

**PARAMETRIC STUDY OF POROUS MEDIA COMBUSTION
FOR HIGH TEMPERATURE HEAT EXCHANGER**

PANU IAMSAKULPANICH

**A THESIS REPORT SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE DEGREE OF
MASTER OF ENGINEERING IN AUTOMOTIVE ENGINEERING
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KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG
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THESIS TITLE	Parametric Study Of Porous Media Combustion For High Temperature Heat Exchanger
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ABSTRACT

This research has studied the experiment of porous media burner for LPG as fuel using porous media in the compact burner. The study parameters of the experiment were porous media size, fuel flow rate, air flow rate, and equivalence ratio. This system is using LPG fuel and using air for combustion. The compact burner diameter size 67 mm, length 200 mm and rape with fiber paper insulator thickness 5 mm all length of the burner and inside of burner contained with porous media long 100 mm. The porous media material in this experiment is Al₂O₃ in sphere shape with diameter size 3mm, 5mm, and 10mm. In this experiment, the temperature distribution was measured by thermocouple 1-20channel every 5 mm. The experiment condition is combustion in range of LPG between 175-525 cc/min with air 15,000-25,000 cc/min. The conditions for increasing the maximum temperature of combustion using porous media that is downsizing porous media ball, increase fuel flow rate, decrease air flow rate and increase equivalence ratio. From the result of the phenomena within LPG porous burner, the outcome is that porous media can help to support the combustion as the flame holder, flame stabilizer, well temperature distribution and slowly responded to change in thermal load.

Keywords: Porous media combustion, Al₂O₃, LPG, Compact burner, High temperature heat exchanger

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LIST OF SYMBOLS

q_k	Rate of heat conduction, <i>Watt</i>
A	The heat transfer Area, m^2
T	The temperature difference across the layer, K
x	The thickness of the layer, m
k	The thermal conductivity of the material, $W/m^2 \cdot K$
q_c	Rate of heat conduction, <i>Watt</i>
h_c	The average convection heat transfer coefficient, $W/m^2 \cdot K$
A	The surface area through which convection heat transfer takes place, m^2
T_s	The surface temperature, K
$T_{f,\infty}$	The temperature of the fluid sufficiently far from the surface, K
T	The temperature distribution in the fluid, K
$\partial T/\partial y _{y=0}$	The temperature gradient at the surface, K
q_r	The rate of heat radiation, <i>Watt</i>
σ	The stefan-boltzman constant value is 5.67×10^{-8} , $W/m^2 \cdot K^4$
A	The surface area of the object, m^2
T	The surface temperature, K
T_1	The surface temperature of blackbody, K

T_2	The surface temperature of the surrounding blackbody, K
ε	The emissivity of the surface, K
$(r \ \phi \ \theta)$	Spherical coordinates
(A / F)	Mass air-fuel ratio, $\text{kg}_{\text{air}}/\text{kg}_{\text{fuel}}$
$(A / F)_{\text{Stoic}}$	Stoichiometric mass air-fuel ratio, $\text{kg}_{\text{air}}/\text{kg}_{\text{fuel}}$
(F / A)	Mass fuel-air ratio, $\text{kg}_{\text{fuel}}/\text{kg}_{\text{air}}$
$(F / A)_{\text{Stoic}}$	Stoichiometric mass fuel-air ratio, $\text{kg}_{\text{fuel}}/\text{kg}_{\text{air}}$
$C_{p_{\text{ambient}}}$	Constant-pressure specific heat of ambient, $\text{kJ}/\text{kg}\cdot\text{K}$
$C_{p_{\text{flue gas}}}$	Constant-pressure specific heat of flue gas, $\text{kJ}/\text{kg}\cdot\text{K}$
D	Diameter of burner, m
h_{fg}	Enthalpy of formation gas, $\text{kJ}/\text{kg}_{\text{H}_2\text{O}}$

LIST OF DEFINITIONS

LPG	Liquefied Petroleum Gas
PMC	Porous Media Combustion
PM	Porous Media
FVM	Finite Volume Method
C	Carbon
H	Hydrogen
S	Sulfur
CO	Carbon Monoxide
O₂	Oxygen
N₂	Nitrogen
CNG	Compressed Natural Gas
LNG	Liquefied Natural Gas
AFR	Air-Fuel Equivalence Ratio
HHV	Higher heating value, kJ/kg
LHV	Lower heating value, kJ/kg

CHAPTER 1

INTRODUCTION

1.1 Research Background

Energy is necessary for human life and it is also a fundamental factor in national development. Thailand has many types of energy sources, but a small amount compared to other countries and the important energy role in driving national development is energy from fossil fuels such as Crude oil, which mostly used for power generation, transportation, and raw materials for the production of many types of industries. Every time the world crisis occurs, such as a war in the Middle East or the volatility in oil world markets, Thailand was affected by both economic and social. Because Thailand still has to rely heavily on imported oil from abroad each year. In addition to the above problems, Thailand still has high energy consumption problems for economic development. In the past 15 years, Thailand of energy consumption rate is more than 1.4 times the economic growth rate. In other words, If the country's economy grows 5 percent per year, energy spending will grow by 7 percent per year, it is quite high. Compared to developed countries such as the United States is 0.80 percent per year, Japan is 0.95 percent per year. It means Thailand use energy consumption for economic development in wastefully. Any wasteful consumption of energy thus puts high production costs and, also losing a lot of money to import energy. The best solution to this problem is the amount of the existing energy saving and maximum efficiency.

At present, the development of heat exchanger is another way to use energy efficiently. The heat exchanger is the main engineering device that widely used in industrial applications. The process of a heat exchanger requires a certain amount of energy to drive the process. Recently, high temperature heat exchangers (800~1000 °C) play an increasingly important role in various industries, for example, electricity industry (high temperature fuel cells and heat exchangers for gas turbines) and the chemical industry (hydrogen production and high temperature reactor). Increasing the temperature of the heat exchange process can increase the thermal efficiency, which can use less fuel and reduce emissions. At present, the porous media combustion technology is used in the high temperature heat exchanger, which has advantages due to the uniform temperature distribution, higher burning rates, increased power dynamic range, extension of the lean

flammability limits, and low emissions of pollutants over the free flame combustion. For these reasons, porous media combustion is selected to use as a burner for the high temperature heat exchanger. In order to study the feasibility of porous media combustion technology in applications, the experimental works will be carried out to explore the possibility of this technology for practical applications and to design and development of capable porous media combustion for the high temperature heat exchanger.

1.2 Research Objectives

To study parameters affecting combustion in porous materials

1. Impact of pm ball size

“ Porous media ball diameter size 5mm and porous media ball diameter size 10mm “

2. Impact of arrange pm ball size in burner

“ Arrange 1 and 2 size of porous media in burner “

3. Impact of fuel flow rate

“ 300 cc/min, 325 cc/min, 350 cc/min, 375 cc/min and 400 cc/min “

4. Impact of air flow rate

“ 15,000 cc/min and 20,000 cc/min “

5. Impact of equivalence ratio

“ Equivalence ratio of the fuel flow rate in the range 200-500 cc/min with air flow rate in the range 15,000-25,000 cc/min condition “

1.3 Scope of work

1.3.1 Scope of an equipment

1. Using the compact burner diameter size 67 mm, long 200 mm and rape with fiber paper insulator thickness 5 mm all length of burner and inside of burner was contained with porous media long 100 mm.

2. Using LPG fuel consisting of propane 70 % and butane 30 % by volume.

3. Using air for combustion.

4. Using data logger model GRAPHTEC GL800 for collect temperature data.

5. Using thermocouple type K for measure temperature in the burner (type K thermocouple operation range is $-270-1260$ °C)

1.3.2 Scope of an experiment

1. The material creates porosity in this experiment is Al_2O_3 in sphere shape with size 3mm, 5mm, and 10mm.
2. The maximum temperature not above 1200 °C
3. The fuel flow rate is in the range 175-525 cc/min
4. The air flow rate is in the range 15,000-25,000 cc/min
5. The arrangement of porous materials in the burner is randomly arranged.

1.3.3 Scope of an analysis

1. Porous media does not react to burns.
2. Chemical equation of LPG is $0.7\text{C}_3\text{H}_8 + 0.3\text{C}_4\text{H}_{10}$
3. Collecting temperature distribution according to the height of the porous combustion burner in each condition.
4. Compared distribution temperature of parameter porous media size between only diameter 5 mm and 10 mm.
5. Compared distribution temperature of parameter arrange porous media ball size in burner between 1 size and 2 size.
6. Compared distribution temperature of parameter fuel flow rate between 300, 325, 350, 375, 400 cc/min.
7. Compared distribution temperature of parameter air flow rate between 15,000, 20,000, 25,000 cc/min.
8. Compared distribution temperature of parameter equivalence ratio.

