ISSN (Print) : 0974-6846
ISSN (Online) : 0974-5645
Indian Journal of Science and Technology, *Vol 11(14), DOI: 10.17485/ijst/2018/v11i14/118145, April 2018*

Experimental Analysis of Thermal Efficiency of a Porous/Swirl Burner Applied to Industrial Cooking

Luis Olmos-Villalba[1*], Bernardo Herrera[2] and Karen Cacua[2]

[1]Research and Innovation on Energy Group, Faculty of Engineering, Institución Universitaria Pascual Bravo, Calle 73 No. 73A-226, Medellín, Colombia; luis.olmos@pascualbravo.edu.co
[2]Advanced Materials and Energy Group, Faculty of Engineering, Instituto Tecnológico Metropolitano, Campus Fraternidad, Calle 54A No 30-01, Medellín, Colombia; bernardoherrera@itm.edu.co, karencacua@itm.edu.co

Abstract

Objectives: To analyze the thermal performance and emission of a porous media combustion/swirl and a conventional burner applied to industrial cooking through a comparative study. **Methods/Statistical Analysis**: A porous/swirl burner with a bed of Al_2O_3 particles coming from grinding wastes has been tested experimentally, and compared to conventional burner from Industrial cooking process. Thermal efficiency of the burner has been evaluated according with specifications of standard NTC 5306. After combustion stability is achieved, a measured quantity of water is heated up and the temperature is measured. The test ends when water gets the boiling point. **Findings:** The results showed that the swirl burner had a stable combustion in a narrow primary equivalence ratio between 1.49 and 1.52. Thermal efficiency in the "radiation-convection" mode of the porous burner was between 15.7 and 23.6%, which are lower than the average thermal efficiency of the conventional free-flame burner, while the swirl burner working independently could improve the thermal efficiency between 3 and 5% in respect of the conventional free-flame burner. **Application/Improvements:** This lower thermal efficiency prevents the potential use in industrial application of the porous/swirl burner and further work is necessary to improve the thermal efficiency of the coupled system.

Keywords: Combustion Stability, Industrial Cooking, Porous Burner, Swirl Burner, Thermal Efficiency

1. Introduction

The dynamics of today's worldwide economy, the rational use and efficient of energy have made significant progress in the direction of the energy efficiency as a production chain concept, in a constantly changing with the new approaches to sustainable development in relation to the decrease of the environmental impacts, the increment of productivity and in the productive processes[1]. In developing countries, the residential sector takes an important place in the energy consumption. Hence, the energy used for cooking plays an interesting role into the sustainable development[2] and in the search for energy savings of a country. Cooking in the food industry refers to a several process to transform food by a thermal treatment and it includes baking, roasting, broiling, boiling, frying and stewing[3]. These processes are using several types of energy sources, which vary depending on the how heat is transferred to the load. According with the review[4], electricity appliances have the largest end-use efficiency for domestic cooking, around 80%, but the efficiency of the total system since production and transfer of energy to end-use is only 18%. On the other hand, natural gas and LPG cooking appliances have lower end-use efficiency but total system efficiency is around 50%, meaning a better utilization of the primary energy.

*Author for correspondence

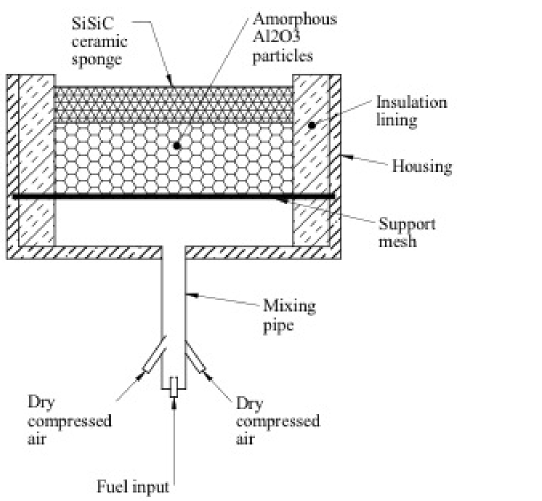
The Bunsen principle is commonly applicable in burners design for cooking process, where a jet of gaseous fuel entrains air from surroundings and combustion of the premix occurs in the head of the burner. The convection is the predominant process on the heat transfer in a cooking process, but the free-flame of the burner contributes to the heat transfer by radiation process. The overall efficiency of this type of burners does not depend only of combustion efficiency of the premix but also upon different conditions such as temperature, pressure, wind speed, specific heat capacity of the vessel, overall shape of vessel, weight of vessel, and size of vessel[5]. Within technologies applied to industrial cooking, which can provide significant environmental impacts and improve the efficiency, Porous Media (PM) burners is a good alternative to be studied, due to combustion in porous medium is proved to be one of the feasible options to tackle the aforesaid topics. PM burner present low CO and NOx emissions, the high thermal efficiency, extension of the lean flammability, wide power modulation ratios, multi-fuel capabilities[6–9]. PM combustion, also known as filtration combustion, is the interaction between two different media, usually a solid and a gas, but a solid and a liquid can also be used[10,11]. The combustion reaction takes place inside of the solid media (porous media), when the combustion begin the hot medium radiates heat in all directions around and this causes the preheating of the incoming air-fuel mixture, expands the reaction zone and generates a homogeneous temperature distribution within PM.

