i.e., CO < 1,000 ppm at 0% O_2) and the Thailand industry standard [29] (i.e., CO < 2,000 ppm at 0% O_2) for household appliances. NO_x emissions of both fuels were lower than 100 ppm for all given *FR*s. NOx emission from ethanol was lower because there was less prompt NOx production, e.g., CH [33], which is most prevalent in rich hydrocarbon flames as in the case of LPG. In addition, for LPG, NO_x emissions increased as *FR* increased due to an increase in peak temperature.

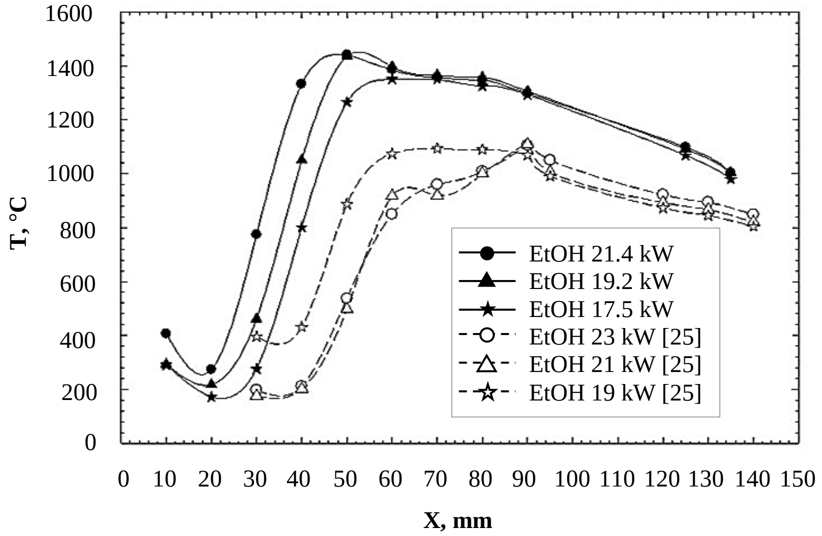


Figure 11 Comparisons of temperature profiles for ethanol burning in the burner

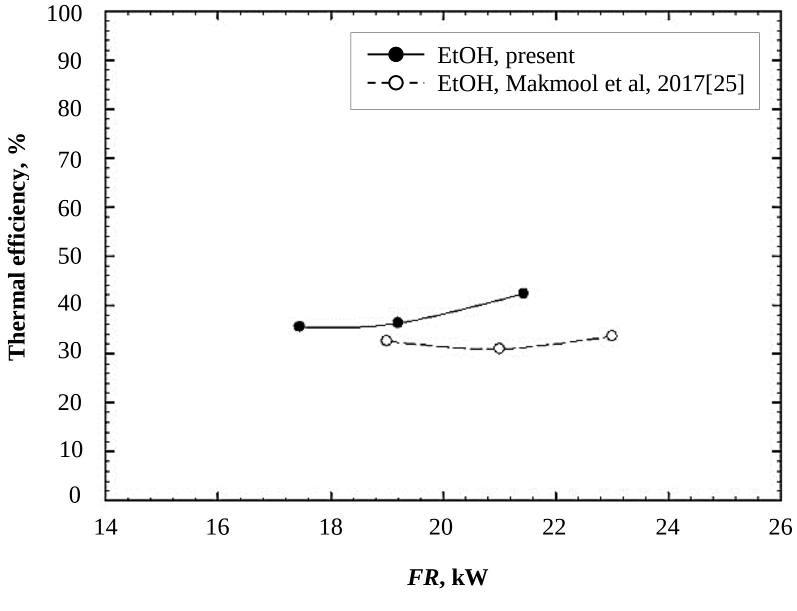


Figure 12 Thermal efficiency of ethanol burning in the burner

3.3 Improvements of the liquid ethanol porous-medium burner

This proposed burner was designed for heavy-duty cooking as a self-aspirating burner where an air compressor was not required. The burner is fuel-flexible, i.e., ethanol can replace LPG as cooking fuel. Even though ethanol has a low calorific value, because of the advantage of porous medium combustion technology, stable and complete combustion was achieved. Moreover, a spray or an atomization unit of liquid ethanol was not required. Liquid ethanol was completely vaporized inside the column that twisted around the combustion chamber. The column was embedded inside the wall and heated by combustion. This makes the burner system less complicated. As a result of newly designed aspects, performances of the burner in terms of thermal efficiency and emissions are improved compared to the previous works, proposed by Makmool et al., and Chaelek et al.