1.4 Expected benefits

- 1.4.1 A Prototype of the compact porous combustion burner for develop the porous combustion burner.
- 1.4.2 Develop the porous combustion burner efficiency.
- 1.4.3 Understand the burning behavior of the porous combustion burner using LPG fuel.

CHAPTER 2

LITERATURE REVIEW

2.1 History of porous media combustion development

Combustion in porous media was widely accepted in superior performance for many applications. For example, low heating value fuel burner, cooking stove burner, and the heat engine applications. The significant advantage is the higher combustion temperature due to the additional heat recirculation from radiation heat transfer of product downstream. Moreover, strong dispersion of the flow within porous media allow thermal energy to spread across the combustion zone, which resulted in a less hot spot in such area (Onthong, Boobchaay, & Wasinarom, 2017). Therefore, it is possible for flame stabilization of highly diluted, low heating value fuel. However, a specific operating condition needs further investigation, as it is significantly sensitive to the performance characteristic. Combustion in porous media is a complex physical phenomenon. It is not easy to investigate the experimental technique alone. It starts with the combustion of the mixture in a porous cavity. Convection heat transfer occurring between the gas cavity and the porous matrix is not thermally equilibrium. Conventional combustion stabilization technique makes use of the recirculation heat from hot products, which is in convection mode. However, the hot product would dilute the incoming mixture. Then, it is not possible to obtain product temperature higher than the adiabatic flame temperature of the mixture. In contrast, with combustion in porous media, radiation heat transfer from hot porous matrix to cold mixture region without mixing of fuel stream is the critical factor of flame stabilization (Voss et al., 2013). This feature of flame stabilization mechanism has distinguished it from the conventional burner. Moreover, it is possible to obtain super adiabatic flame temperature at some region because of non-diluted fuel stream (Coutinho & de Lemos, 2012).

Srilomsak et al. (Srilomsak, Aungkharuengrattana, Sesuk, Charoensuk, & Charochrojkul, 2015) investigated an arranged of various porous media materials for needed energy to fulfill the requirement of the reactor. The porous media material in this experiment is 25mm ceramic saddles, random size bio-filter media, ceramic foam, and ceramic balls. The highest thermal efficiency is using a ceramic foam, bio-filter media and ceramic saddles, respectively as porous media. The thermal efficiency of this

burner system is 18.86%. Most of the energy loss was due mainly to: 1) heat loss at the top of the reactor where the metal part was directly exposed to the atmosphere, 2) a large amount of energy loss at the furnace stack, and 3) an insufficient mixing at the early stage of combustion at the bottom of the furnace as noticed by high carbon monoxide (CO) concentration in the exhaust gas.

2.2 Fundamental of heat transfer

When two positions have different temperatures, they will have energy transferred from high temperature position to low temperature position. Energy is moving with the effect of different temperatures. This is called heat. Although thermodynamics is a study of energy transfer. However, it is only studied the heating system in a balanced state. Therefore, the rules in thermodynamic subjects can only be used to predict the amount of energy used to change the state of the heating system in a balanced state. However, it is unable to tell how fast the changing state occurred. The heat transfer can be used to define the rate of heat transfer.

Methods to be used in thermodynamics and heat transfer analysis is considered by the hot iron bar will heat up when it is soaked in hotter water. By the rules of thermodynamics, the temperature of the iron bar and hot water can be obtained when the iron bar and hot water are in a balanced state. But the heat transfer rate, the temperature in the given time period, and the time that the iron and hot water have the desired temperature cannot be determined. Analysis of heat transfer can predict the rate of heat transfer from hot water to steel bars. In addition, it can also calculate the temperature of steel bars and hot water in time term.

Heat can be transferred in 3 different modes (CENGEL, 2000), which are

2.2.1 Heat conduction

2.2.2 Heat convection

2.2.3 Heat radiation

2.2.1 Heat conduction

Conduction is the transfer of energy from more energetic particles to less energetic particles. When the temperature gradient occurs in an object. The rate of heat conduction q_k is proportional to the temperature difference, the heat transfer area, and inversely proportional to the thickness. That is,

$$\text{Rate of heat conduction} \propto \frac{(\text{Area})(\text{Temperature difference})}{\text{Thickness}}$$

Or,

$$q_k = kA \frac{T_1 - T_2}{\Delta x} = -kA \frac{\Delta T}{\Delta x} \quad (2-1)$$

Where, q_k = Rate of heat conduction, *Watt*
 A = The heat transfer Area, m^2
 T = The temperature difference across the layer, K
 x = The thickness of the layer, m
 k = The thermal conductivity of the material, $W/m^2 \cdot K$

Or reduces above equation to the differential form,

$$q_k = -kA \frac{dT}{dx} \quad (2-2)$$

which is called Fourier's law of heat conduction. The negative sign in Equation (2-2) ensures that heat transfer is a positive quantity because Heat is conducted the direction of decreasing temperature so the slope of the temperature curve on a T - x diagram in Figure 2-1 is negative quantity.

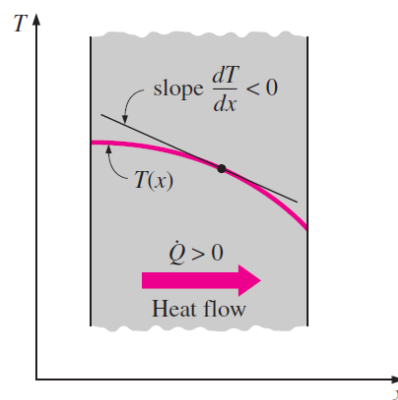


Figure 2-1 The direction of heat flow

Source: CENGEL, Y. A. (2000). HEAT TRANSFER: A Practical Approach. McGraw-Hill (Vol. 57)

2.2.2 Heat convection

Convection is the mode of energy transfer between a solid surface and the fluid. When the fluid move passes the surface of solid with a different temperature, it will transfer heat. This convection process is a common phenomenon. However, in reality, the convection process is very complicated. When currents are produced only by temperature-derived density differences in the fluid, it is known as natural convection. When the convection currents are due to an external factor such as a pump or fan, this is forced convection.

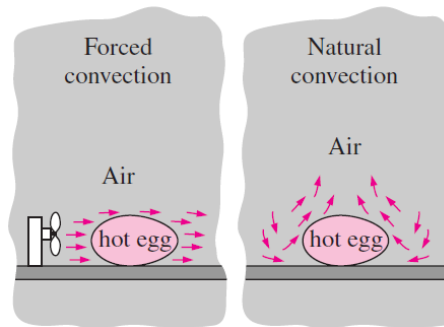


Figure 2-2 The cooling of a boiled egg by forced and natural convection

Source: CENGEL, Y. A. (2000). HEAT TRANSFER: A Practical Approach. McGraw-Hill (Vol. 57)

Despite the complexity of convection, the rate of convection heat transfer is observed to be proportional to the temperature difference, and is conveniently expressed by Newton's law of cooling as

$$q_c = \bar{h}_c A (T_s - T_{f,\infty}) \quad (2-3)$$

Where,

- q_c = Rate of heat conduction, *Watt*
- \bar{h}_c = The average convection heat transfer coefficient, $W/m^2 \cdot K$
- A = The surface area through which convection heat transfer takes place, m^2
- T_s = The surface temperature, K
- $T_{f,\infty}$ = The temperature of the fluid sufficiently far from the surface, K

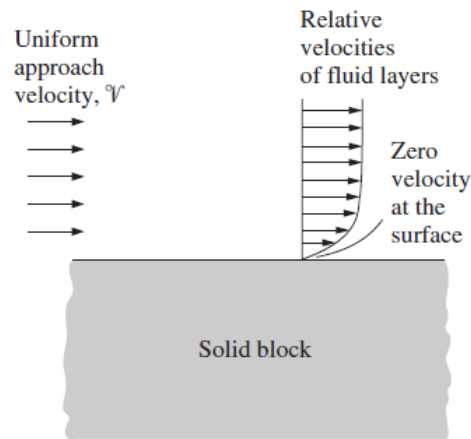


Figure 2-3 A fluid flowing over a stationary surface comes to a complete stop at the surface because of the no-slip condition

Source: CENGEL, Y. A. (2000). HEAT TRANSFER: A Practical Approach. McGraw-Hill (Vol. 57)

Point of observation of Figure 2-3 is the velocity that is reduced in the direction that enters the surface of the solid block with the effect of friction force due to the viscosity of the fluid. And the fluid in motion comes to a complete stop at the surface and assumes a zero-velocity relative to the surface. The heat transfer from the solid surface to the fluid layer adjacent to the surface is by pure conduction, and can be expressed as

$$q_{cond} = -K_f \left. \frac{\partial T}{\partial y} \right|_{y=0} = q_{conv} = \bar{h}_c (T_s - T_{f,\infty}) \quad (2-4)$$

Where, T = The temperature distribution in the fluid, K

$\partial T / \partial y|_{y=0}$ = The temperature gradient at the surface, K

From this point, it is possible to find $\partial T / \partial y|_{y=0}$ value at the surface of the object and the fluid layer attached to the surface that the heat conduction rate is equal to the heat convection rate.