PM burner have two zones, preheating zone and the combustion zone, these zones are heavily dependent of the critical pore size. If the size of the pores is smaller than its critical dimension, flame propagation is not allowed, so it is a preheating zone; the flame is always quenched. On the other hand, the combustion zone is when the pore size exceeds its critical dimension and thus flame propagation inside the porous structure is possible[9,12]. These zones are also a function of the flame stabilization parameters like Peclet number and porosity, among others. Porous burners are suited for numerous applications including heat exchangers systems, off gas burners, reformers and household heating appliances and can be operated with a wide range of gaseous fuel[13]. However, few studies have been focused on the thermal efficiency of porous burner for cooking. In[9] assessed the thermal efficiency and emissions of a porous radiant burner with different porosities and at different equivalence ratios (0.5–0.7) and wattages (1.3–1.7 kW) for LPG domestic cooking application. The maximum porous burner thermal efficiency was founded with 90% in the porosity of the porous media and it was 10% higher than that of the maximum thermal efficiency of the conventional LPG domestic cooking stoves available in the Indian Market and the measured values of CO and NO emissions were in the range of 250–650 and 4–7 mg/m^3, respectively. In[14] studied the thermal efficiency of a two-layer porous media for domestic cooking with LPG whose combustion zone was made up of silicon carbide, and alumina balls formed the preheating zone. The maximum thermal efficiency was found to be 3% higher than that of the maximum thermal efficiency of the conventional domestic LPG cooking stoves and the CO and NOX emissions were found in the ranges of 25–350 mg/m^3 and 12–25 mg/m^3, respectively. Nevertheless, both of the studies aforementioned used compressed air to overcome the pressure drop into the porous media which restricts the application in domestic cooking since an air compressor is not a common appliance in homes.

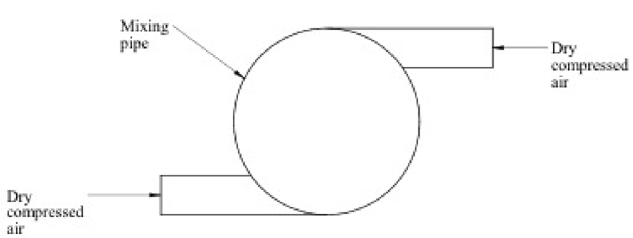
At this moment, researches have studied the behavior of the temperature, thermal efficiency, CO and NOX emissions, the equivalence ratio, the air and fuel mixing process and flame stabilization phenomenon into a PM burner for several applications[9,13–19]. In[20] made an experimental research to assess the effects of swirl flow on the burner performance and propose suitable design or operational factors for domestic gas burners. As main results achieved include an increase of thermal efficiency about 12% and an increase of CO emission about 95 ppm of the swirl flow burner with the semi-confined combustion flame yields compared with the conventional radial flow burner with open flame at the maximum thermal input of 4.41 kW. Based on the above, this work presents a comparative thermal performance and Emission analysis study porous media combustion/swirl and conventional burner applied to industrial cooking.