Compared to the previous ethanol burner, Makmool et al. [25] the present burner appears to have a better combustion performance in terms of combustion temperature (Figure 11). In turn, it has higher thermal efficiency (Figure 12) as compared at the same range of *FR*. The maximum thermal efficiency reaches 42.3% at *FR* of 21.4 kW, which is approximately 10% higher than the previous one. The higher combustion temperatures, compared to the previous burner, suggest that leaner flames (i.e., the nearer stoichiometric combustion) take place in this burner. The improvement in the burner's performance comes from balancing the flow velocity and flame burning velocity that had been taken into account in the design method. This results in the combustion chamber reduced by 60% by volume. The previous ethanol burner had a large surface area that resulted in a lower mixture velocity at the given power input. Moreover, it was not well matched to the flame velocity, which increased as the air entrainment rate increased (by increasing the opened area at the entrance of the mixing tube via the air adjustor). In other words, the prior burner has low flame stability, i.e., the flame had favourable flashback near the stoichiometric condition. Insufficient air was supposed to be a reason for the lower combustion temperature as compared to the present work.

A progress comparatively of the burner performances with the previous work was summarized as Table1.

Table 1 Comparison of the present work with previous published on the self-aspirating porous medium burner for heavy-duty cooking

Authors	**Fuel**	**Power input, kW**	**Max. temperature, °C**	**Max.thermal efficiency, %**	**Emissions**	
					CO, ppm	**NOx, ppm**
Present work	Ethanol	16.5-21.4	1,440	42.5	180-1,220	0-22
	LPG	21.6-32.7	1,430	52.0	40-1,700	15-125
Chaelek et al. [27]	LPG	21.0-44.0	1,436	51.0	1,000 - >6,000	40-50
Makmool et al. [25]	Ethanol	19.0-23.0	1,155	33.6	<1,000	<50

In addition, as considering the burner's performances of using LPG, the present work has improvement of both thermal efficiency and CO emissions as compared to the previous work proposed by Chaelek et al. [27], in which, the preheating unit was used to improve thermal efficiency. Unfortunately, with the preheating unit, rich combustion was taking place due to the increase of air temperature. The density of air reduces, and its viscosity increases, thus, reducing the air flow rate being entrained. The equivalence ratio increases and thus CO emissions increase as well.

As a result of newly designed aspects, the burner's performances in terms of thermal efficiency and emissions are improved compared to the previous works. Nevertheless, there are still some limitations of the present work. N_2 tank was required to pressurize ethanol. This system might not be easy to use for cooking in the household or where the user has no experience. In addition, to start the combustion, LPG is still needed for ignition and for preheating the burner for the evaporation process of ethanol.

4. Conclusions

This study presents the successful development of a fuel-flexible, natural-draft, porous-medium burner for heavy-duty cooking. The key parameters that control the performance of the burner in terms of combustion stability and combustion efficiency were taken into account in the design of the burner. The mixture flow velocity and the burning velocity were balanced by means of the estimation of the fuel equivalence ratio at the designed firing rate. The size of the burner is 60% less by volume compared to the previous ethanol burner, yet it yields better combustion and thermal performance. Conventional LPG and ethanol were injected through a common conventional gas nozzle without modifications. The results show that the burner can be operated at a wide range of *FR*, from 21.6 kW to 32.7 kW for LPG, and from 17.5 kW to 21.4 kW for ethanol. Even though the corresponding *FR*s of ethanol are lower because it has less heating value than LPG, the maximum combustion temperature of ethanol is 1440 °C, which is approximately the same as the maximum combustion temperature of LPG. This shows that the burner has a high potential to replace the consumption of fossil fuel with a renewable energy resource. This helps reduce the environmental impact. Even thought, ethanol is still more expensive than LPG with respect to heating value. Fortunately, ethanol is produced domestically that is sustainable. Replacement of LPG by ethanol for cooking will help to reduce demand for LPG, in the cooking sector, which is more than 2,000 ton per year or 32% of the total LPG consumption in Thailand [34].

5. Acknowledgements

This study was carried out at the Combustion and Engine Research Laboratory (CERL) and the Pilot Plant and Training Institute (PDTI), King Mongkut's University of Technology Thonburi (KMUTT). It was supported by the Higher Education Research Promotion and National Research University Project of Thailand (NRU), Office of the Higher Education Commission.

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