The average convection heat transfer coefficient will depend on the density, viscosity and velocity of the fluid. Also, including depending on various thermal properties such as the thermal conductivity of the material k and the specific heat C_p . In the forced convection, the velocity of the fluid in system is determined by the pump or fan but the velocity of the fluid in the free convection will depend on the difference between the temperature of the surface of solid and the fluid, the coefficient of fluid expansion and the field of force, such as gravity, when the system is on the earth etc.

2.2.3 Heat radiation

Conduction and convection heat transfer requires a medium to pass through, but radiative heat transfer does not, such as in a vacuum area. The radiation heat energy will move in the form of electromagnetic waves that have speeds same to the speed of light 3×10^{10} cm/s. The energy that moves in the form of electromagnetic waves has many types, such as X-rays, infrared rays. But for this, only the heat energy from radiation is considered.

The amount of energy transmitted from the surface of the object in the form of heat radiation depends on the absolute temperature and the characteristics of the surface of the object in maximum radiation. Also known as a blackbody, the rate of heat radiation is

$$q_r = \sigma AT^4 \quad (2-5)$$

Where,
 q_r = The rate of heat radiation, *Watt*
 σ = The stefan-boltzman constant value is 5.67×10^{-8} , $W/m^2 \cdot K^4$
 A = The surface area of the object, m^2
 T = The surface temperature, K

The equation (2-5) is shown that the rate of heat radiation is direct variation with the absolute temperature and the surface area of the object emitting or absorbing thermal radiation. The rate of net radiation does not depend on the environment but depends on the temperature difference between the objects that are exchanging heat. If a blackbody radiates heat to the surrounding object and the surrounding object is blackbody that can absorb all the energy from the radiation that comes from another. The rate of heat transfer from the radiation of a blackbody with a temperature of T_1 to a blackbody at a temperature of T_2 is expressed as

$$q_r = \sigma A(T_1^4 - T_2^4) \quad (2-6)$$

Where,
 T_1 = The surface temperature of blackbody, K
 T_2 = The surface temperature of the surrounding blackbody, K

The real objects will not be able to radiate as much heat as the imaginary radiators, called blackbody. But if the real object can radiate heat same with blackbody in constant ratio. The radiate heat rate of real objects is expressed as

$$q_r = \sigma \varepsilon A T^4 \quad (2-7)$$

Where, ε = The emissivity of the surface, K

The property emissivity has value in the range $0 < \varepsilon < 1$, and a blackbody is 1.

2.2.4 Heat conduction in the sphere

For the heat conduction in the spherical coordinates $(r \ \theta \ \phi)$ from Figure 2-4, considering the coordinates of a point in rectangular and spherical coordinate systems:

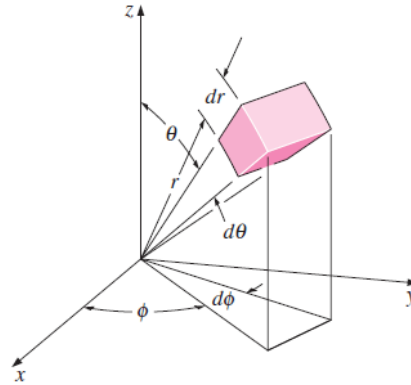


Figure 2-4 A differential volume element in spherical coordinates $(r \ \theta \ \phi)$

Source: CENGEL, Y. A. (2000). HEAT TRANSFER: A Practical Approach. McGraw-Hill (Vol. 57)

$$\begin{aligned} \frac{1}{r^2} \frac{\partial}{\partial r} \left(k r^2 \frac{\partial t}{\partial r} \right) + \frac{1}{r^2 \sin^2 \theta} \frac{\partial}{\partial \phi} \left(k \frac{\partial t}{\partial \phi} \right) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \theta} \left(k \sin \theta \frac{\partial T}{\partial \theta} \right) + q_0 \\ = \rho C_p \frac{\partial t}{\partial t} \end{aligned} \quad (2-8)$$

In the case of heat conduction equation in one dimensional

$$\frac{1}{r^2} \frac{\partial}{\partial r} \left(k r^2 \frac{\partial t}{\partial r} \right) + q_0 = \rho C_p \frac{\partial t}{\partial t} \quad (2-9)$$

In the case of heat conduction equation in one dimensional, steady-state condition and no heat generation:

$$\frac{1}{r^2} \frac{d}{dr} \left(k r^2 \frac{dt}{dr} \right) + q_0 = 0 \quad (2-10)$$

In the case of heat conduction equation in one dimensional, steady-state condition, no heat generation and k are constant value:

$$\frac{1}{r^2} \frac{d}{dr} \left(r^2 \frac{dt}{dr} \right) + \frac{q_0}{k} = 0 \quad (2-11)$$

2.3 Fundamental of combustion

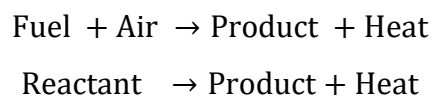
Combustion is an oxidation reaction at high temperatures between fuels or substances that can burn quickly with oxygen and exude heat during the reaction. Substances that can be burned in fuel are carbon (C), hydrogen (H) and sulfur (S). Combustion can be divided into 3 types, which are

- 1 Solid fuel combustion
- 2 Liquid fuel combustion
- 3 Gaseous fuel combustion

This study only focuses on solid fuel combustion. The principles of combustion reactions that occur in elements not different in all three types of combustion.

2.3.1 Combustion reaction

The combustion reaction is an oxidation reaction between the various components in the fuel (C H N O S) with oxygen in the air and the product of combustion in the form of combustion gases. The heat from the combustion reaction is shown in the equation.



2.3.2 Factors involved in combustion

2.3.2.1 Sufficient air to combustion: The amount of air that is sufficient to fuel combustion is required to have sufficient air or oxygen proportions for combustion reactions. The minimum amount of air that can cause complete combustion is called theoretical air, which can be found from the theoretical complete combustion reaction equation of carbon, hydrogen and sulfur above.

2.3.2.2 Burning time: The reaction combustion, it takes time to get a complete combustion reaction. In particular, the burning in the combustion burner requires that

the fuel is burned for a long period of time to be reacted to burn out completely. If the burning time is not enough, will result in low fuel efficiency and low fuel consumption.

2.3.2.3 Combustion temperature: Fuel combustion is a chemical reaction which the speed of the reaction depends on the temperature of the reaction. If the combustion temperature is high, the speed of combustion will also be high as well. But in the burning process, it is necessary to control the temperature to the extent that the furnace material can be high heat resistant. Which is about 1500 °C.

2.3.2.4 Turbulence: Fuel combustion is a reaction of fuel and oxygen or air. If during the burning have turbulence in the burner, A good mix of fuel and oxygen causes more complete combustion.

2.3.3 The theoretical ratio of excess air and fuel

From the factors involved combustion above, it was found that the most important for complete combustion is sufficient air volume. And the factors for complete combustion, which is called the 3T factor; Time, Temperature and Turbulence. Therefore, in fuel combustion, it is necessary to know the sufficient amount of air (theoretical air) to theoretical combustion. Each type of fuel uses air volume per fuel volume (air to fuel ratio; A/F) in theoretical combustion is different. In addition, the amount of air used in the actual fuel combustion may be greater than the theoretical air volume. In fact, when using the air volume equal to the theory, the chance that the oxygen will react with the fuel in complete combustion is not much or take a very long time to combustion completely. Therefore, it is necessary to supply air in quantities greater than the theory called excess air. For the effects of air volume or air-to-fuel ratio on combustion are as follows:

2.3.3.1 If the exact amount of air in combustion is less than the theoretical air ($A / F \text{ actual} < A / F \text{ theoretical}$), Incomplete combustion occurs that cause low heat and cause pollution such as CO occurs.