2. Methodology

The porous burner for this experiments is divided into two main zones as shown in Figure 1(a). A preheating zone where particles of Al_2O_3 resulting from waste of grinding in industrial ball mills. These particles have an average equivalent diameter of 11 mm and it is supposed that this material can reduce the cost for manufacturing of the porous burner. The other zone is the main combustion zone and it consists of a ceramic foam 20 ppi (pores per inch) composed of SiSiC. A ceramic fiber lining surrounds both materials in order to reduce the heat loss through the metallic housing of the burner. The Al_2O_3 particles are supported in a metallic mesh with a diameter of 6 mm in each opening. Fuel has been supplied to porous burner from a pressurized cylinder and it enters to burner at 23 mbar (gauge). The flow of fuel is controlled by an electronic mass flow controller (OMEGA FMA 5400). Dry air has been supplied from a mechanical screw air compressor at 700 mbar and its flow is regulated by a suitable control needle valve and a rotameter. Both, fuel and air are mixed in a steel tube by mean of an ascendant helicoidally path caused by the disposition of two air inlets as shown in Figure 1(b).



(a)



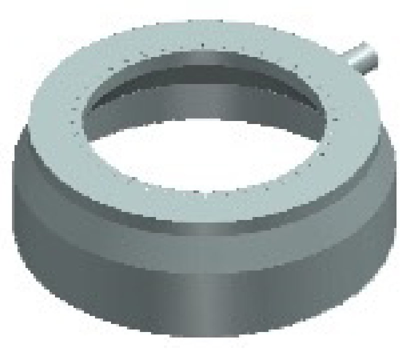
(b)

Figure 1. Experimental setup. **(a)** Scheme of the porous burner. **(b)** Top view of the mixing pipe and the air inlet.

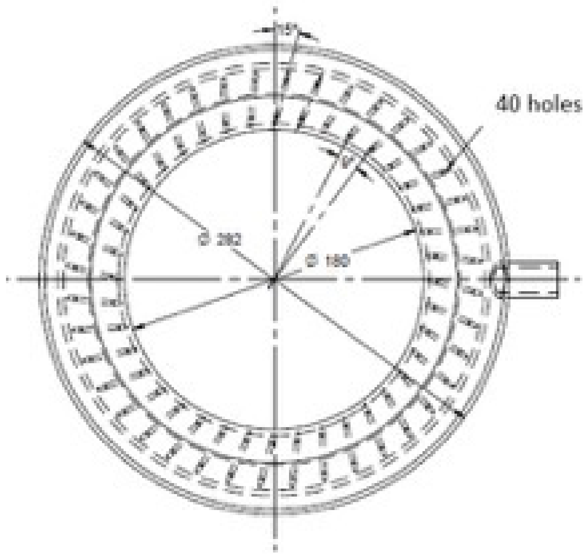
Swirl burner is located on the top of the porous burner forming a concentrically ring as shown in Figure 2(a). It has an independent supply of fuel through a self-aspirating venturi mixer, which entrains primary air from the surroundings. The ring of the swirl burner has 40 circular ports, which have inclinations with respect to normal and radial direction in order to generate the swirl effect in the flue gases as shown in Figure 2(b). An electronic mass flow controller (OMEGA FMA 5400) also controls the flow of fuel in the swirl burner. The primary air equivalence ratio in the swirl burner has been calculated according to the Equation (1):

$$\Phi_{pri} = \frac{\%O_{2,stch}}{\%O_{2,premix}} \tag{1}$$

Where $\%O_{2,stch}$ and $\%O_{2,premix}$ are the oxygen percentage in a stoichiometric premix (26,6%) and in the premix before combustion, respectively. Burner operation starts with a preheating of the Al_2O_3 balls up to ignition temperature of the fuel near to 550°C. Preheating is done by a free flame located upstream of the Al_2O_3 balls and the premix entering the burner in this stage has an equivalence ratio around 0.92. Once the ignition temperature is achieved after 8 to 12 minutes, porous bed combustion starts as shown in Figure 3(a) by setting the set point of the mass flow controller in order to adjust the input heat rate and then the air flow is regulated for adjusting the equivalence ratio. Swirl burner starts with the preheating of the porous burner since it has an independent supply of fuel. The stability of the porous combustion for every input heat rate is determined by varying the equivalence ratio between two limits: An upper limit where a free flame appears upstream of the Al_2O_3 balls and the lower limit where the flame front exceeds the height of the porous bed and a dark area appears on the surface of the ceramic sponge at top of the burner as shown in Figure 3(b). This dark area is followed by a cooling of the ceramic sponge and high CO emissions. On the other hand, stability of the swirl burner has been studied by variation of heat input rate until flashback and lift-off appear.