2.3.3.2 If the exact amount of air in combustion is more than the theoretical air ($A / F \text{ actual} > A / F \text{ theoretical}$), The chance of complete combustion has increased. But the excess air condition will cause some amount of heat from the reaction to burning O_2 , N_2 to rise temperature.

2.3.4 Determination of excess air

For finding excess air, it can be done in 2 ways:

1. Measuring the amount of air entering the actual combustion reaction ($A / F \text{ actual}$)

2. Measuring the amount of oxygen remaining from the combustion at the exit of the combustion furnace and then calculating back to find the amount of air remaining from the combustion, which considers that this air is the air that is inserted beyond the requirement. In general, the excess air volume explains in percentage. The percentage of excess air (% Excess air) or the percentage of theoretical air (% Theoretical air) which can be calculated as follows.

$$\%EA = \left(\frac{A/F_{ac} - A/F_{th}}{A/F_{th}} \right) \times 100\% \quad (2-12)$$

2.4 The principles of porous material

The material of the porous material that is used as a component in the combustion burner is ceramics with alumina mixture.

2.4.1 Heat transfer in porous material

The porous material is a material that has a high surface area per volume, high temperature resistance, and has a significant contribution to combustion, which causes energy recirculation. In the system, the porous material will absorb some enthalpy of hot product and then radiating heat back to the source of the burner and because the porous material has a high surface area to volume ratio, heat transfer is highly effective too. Currently, the porous material has been produced in various forms due to the use of porous materials to increase the efficiency of the burners. And also use porous materials for other thermal devices such as heat engines, power generation equipment using temperature differences, high temperature heat exchangers.

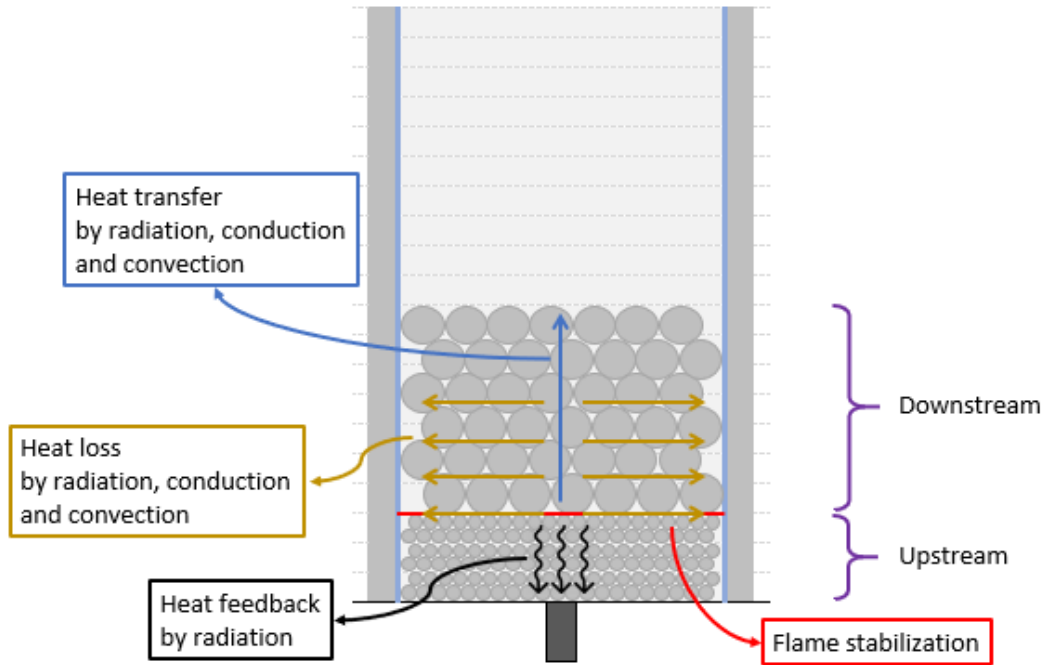


Figure 2-5 Schematic of a typical two-layer premixed PM burner, showing the major heat transfer modes and directions

Figure 2-5 The principles of heat transfer in porous media is when hot gas from combustion flows through porous materials, the porous material will absorb heat or enthalpy from the hot gas and radiant heat to the porous material on the upstream. Therefore, when the air and fuel with low temperatures flow through, it will get heat with enthalpy that the porous material absorbs. The air and fuel have a higher enthalpy value. As mentioned above, the combustion system under the heat radiation of porous materials has a high combustion efficiency and the fuel economy to the combustion system.

2.4.2 Heat recirculating combustion in porous material

Heat recirculating combustion system is a high efficiency combustion system, energy saving, and low CO release. At present it widely used in industrial plants because this system, when applied, can increase combustion efficiency and save energy efficiently.

2.4.2.1 Heat recirculating combustion operation :

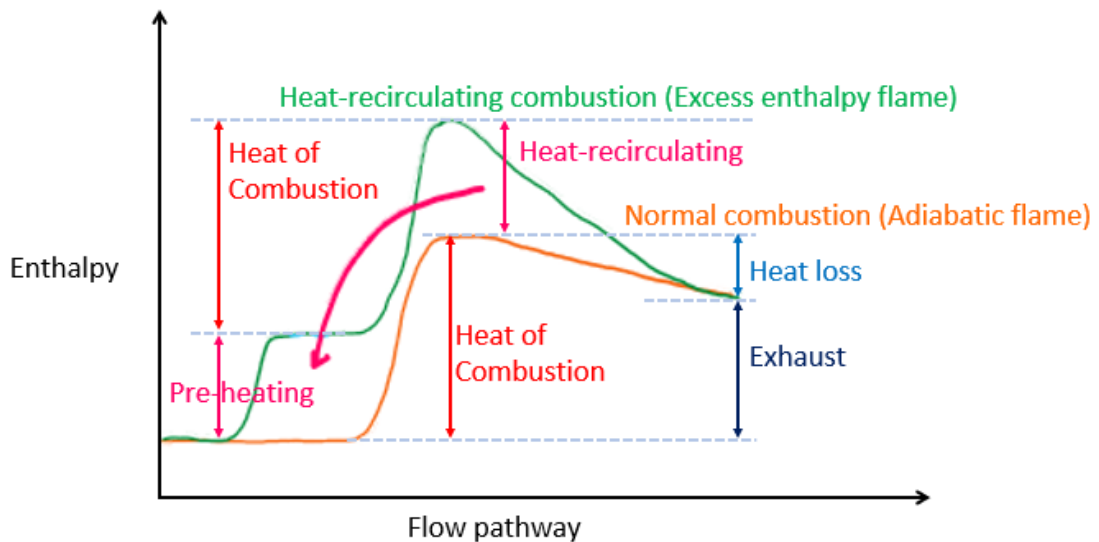


Figure 2-6 Comparison of operation the system combustion between with and without heat circulation

Figure 2-6, enthalpy history without heat recirculation, Fuel flow through the intake into the left-hand side and the burning occurs, the combustion temperature will be increased. The exhaust gas generated in the system will flow out on the right-hand side, with the heat from the exhaust gases that not being stored. Compare with enthalpy history which shows the system that has heat circulation from the exhaust gas to preheat the intake gas. The combustion with continuous preheat have higher combustion temperatures and combustion rates than the combustion without preheating. And in the case of same the highest temperature of the combustion, it was found that the amount of fuel used in the combustion with the heat recirculation system uses less than the combustion without heat recirculation, thus saving a lot of energy. In addition, the combustion with heat recirculation also has a high burning velocity and the high combustion intensity. The heat recirculation system can be designed for a combustion burner to be small, compact, and also help to expand the flammability limits. And it is suitable for a burner that cannot be burned in very little fuel condition

2.4.2.2 Heat circulation by thermal deformation during convection and radiate heat in porous materials :

The porous material is a high surface area, high heat transfer coefficient, high heat absorption coefficient and high resistant temperature. Important properties of porous materials are able to change energy reverse between the enthalpy of gas and heat radiation because the porous material has a high surface area to volume ratio. Therefore, the heat transfer is highly effective. The working principle is when hot gas flows

through the right porous material as shown in Figure 2-7. The porous material absorbs a part of the hot gas enthalpy. And changing this absorbed enthalpy to heat radiation which calls the emitter. Radiation counterflow against the flow of hot gas to the porous material on the other side that as the absorber. The hot air flows through the emitter will decrease the temperature and when the cold air flows through the absorber that absorbs the heat absorbed, the air has an enthalpy and the temperature increases. In such cases, if the emitter and the absorber adjacent together, the high-temperature exhaust gas that flows through the porous material that acts as the emitter, then the cold air flows through the porous material that acts as the absorber that effect to the circulation of heat in system burn efficiently by using porous materials as medium.