(a)



(b)

Figure 2. Swirl burner. **(a)** View of the construction. **(b)** Details of the holes inclinations and dimensions.

(a)

(b)

Figure 3. Operation of the porous burner. **(a)** Stable operation. **(b)** Unstable operation in the lower equivalence ratio.

Flue gases were collected in a hood which was made according with specifications of standard NTC 5306 available in Colombia for energy efficiency in cooking appliances burning gas. Then, the concentrations of CO_2 and CO were measured by infrared dispersion while concentration of O_2 was measured by a paramagnetic cell in a gas analyzer SICK GMS 810. Thermal efficiency of the burner has been evaluated according with specifications of standard NTC 5306. After stability of the porous combustion is achieved, a measured quantity of water (1±0,005 kg) is heated up and the temperature is measured with a mercury thermometer (accuracy±1 °C). After water reached 80°C there was a mechanical stirring for homogenizing the temperature in the last stage of heating. The test ends when water get the boiling point (94°C in Medellín, Colombia) and the time since the heating started is noted. Since flow of fuel was kept constant and controlled during the heating process, the thermal efficiency of the burner was calculated as shown in the Equation (2):

$$\eta = \frac{\dfrac{m_w \times C_{pw} \times \left(T_b - T_0\right)}{t}}{\dot{m}_f \times LHV_f} \tag{2}$$

Where η is the thermal efficiency, m_w is the mass of water in kg, C_{pw} is the specific heat of water (4,18 kJ/kg.°C), T_b and T_0 are the boiling point and the initial temperature of water in °C, t is the time to reach the boiling point in seconds, $\dot{m}_f$ is the mass flow of fuel in kg/h and LHV_f is the low heating value of the fuel in kJ/kg. Thermal efficiency test have been achieved three times for the porous burner and the swirl burner working separately and working coupled in order to compare the influence of the individual efficiency of the burners in the coupled system as shown in Figure 4. For each input heat rate in the porous burner, thermal efficiency tests were done locating the aluminum vessel 5 cm above the surface of the ceramic sponge at lower stable equivalence ratio where maximum temperature on the surface of the ceramic sponge was reached. This is because of higher thermal efficiency at lower equivalence ratio can be achieved due to the combustion of lean mixture and also the movement of reaction zone downstream due to higher air flow rates, resulting in maximum volumetric heat release[9].

Figure 4. Porous and swirl burner working coupled.

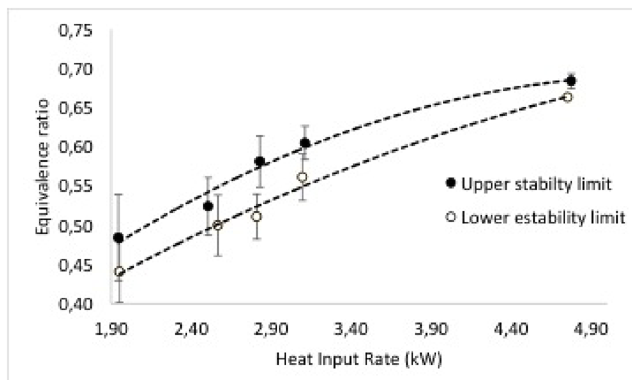
According with accuracy of the measurement equipment used in the thermal efficiency tests, uncertainty of the experiments was estimated in ±0.4%. Finally, a conventional free flame self-aspirating burner for industrial cooking was tested according with the standard NTC 5306 under similar input heat rates than the porous and swirl burner in order to obtain a comparison in thermal efficiency. Such a burner has a “concentric-cross” distribution of the flame ports as is shown in Figure 5 and it has the same diameter of the ceramic sponge in the porous burner.

Figure 5. Conventional “concentric-cross” free flame self-aspirating burner for industrial cooking.

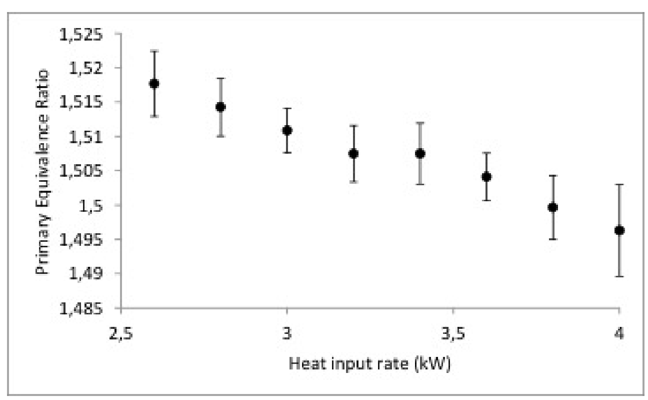
3. Results and Discussion

3.1 Combustion Stability

Combustion stability is considered as the property of a burner where a combustion zone can work for a long time without flashback, lift-off or high CO emissions resulting from incomplete combustion. Although porous burners do not have a visible flame front, the concept of stability is appropriate since flashback or lift-off can occur depending on the equivalence ratio. On the other hand, a swirl burner is susceptible to show flashback, lift-off and high CO emissions if equivalence ratio is not suitable or if the velocity of the premixed fuel and air are not enough to achieve the swirl zone for stabilization of the combustion. Figure 6(a) shows a stability range of the porous and the swirl burner working separately with LPG for several input heat rates. It can be seen that for input heat rate lower than 3.1 kW, the upper and lower equivalence ratio on the stability limit of the porous burner follow approximately a linear trend as well as the wide of the range of stability remains constant but this trend is broken when higher input heat rate is applied. Also, the equivalence ratio range stability of porous burner gets narrower as the input heat rate increases and any increasing in the heat input rate causes an increasing in the equivalence ratio in order to combustion become steady. In despite of a higher diameter burner of 16 cm has been used for this work, the ranges of stable equivalence shown in Figure 6 have high correspondence with those values reported in the literature[9,14].



(a)



(b)

Figure 6. Stability range of. **(a)** The porous burner. **(b)** The swirl burner working with LPG for several heat input rates.

Concerning the operation of the swirl burner, Figure 6(b) shows that stable combustion works for heat input rate between 2.6 and 4.0 kW since lower heat input rates made flashback appeared and upper heat input rates increased the yellow tips and CO emissions until lift-off occurred. It is worth mentioning that the air and fuel venturi mixer was designed for heat input rates around 3 kW. In this case, stability of combustion was achieved for primary equivalence ratios between 1,49 and 1,52, although a lower equivalence ratio is expected when secondary air mixes at the combustion zone. The combustion stability of the swirl burner is conditioned by the maximum air entrainment that is achievable in the venturi mixer which in turn regulates the velocity of the air/fuel premix at the outlet port of the burner, i.e., high air entrainment increases the velocity of the premix jets in the combustion zone and the swirl turbulence formed helps to stabilize the flame. However, primary equivalence ratios lower than 1.49 generates premix velocity quite higher than the flame speed, which is not enchanced by preheating of the premix as occurs in the porous burner. Similarly, equivalences ratios higher than 1.52 showed tendency to flashback because of the velocity of the premix is not enough to establish a swirl flow on the head of the burner. On the other hand, the higher heat input rate of 4.0 kW admitted larger variation of the stable equivalence ratio, which is due to the larger velocity or the premix when LPG flow increases and entrainment of primary air is higher.

The measurements of CO emissions are shown in Figure 7. Emissions from the porous burner are kept in acceptable values lower than 25 ppm for input heat power lower than 3.1 kW but an important rising in the CO emissions can be seen when the burner works at higher input heat rate. The destabilization in the emissions of CO is probably a consequence of a moderate lift-off on the surface of the ceramic sponge because a higher mass flow of the air and fuel premix. The emissions from the swirl burner are lower than 60 ppm in the range of heat input rate between 2.6 and 4.0. No trend is identified in the CO emissions respect to heat input rate in the swirl burner, indeed, no significant variation has been found since error bars overlap for every thermal input measured. Heat input rate higher than 4.0 kW yielded CO emissions above 100 ppm (not shown in Figure 7) which is a consequence of the higher flow of fuel and a reduction of the entrainment of air due to restrictions in the venturi mixer. The stability of the combustion in the porous burner is more limited at high input heat rate and the increasing in the emissions of CO are probably a consequence of a moderate lift-off on the surface of the ceramic sponge because a higher mass flow of the air and fuel premix, although a dark zone was not seen for this case, in contrast to the total destabilization shown in Figure 3(b). The CO emissions of swirl burner show that combustion is not completely stable in the range of 2.6 to 4.0 kW since there is a high variation of the CO concentration in the flue gas. This variation is likely due to changes in the patterns of mixing of the premix and secondary air at the head of the burner and these changes can be produced by alterations in the velocity of the premix jets and the swirl effect. However, an experimental study has to be done to verify the validity of the precedent conclusion.