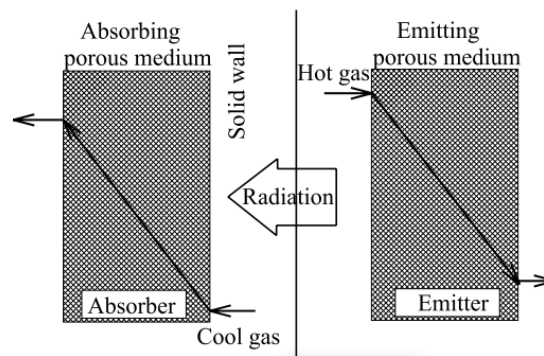


Figure 2-7 Heat exchanger based on energy conversion between convection and thermal radiation by porous material

Source: Jugjai, S., & Rungsimuntuchart, N. (2002). High efficiency heat-recirculating domestic gas burners High efficiency heat-recirculating domestic gas burners. *Experimental Thermal and Fluid Science* 26, 1777(July)

2.4.2.3 Equation of preheat air temperature :

When the porous material is heated by heat from the hot gas that generated by the combustion, and counter air flow direction, the porous material will transfer heat to the air, The porous material will transfer heat to the air then the temperature to decrease by time. Considering the porous material as a lump. Can find the reduced temperature of the porous materials from the following theory;

The heat equation from the porous material is transferred to the air by convection.

$$Q = hv(T_s - T_\infty) \quad (2-13)$$

In the case of one dimensional and unsteady heat transfer

$$-qdt = mc_p dT_s \quad (2-14)$$

From the equation (2-13) and (2-14). The heat from air to the porous material is equal to the heat that the porous material absorbs.

$$-\frac{dT_s}{T_s - T_\infty} = \frac{hv}{mc_p} dt \quad (2-15)$$

Integrate the equation (2-15)

$$\ln(T_s - T_\infty) - \ln(T_{S_0} - T_\infty) = -\frac{hv}{mc_p} t \quad (2-16)$$

Find the heat value of porous material T_s from the equation (2-16). And putting the T_s value into the equation (2-14), will know the heat value q that the porous material absorbs. Then find the total heat value Q_{conv} that the porous material absorbs by convection, From the exhaust heat in each half of period t_{hp} from the equation

$$Q_{conv} = \int_{t=0}^{t=t_{hp}} q dt \quad (2-17)$$

The part of total heat convection Q_{conv} will be lost into make the liquid fuel evaporate completely at 250 °C. This heat which can be calculated as follows.

$$Q_{evap} = \dot{m}c_p\Delta T + \dot{m}LHV \quad (2-18)$$

Therefore, the heat that the porous material will be used to preheat air.

$$Q = Q_{conv} - Q_{evap} \quad (2-19)$$

The heat Q from the equation (2-19) will be used to find the preheated air temperature which can be calculated as follows.

$$T_{pre} = T_{\infty} + \frac{Q}{\dot{m}t_{hp}c_p} \quad (2-20)$$

2.4.3 Ceramic porous media

The advantages of the ceramic material are that it is resistant to temperature, meaning it is resistant to thermal changes. Also has high thermal conductivity, resulting in a temperature distribution that does not occur in hot spots. The ceramic is used as a ceramic ball.

The material of Porous media burner (P. Chitmongkol, 2014), Aluminum oxide (Al_2O_3), silicon carbide (SiC), and zirconium dioxide (ZrO_2) proposed as suitable materials for the application. Al_2O_3 and ZrO_2 were recognized as high temperature resistant materials. SiC shows excellent thermal shock resistance, mechanical strength, and conductive heat transport. SiC also has a high melting point (3260 K), against cyclic thermal stress and strength retention at the peak regenerator temperature (1673 K), and excellent oxidation resistance. Metallic materials were found less suitable for PM because of their inadequate thermal stability and high thermal inertia.

Aluminium oxide (Al_2O_3) can be used up to approximately 1900°C under air. High SiO_2 contents between 20 and 40% reduce the application limit to approximately 1650°C. Because of its high thermal expansion coefficient and an average thermal conductivity, the resistance to a temperature change of Al_2O_3 ceramics is relatively low. Silicon carbide (SiC) can also be used at very high temperatures, usually up to 1600°C. The resistance to a temperature changes of SiC ceramics is excellent, which is due to the low expansion coefficients and the high thermal conductivity.

Zirconium oxide (ZrO_2) can be used up to 2300°C but a change associated with a volume increase during cooling can destroy the structure. The high-temperature modification is stabilized by incorporating various additives. In this way, fully and partially stabilized ZrO_2 can be used up to approximately 1800°C. The high thermal expansion and the low thermal conductivity have a negative effect on the resistance to temperature change.

A comparison of the data for materials relevant for use in Porous media burners is shown in Table 2-1

Table 2-1 Most important material data for Al_2O_3 , SiC and ZrO_2

Property	Unit	Al ₂ O ₃	SiC	ZrO ₂
Maximum use temperature in air	°C	1900	1650	1800
Thermal expansion coefficient α (20-1000 °C)	10 ⁻⁶ K ⁻¹	8	4-5	10-13
Thermal conductivity λ at 20 °C	W m ⁻¹ K ⁻¹	20-30	80-150	2-5
Thermal conductivity λ at 1000 °C	W m ⁻¹ K ⁻¹	5-6	20-50	2-4
Specific thermal capacity	J g ⁻¹ K ⁻¹	0.9-1	0.7-0.8	0.5-0.6
Thermal stress resistance parameter, hard shock, R ($\sigma/E\alpha$)	K	100	230	230
Thermal stress resistance parameter, mild thermal shock, R' ($R \cdot \lambda$)	10 ⁻³ W m ⁻¹	3	23	1

Source: Avdic, F. (2004). Application of the Porous Medium Gas Combustion Technique to Household Heating Systems with Additional Energy Sources. Universität Erlangen-Nürnberg.

2.5 Characteristic of fuel gas

The main fuel gas that is currently used is natural gas(CNG) and liquefied petroleum gas(LPG). In addition, fuel gas can also be produced from coal, wood, natural gas and petroleum.

2.5.1 Type of fuel gas

Natural gas is often found near or over oil fields in rocky areas and compressed with high pressure in the underground. Natural gas is a mixture of various hydrocarbons such as methane, ethane, propane, butane and pentane. Methane is the most common. There will be other types of gas as a minority and sometimes found carbon dioxide and nitrogen gas. Natural gas can be used by compressing and cooling until heavy hydrocarbon compounds become liquids and then separate a dry and high-pressure natural gas sent along the pipe for use. In sometimes, natural gas is cooled to -164°C until it becomes a liquefied natural gas (LNG) and then transported to the ship.

Liquefied petroleum gas (LPG) produced from natural gas in a gas separation plant that has main components ethane, propane and butane. In addition, LPG also comes from liquefied gases obtained from crude oil refining in oil refineries such as ethylene, propylene and butylene. LPG transportation can be done by filling tanks under normal pressure and LPG will become vapor under standard atmosphere. Producer gas is a gas produced by releasing air in quantities less than sub stoichiometric gas through the hot layers of coal, peat, wood or agricultural leftovers. The main components of producer

gas are carbon monoxide, hydrogen, nitrogen (up to 55%) and small amounts of carbon dioxide. The user may be used as soon as it is produced. While still hot or purified to remove crude oil and soot. However, the producer gas has a low heat value because there is a lot of nitrogen gas, so if you want to increase the heat value, you need to use oxygen less than the theory.

2.5.2 Properties of LPG

2.5.2.1 Chemical properties

The complexity of LPG components containing the lower number of carbon fuels. This hydrocarbon group consists of

Propane (C ₃ H ₈)	Propylene (C ₃ H ₆)
Butane (C ₄ H ₁₀)	Butylene (C ₄ H ₈)

Moreover, compound with a few non-hydrocarbon compounds due to the variety of components. Thus, setting the standard of LPG used as fuel for information in designing for correctly and safely. However, LPG gas may be divided into two groups: Saturated and Unsaturated, different in a covalent bond between carbon atoms.