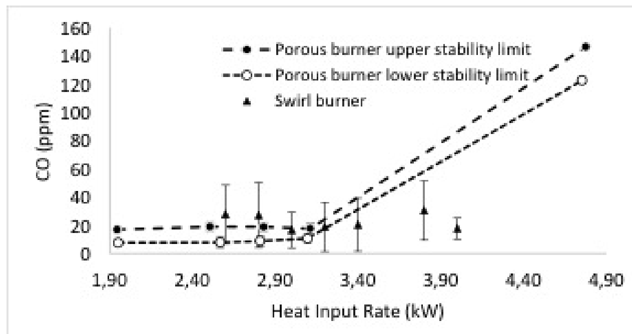


Figure 7. CO emissions of porous and swirl burner

3.2 Thermal Efficiency

Thermal efficiency in a heating device is referred to the useful energy, which actually rises the temperature or changes the phase of a solid or fluid with respect to primary energy in the device. High thermal efficiency equipment in the food industry is a request for improvement of productivity and competitiveness as same as reduction of greenhouse gases emissions and other atmospheric pollutants. In Colombia, a large number of heating devices in food industry are based on the free-flame propagating burners which are manufactured with different sizes and shapes but most of cases the manufacturing does not follow a design based on energy efficiency. That is why comparison of the performance of a porous burner for industrial cooking was necessary on the reference of a "concentrically-cross" free flame burner which is a conventional burner for food processing and the results are shown in Figure 8. It can be seen that thermal efficiency of the conventional burner does not show a significant variation with changes of thermal input in the burner and the value oscillated around 25.5%. It is worth to note that the mass of the pot was not included in the calculation of the thermal efficiency since only useful energy for cooking is considered. On the other hand, experiments of thermal efficiency in the conventional burner were made up to input heat rate of 3,7 kW because of yellow-tips appeared in the flame beyond this value and emissions of CO larger than 500 ppm were measured. So, modulation of this burner, i.e., ratio between higher and lower input heat rate is close to 1,85.

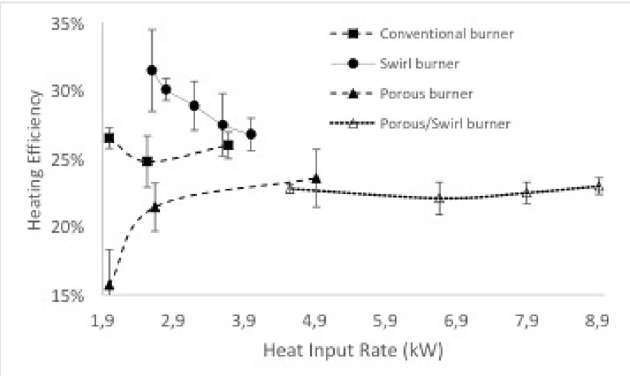


Figure 8. Thermal efficiency for conventional free-flame burner and porous burner in "conduction" mode and "convection-radiation" mode.

Figure 8 also shows thermal efficiency of the porous burner working in the "radiation-convection" mode where the pot is located 3 cm above the surface of the ceramic sponge and heating is transferred by radiation of the surface at temperatures between 667 and 916 °C and convection coming from flue gases leaving the burner. It can be seen that radiation-convection mode had a thermal efficiency between 15.7 and 23.6% which are lower than the average thermal efficiency of the conventional burner. This lower efficiency can be explained because of the lower temperature achieved on the surface of the ceramic, i.e., radiation mechanism does not have a representative influence on heat transfer for temperatures on the surface around 600 to 900 °C. Previous works have shown similar results, for example[14], found efficiencies up to 15% below conventional LPG free flame burners in India, and they could obtain an increase in efficiency around 3% only with an equivalence ratio of 0.38 and 8 cm in the diameter of the porous bed.