Saturated hydrocarbon → Propane, N-Butane, iso-Butylene

Unsaturated hydrocarbon → Propylene, N-Butylene, iso-Butylene

2.5.2.2 Physical properties

When considered LPG in the liquid phase, LPG has a shallow boiling point (propane 42°C, butane 0.5°C, isobutane 11.72°C). Therefore, LPG is a gas state in the atmosphere. Except in cases of LPG inside the tank, the state of LPG is liquid.

Table 2-2 LPG properties in liquid status

Property	propane	butane	isobutane
Molecular weight	44.08	58.12	58.12
Boiling point (°C at 1atm)	42.05	0.50	11.72
Critical temperature (°C)	96.67	152.03	134.99

Source: Lapidattanakun, A. (2017). Influence of air staging and steam assisted atomization on combustion characteristics in porous media. KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG.

When considering LPG in the gas phase, the combustion range of the combustion gases will have only one point that ignites and burns because the air is mixed with fuel in

moderation. The combustion range shows the percentage of gas volume per air. LPG is a mixture of propane and butane, ignition range in air of propane having 2.4-9.5 % and ignition range in air of propane having 1.8-8.4 %. If air/fuel mixture is less or more than this value, ignition does not occur.

2.5.3 Ignition temperature of LPG

When gradually increasing the temperature to the fuel until the lowest temperature at which the fuel will ignite by itself without the presence of a spark or flame. The lowest temperature of this natural ignition is called the ignition temperature. The ignition temperature of propane is 470 °C, And the ignition temperature of butane is 405 °C.

2.5.4 Adiabatic flame temperature of LPG

The highest temperature a system can achieve at constant pressure when there are no heat losses from the system. Adiabatic flame temperature of propane is 1930 °C, And the adiabatic flame temperature of butane is 2246 °C.

2.5.5 Volume ratio of liquid and gas of LPG

LPG in the gas phase, the volume will change dramatically by volume 1 unit of propane when becomes a gas then the volume will be 2 units at 15.5 °C. The volume 1 unit of butane when becomes a gas then the volume will be 233 units at 15.5 °C.

2.5.6 Heating value of combustion of LPG

The heat value of the combustion means the amount of heat generated by gas 1 unit of gas or 1 unit of volume burn at normal pressure and temperature 25 °C.

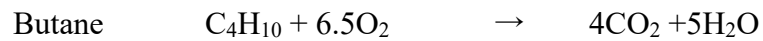
Table 2-3 Heat value of gas combustion

Temperature at 25°C	Unit	Propane	n-Butyl	iso-Butyl
Gross	Kcal / Kg	11943	11742	11714
	BTU / pound	21497	21135	21085
	Kcal / cubic m.	23700	30600	30500
	BTU / cubic feet	2663	3438	3427
Net	Kcal / Kg	10989	10837	10810
	BTU / pound	19780	19506	19458
	Kcal / cubic m.	22250	29400	29200
	BTU / cubic feet	2500	3304	3281

Source: Lapirattanakun, A. (2017). Influence of air staging and steam assisted atomization on combustion characteristics in porous media. KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG.

2.5.7 Air requirement

Oxygen is a gas that is mixed in the air by 21% by volume and is an important factor for burning. Therefore, the amount of air inputted into the combustion burner must have a certain volume. In the case of LPG gas complete combustion, all gas will become carbon dioxide. Written as a chemical equation as follows.



From a chemical equation, the volume of oxygen needed to complete combustion in the case of propane is 5 times and in the case of propane/butane is 6.5 times. The volume of oxygen in the air will be 21%. Therefore, in burning propane completely, 1 cubic meter will have to use the air 24 cubic meters and for butane 1 cubic meter will have to use air 31 cubic meters.

CHAPTER 3
RESEARCH METHODOLOGY

3.1 Calculation of equivalence ratio in air theory volume

The air-fuel equivalence ratio (AFR, λ) of equation is

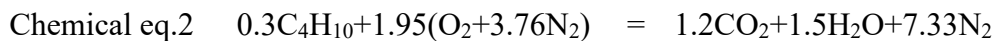
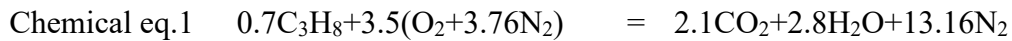
$$\text{AFR, } \lambda = \frac{\text{inlet air stoi/fuel stoi}}{\text{inlet air fact/fuel fact}} \quad (3-1)$$

Air volume in the theory of LPG

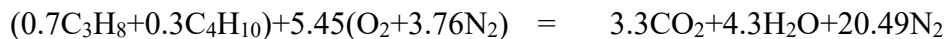
LPG has propane $\text{C}_3\text{H}_8 = 70\%$ by volume

LPG has butane $\text{C}_4\text{H}_{10} = 30\%$ by volume

The proportion by the mole of LPG in air theory volume written as the chemical equation as follows



Chemical equation 1 + chemical equation 2 become LPG chemical equation



\therefore LPG or $0.7\text{C}_3\text{H}_8 + 0.3\text{C}_4\text{H}_{10}$ 1 mole has mass = $1 \text{ mol} \times \{0.7[(12 \times 3) + (1 \times 8)] + 0.3[(12 \times 4) + (1 \times 10)]\} \frac{\text{g}}{\text{mol}} \times \frac{1}{1000} \frac{\text{kg}}{\text{g}} = 0.0482 \text{ kg}$ and mass of C_3H_8 is 70% of 0.0482 kg or 0.0308 kg, mass of C_4H_{10} is 30% of 0.0482 kg or 0.0174 kg

In case of using

1. LPG 300 cc/min or $7.38 \times 10^{-6} \text{ kg/sec}$

2. air 20,000 cc/min or $3.75 \times 10^{-4} \text{ kg/sec}$

Calculate of FUEL at stoichiometric and fact

Mass of fuel at stoichiometric and fact is mass of using LPG $7.38 \times 10^{-6} \text{ kg}$

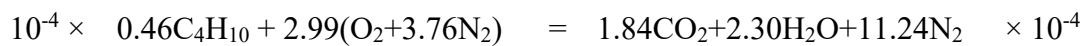
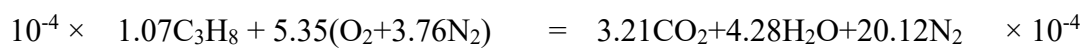
Calculate of air at stoichiometric and fact

Mass of air at stoichiometric is calculated from the proportion of LPG in case of using LPG 300 cc/min in stoichiometric condition.

C₃H₈ mass of LPG in 1 second is 70% of 7.38 ×10⁻⁶ kg or 4.72 ×10⁻⁶ kg or 1.07 ×10⁻⁴ mole

C₄H₁₀ mass of LPG in 1 second is 30% of 7.38 ×10⁻⁶ kg or 2.66 ×10⁻⁶ kg or 4.59 ×10⁻⁵ mole

The proportion by the mole of LPG in case of using LPG 300 cc/min in stoichiometric condition written as the chemical equation as follows



Mass of air at stoichiometric in case of using LPG 300 cc/min is 5.35×10⁻⁴ + 2.99×10⁻⁴ mole = 8.34×10⁻⁴ mole or 1.15×10⁻⁴ kg

Mass of air at fact, in this case, using air 20,000 cc/min is 3.75×10⁻⁴ kg

∴ From equation (3-1) AFR,λ that is using LPG 300 cc/min and air 20,000 cc/min is

$$= \frac{1.15 \times 10^{-4} / 7.38 \times 10^{-6}}{3.75 \times 10^{-4} / 7.38 \times 10^{-6}} = 0.31$$

In the same method, the value of all air-fuel equivalence ratio in the experiment is in Table 3-1

Table 3-1 Summary of air-fuel equivalence ratio in each experiment conditions

LPG (cc/min air)	air (cc/min air)	AFR, λ
200	15,000	0.27
225	15,000	0.31
250	15,000	0.34
275	15,000	0.37
300	15,000	0.41
275	20,000	0.28
300	20,000	0.31
325	20,000	0.33
350	20,000	0.36
375	20,000	0.38
400	20,000	0.41
375	25,000	0.31
400	25,000	0.33
425	25,000	0.35
450	25,000	0.37
475	25,000	0.39
500	25,000	0.41

3.2 Calculation of the excess air ratio

The excess air ratio (EA) of equation is

$$EA = \frac{\left(\frac{\text{Inlet air fact}}{\text{Fuel fact}} - 1\right)}{\frac{\text{Inlet air stoi}}{\text{Fuel stoi}}} \quad (3-2)$$