While the "radiation-convection" mode of the porous burner did not show an increasing of the thermal efficiency, the swirl burner working independently made visible an advantage respect to free flame conventional burners as shown in Figure 8. It can be seen that heating the load with the convection of the swirl gases could improve the thermal efficiency between 3 and 5%. The maximum thermal efficiency was obtained when an input heat rate of 2.6 kW was applied and the efficiency decreased as the input heat rate is increased up to 4.0 kW. This behavior is a consequence of the increasing of energy losses in the flue gases as the input heat rate is larger, i.e., larger loss of sensible heat in the flue gases are expected when both fuel and air flow increase. Also, higher heat input rate implies higher velocity of the flue gases, which impinge on the bottom of the pot breaking the recirculation effect of the swirl.

The thermal efficiency of the porous and swirl burner working together is also shown in Figure 8. The range of heat input rate for this combined mode runs between 4.95 and 8.92 kW, which limits correspond to the summary of the minimum and maximum limits of stability of the porous and swirl burner working separately. The combined mode had thermal efficiency around 22.5% for every heat input rate which results lower than the thermal efficiency of the conventional and swirl burner. As can be seen in Figure 8, thermal efficiency of swirl burner showed a reduction as the heat input rate increases while thermal efficiency of the porous burner on "radiation convection" mode had an increment when heat thermal input was risen from 2.0 to 4.0 kW. This contrary tendency of the two burners working separately can explain why thermal efficiency of the combined mode remains approximately constant in the whole range of thermal input. However, the lower thermal efficiency prevents the potential use in industrial application of the porous and swirl burner working simultaneously and further work is necessary to improve the thermal efficiency of the coupled system, mainly the thermal efficiency of the "convection-radiation" mode of the porous burner.

4. Conclusions

A porous burner made of a ceramic sponge of SiSiC and a bed of Al_2O_3 particles coming from grinding wastes and combined with an aspirating swirl burner has been evaluated respect to combustion stability and thermal efficiency for cooking in food industry. The results showed that for input heat rate lower than 3.1 kW in the porous burner, the upper and lower equivalence ratio on the stability limit follow approximately a linear trend as well as the wide of the range of stability remains constant but this trend is broken when higher input heat rate is applied. Every equivalence ratio for stable combustion was in the lean ratio and combustion close to stoichiometric values was not feasible because flashback occurred. Concerning to the swirl burner, it had a stable combustion in a narrow primary equivalence ratio between 1.49 and 1.52. The total equivalence ratio including secondary air in this burner was not measured.

Emissions of CO in the porous burner were in acceptable values lower than 25 ppm for input heat power lower than 3.1 kW but an important rising in the CO emissions could be seen when the burner works at higher input heat rate. The destabilization in the emissions of CO is probably a consequence of a moderate lift-off on the surface of the ceramic sponge because of the higher mass flow of the air and fuel premix. The emissions from the swirl burner are lower than 60 ppm and no trend is identified in respect of heat input rate. This randomly variation of emission of CO is likely due to changes in the patterns of mixing of the premix and secondary air at the head of the burner and these changes can be produced by alterations in the velocity of the premix jets and the swirl effect.

Thermal efficiency was calculated for each burner working separately and combining both of them. Thermal efficiency in the "radiation-convection" mode of the porous burner was between 15.7 and 23.6% which are lower than the average thermal efficiency of the conventional free-flame burner. This lower efficiency is due to the lower temperature achieved on the surface of the ceramic sponge and a diameter of the burner that exist an optimal value according with precedent results in the literature. On the other hand, the swirl burner working independently could improve the thermal efficiency between 3 and 5% in respect of the conventional free-flame burner and the efficiency decreased as the input heat rate is increased as a consequence of the increasing of energy losses in the flue gases as the input heat rate is larger and the impingement on the bottom of the pot breaking the recirculation effect of the swirl.