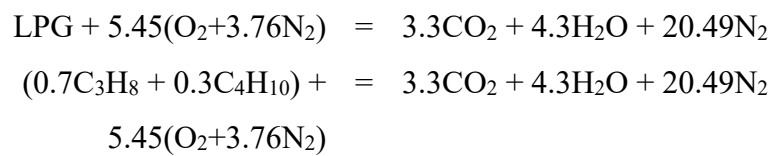
Use the value of the variable from section 3.2 and calculate the excess air ratio from the equation (3-2). Therefore, the value of all the excess air ratio in experiment is in Table 3-2

Table 3-2 Summary of the excess air ratio in each experiment conditions

LPG (cc/min air)	air (cc/min air)	EA
200	15,000	3.6
225	15,000	3.2
250	15,000	2.9
275	15,000	2.6
300	15,000	2.4
275	20,000	3.5
300	20,000	3.2
325	20,000	3.0
350	20,000	2.7
375	20,000	2.6
400	20,000	2.4
375	25,000	3.2
400	25,000	3.0
425	25,000	2.8
450	25,000	2.7
475	25,000	2.5
500	25,000	2.4

3.3 Calculation of the adiabatic flame temperature in air theory

The proportion by the mole of LPG written as the chemical equation as follows



The formulation of chemical reaction in steady flow and the components that enters the combustion burner at temperature 298 K and pressure 1 atm (Turns, 2000).

$$\begin{aligned}
 H_{\text{react}} &= H_{\text{prod}} \\
 \sum N_r (\bar{h}_f^0 + \bar{h} - \bar{h}^0)_{\text{react}} &= \sum N_p (\bar{h}_f^0 + \bar{h} - \bar{h}^0)_{\text{prod}}
 \end{aligned}$$

Where, \bar{h}_f = Enthalpy of formation

\bar{h} = Enthalpy at temperature

\bar{h}^0 = Enthalpy at 298 K

Table 3-3 Table of enthalpy values at standard reference conditions 298 K at 1 atm

Substance	h_f^0 (kJ/kmol)	$\bar{h} - \bar{h}_{298K}$
H ₂	0	$\bar{h}_{H_2} - 8,468$
O ₂	0	$\bar{h}_{O_2} - 8,682$
N ₂	0	$\bar{h}_{N_2} - 8,669$
CO ₂	-393,546	$\bar{h}_{CO_2} - 9,364$
H ₂ O _(gas)	-241,845	$\bar{h}_{H_2O(g)} - 9,904$
H ₂ O _(liquid)	-285,830	$\bar{h}_{H_2O(l)} - 9,904$
C ₃ H ₈	-103,847	0
C ₄ H ₁₀	-124,733	0

Source: Turns, S. R. (2000). An introduction to combustion: concepts and applications. System. <https://doi.org/10.1016/j.ijhydene.2008.07.121>

Substitute,

$$\begin{aligned}
 & N_{CO_2}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{CO_2} + N_{H_2O}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{H_2O} \\
 & \quad + N_{N_2}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{N_2} \\
 & \quad = \\
 & N_{C_3H_8}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{C_3H_8} + N_{C_4H_{10}}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{C_4H_{10}} \\
 & \quad + N_{O_2}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{O_2} + N_{N_2}(\bar{h}_f^0 + \bar{h}_{ad} - \bar{h}_{298K})_{N_2} \\
 & 3.3(-393,546 + \bar{h}_{ad} - 9,364)_{CO_2} + 4.3(-241,845 + \bar{h}_{ad} - 9,904)_{H_2O} \\
 & \quad + 20.49(0 + \bar{h}_{ad} - 8,669)_{N_2} \\
 & \quad = \\
 & 0.7(-103,847 + 0)_{C_3H_8} + 0.3(-124,733 + 0)_{C_4H_{10}} + 5.45(0 + 0)_{O_2} \\
 & \quad + 20.3(0 + 0)_{N_2}
 \end{aligned}$$

$$N_{C_3H_8}(\overline{h_f^0} + \overline{h_{ad}} - \overline{h_{298K}})_{C_3H_8} + N_{C_4H_{10}}(\overline{h_f^0} + \overline{h_{ad}} - \overline{h_{298K}})_{C_4H_{10}} \\ + N_{O_2}(\overline{h_f^0} + \overline{h_{ad}} - \overline{h_{298K}})_{O_2} + N_{N_2}(\overline{h_f^0} + \overline{h_{ad}} - \overline{h_{298K}})_{N_2}$$

$$5.05 \times 10^{-2}(-393,546 + \overline{h_{ad}} - 9,364)_{CO_2} \\ + 6.58 \times 10^{-4}(-241,845 + \overline{h_{ad}} - 9,904)_{H_2O} \\ + 1.03 \times 10^{-2}(0 + \overline{h_{ad}} - 8,669)_{N_2} \\ + 3.8 \times 10^{-3}(0 + \overline{h_{ad}} - 8,682)_{O_2}$$

=

$$1.07 \times 10^{-4}(-103,847 + 0)_{C_3H_8} + 4.59 \times 10^{-5}(-124,733 + 0)_{C_4H_{10}}$$

Assume the gas temperature is 1000 K = 474.05

Assume the gas temperature is 1020 K = 484.53

Estimate the range of the gas temperature to find the adiabatic flame temperature

∴ T_{ad} = 1,000.90 K or 727.90 °C

Table 3-4 Summary of the adiabatic flame temperature in excess air condition

LPG (cc/min air)	air (cc/min air)	AFR, λ	T _{ad} (K)
200	15,000	0.27	927.17
225	15,000	0.31	1000.90
250	15,000	0.34	1073.81
275	15,000	0.37	1145.97
300	15,000	0.41	1217.43
275	20,000	0.28	945.68
300	20,000	0.31	1000.90
325	20,000	0.33	1055.64
350	20,000	0.36	1109.98
375	20,000	0.38	1163.90
400	20,000	0.41	1217.43
375	25,000	0.31	1000.90
400	25,000	0.33	1044.72
425	25,000	0.35	1088.30
450	25,000	0.37	1131.59
475	25,000	0.39	1174.61
500	25,000	0.41	1217.43

3.5 Calculation of the heating value (HHV, LHV)

Higher heating value (HHV) or gross calorific value is determined by bringing all the products of combustion back to the original pre-combustion temperature, and in particular condensing any vapor produced.

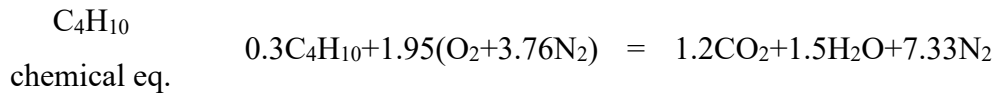
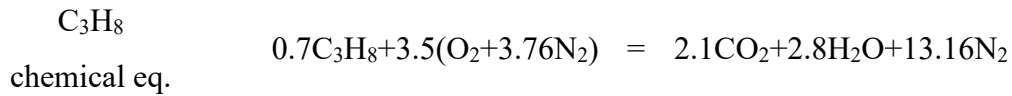
Lower heating value (LHV) or net calorific value is determined by subtracting the heat of vaporization of the water from the higher heating value

$$\text{LHV} = \text{HHV} - (N_{\text{H}_2\text{O}} \times \overline{h_{f,\text{H}_2\text{O}}^0}) \quad (3-3)$$

Where, $N_{\text{H}_2\text{O}}$ = the mole of water formed by combustion

$\overline{h_{f,\text{H}_2\text{O}}^0}$ = the latent heat of vaporization of water, kJ/kmol

The proportion by the mole of LPG in air theory written as the chemical equation as follows



Substitute in the equation (3-3),

For C_3H_8

$$\begin{aligned} \text{LHV}_{\text{C}_3\text{H}_8} &= (N_{\text{C}_3\text{H}_8} \times \overline{h_{f,\text{C}_3\text{H}_8}^0}) - (N_{\text{H}_2\text{O}} \times \overline{h_{f,\text{H}_2\text{O}}^0}) \\ &= [0.7 \times (-103,847)] - [2.8 \times (-241,845)] \\ &= 604,473.10 \text{ kJ/kmol} \\ &= 13,738.02 \text{ kJ/kg} \end{aligned}$$

For C_4H_{10}

$$\begin{aligned} \text{LHV}_{\text{C}_4\text{H}_{10}} &= (N_{\text{C}_4\text{H}_{10}} \times \overline{h_{f,\text{C}_4\text{H}_{10}}^0}) - (N_{\text{H}_2\text{O}} \times \overline{h_{f,\text{H}_2\text{O}}^0}) \\ &= [0.3 \times (-124,733)] - [1.5 \times (-241,845)] \\ &= 325,347.60 \text{ kJ/kmol} \\ &= 5,609.44 \text{ kJ/kg} \end{aligned}$$

$$\therefore \text{LHV}_{\text{LPG}} = \text{LHV}_{\text{C}_3\text{H}_8} + \text{LHV}_{\text{C}_4\text{H}_{10}} = 13,738.02 + 5,609.44 = 19,347.47 \text{ kJ/kg}$$