The combined mode had thermal efficiency around 22.5% for every heat input rate, which results lower than the thermal efficiency of the conventional and the swirl burner as a consequence of the contrary tendency of the thermal efficiency in the two burners working separately. This lower thermal efficiency prevents the potential use in industrial application of the porous and swirl burner working simultaneously and further work is necessary to improve the thermal efficiency of the coupled system.

5. Acknowledgements

The authors would like to thank the Institución Universitaria Pascual Bravo Project (P13109) and the Instituto Tecnologico Metropolitano de Medellín e ITM-Project (P13109) for their financial support in this research.

6. References

1. Prias CO. Programa De Uso Racional Y Eficiente De Energía Y Fuentes No Convencionales – Proure. Bogota; 2010 Apr. p. 1–151.
2. Lucky RA, Hossain I. Efficiency study of Bangladeshi cookstoves with an emphasis on gas cookstoves. Energy. 2001 Mar; 26(3):221–37. Crossref.
3. Karel M, Lund DB. Physical principles of food preservation. 2nd ed. New York: Marcel Dekker, Inc; 2003. p. 1–603. Crossref.
4. Hager TJ, Morawicki R. Energy consumption during cooking in the residential sector of developed nations: A review. Food Policy. 2013 Jun; 40:54–63. Crossref.
5. Gohil PP, Channiwala SA. Experimental investigation of performance of conventional LPG cooking stove. Fundamental Journal of Thermal Science and Engineering. 2011; 1(1):25–34.
6. Liu JF, Hsieh WH. Experimental investigation of combustion in porous heating burners. Combustion and Flame. 2004 Aug; 138(3):295–303. Crossref.
7. Takami H, Suzuki T, Itaya Y, Hasatani M. Performance of flammability of kerosene and NOx emission in the porous burner. Fuel. 1998 Feb; 77(3):165–71. Crossref.
8. Kaplan M, Hall MJ. The combustion of liquid fuels within a porous media radiant burner. Experimental Thermal and Fluid Science. 1995 Jul; 11(1):13–20. Crossref.
9. Muthukumar P, Shyamkumar PI. Development of novel porous radiant burners for LPG cooking applications. Fuel. 2013 Oct; 112:562–6. Crossref.
10. Mujeebu MA, Abdullah MZ, Bakar MZA. Applications of porous media combustion technology- A review. Applied Energy. 2009; 86:1365–75. Crossref.
11. Mujeebu MA, Abdullah MZ, Bakar MZA. Combustion in porous media and its applications- A comprehensive survey. Journal of Environmental Management. 2009 Jun; 90(8):2287–312. Crossref. PMid:19299066
12. Durst F, Trimis D, Pickenäcker K. Compact porous medium burner and heat exchanger for household applications. Final Report, Contract No . JOE3-CT95-0019; 1996. p. 1–85.
13. Keramiotis C, Stelzner B, Trimis D, et al. Porous burners for low emission combustion: An experimental investigation. Energy. 2012 Sep; 45(1):213–9. Crossref.

14. Pantangi VK, Mishra SC, Muthukumar P. Studies on porous radiant burners for LPG (Liquefied Petroleum Gas) cooking applications. Energy. 2011 Oct; 36(10):6074–80. Crossref.
15. Gao H, Qu Z, Feng X. Combustion of methane/air mixtures in a two-layer porous burner: A comparison of alumina foams, beads, and honeycombs. Experimental Thermal and Fluid Science. 2014 Jan; 52:215–20. Crossref.
16. Wang H, Wei C, Zhao P. Experimental study on temperature variation in a porous inert media burner for premixed methane air combustion. Energy. 2014 Aug; 72:195–200. Crossref.
17. Gao HB, Qu ZG, Feng XB. Methane/air premixed combustion in a two-layer porous burner with different foam materials. Fuel. 2014 Jan; 115:154–61. Crossref.
18. Zhang T, Liu F, You X. Optimization of gas mixing system of premixed burner based on CFD analysis. Energy Conversion and Management. 2014 Sep; 85:131–9. Crossref.
19. Keramiotis C, Founti MA. An experimental investigation of stability and operation of a biogas fueled porous burner. Fuel. 2013 Jan; 103:278–84. Crossref.
20. Hou SS, Lee CY, Lin TH. Efficiency and emissions of a new domestic gas burner with a swirling flame. Energy Conversion and Management. 2007 May; 48(5):1401–10. Crossref.