In case of using

$$\text{- LPG } 300 \text{ cc/min or } 7.38 \times 10^{-6} \text{ kg/sec}$$

$$\therefore \text{Heat power is } 19,347.47 \text{ kJ/kg} \times 7.38 \times 10^{-6} \text{ kg/sec} = 0.14 \text{ kJ/sec, kwatt}$$

Diameter of burner is 0.075m, Cross area is 0.004418 m^2

$$\therefore \text{Heat power per area is } 0.14 \text{ kJ/sec} / 0.004418 \text{ m}^2 = 32.32 \text{ kwatt/m}^2$$

3.6 Experimental systems and equipment

3.6.1 Burner system setup

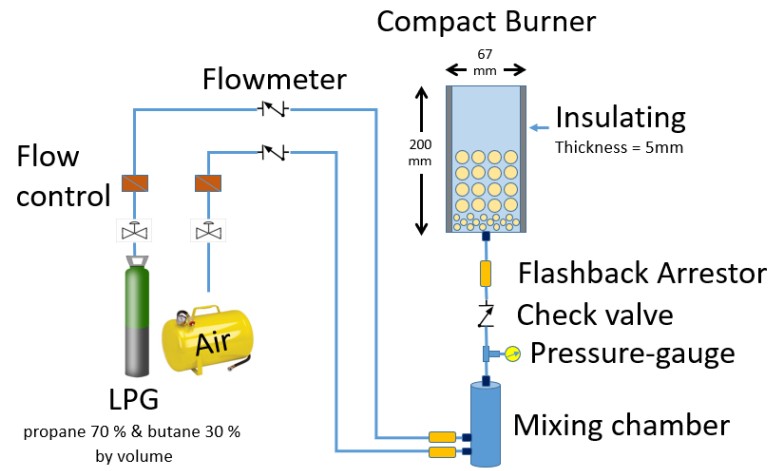


Figure 3-1 Schematic of the experimental setup



Figure 3-2 Photo of the experimental setup

Figure 3-1 and Figure 3-2 shows the schematic and photo of the experimental setup. This system is composed of LPG gas and air supply systems, a mixing chamber, three sizes of porous media, and a measurement system. MFC controlled LPG gas and air and then mixed in the mixing chamber. The LPG/air mixture flowed through flashback protection and after that combustion in a stainless tube 67 mm. in diameter and 200 mm. long and wrap with fiber paper insulating 0-200 mm.

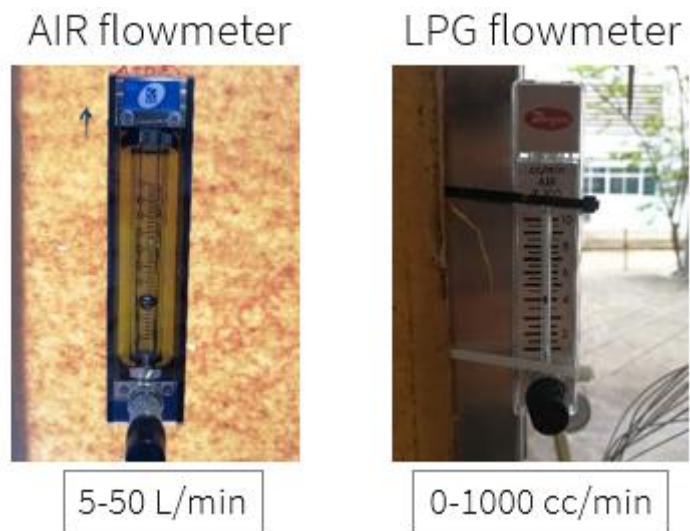


Figure 3-3 Photo of the rotameter for air and LPG

Figure 3-3 In this system, Flowmeter that uses in this system is rotameter. The scale of rotameter for air is 5-50 L/min and scale of rotameter for LPG is 0-10 x100 AIR cc/min

3.6.2 Measurement setup



Figure 3-4 Photo of setting thermocouples in system

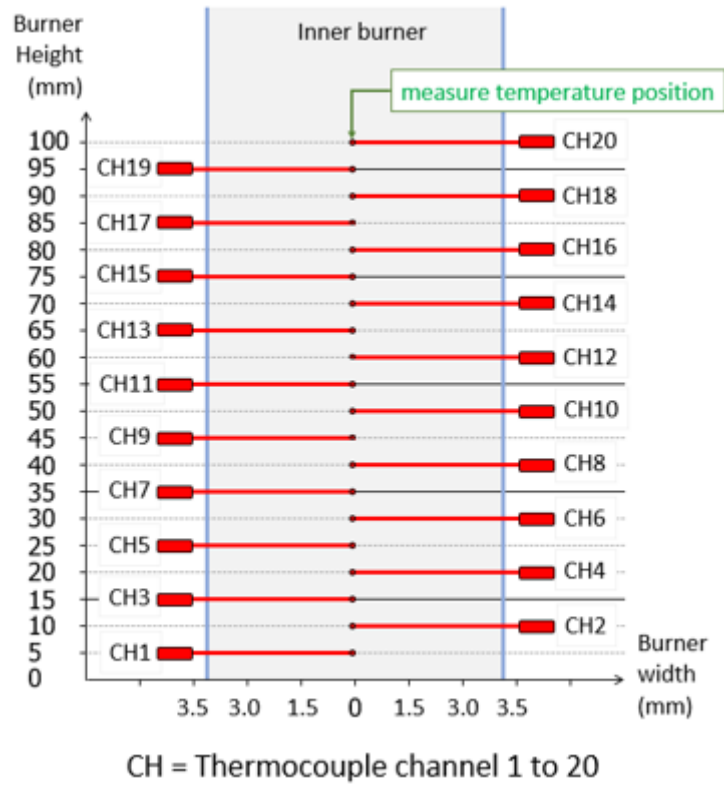


Figure 3-5 Schematic of the compact burner

Figure 3-4, Figure 3-5 Thermocouples type K were set up 1-20 channel in every 5 mm. Start measurement at 5 mm and end at 100mm from the bottom of burner. And, data logger “Graphtec GL840” was used to collect temperature data, as shown in Figure 3-6



Figure 3-6 Photo of data logger “Graphtec GL840”

3.6.3 The parameter of porous media ball diameter size

This experiment uses porous media ball material aluminium oxide(Al_2O_3) diameter size 3 size 3mm, 5mm, and 10mm, as shown in Figure 3-7

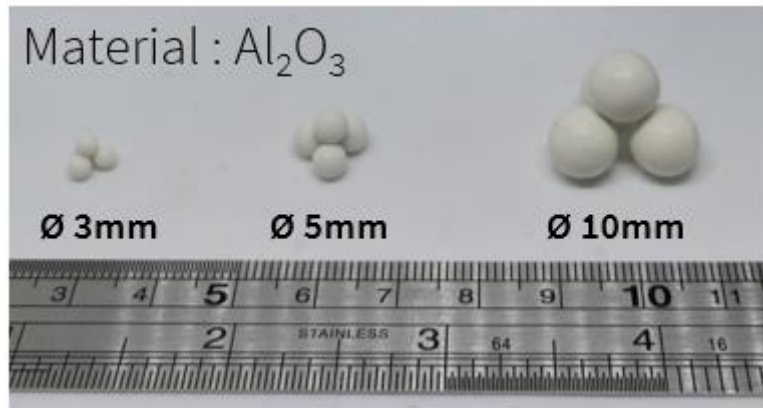


Figure 3-7 Alumina ball 3 sizes 3 mm, 5 mm and 10 mm

3.6.4 The patterns of inserted porous media ball in the burner

This experiment was collected temperatures when burning in each condition of LPG with air 15,000, 20,000, 25,000 cc/min in a compact combustion burner that consists of the ceramic ball. In this study, the three patterns of porous media, Pattern 1st is first layer input porous media diameter size 3 mm at 0-30 mm of burner length, next layer input porous media diameter size 10 mm at 30-200 mm of burner length, Pattern 2nd is only 1 size, input porous media diameter size 5 mm at 0-200 mm of burner length and pattern 3rd is only 1 size, input porous media diameter size 10 mm at 0-200 mm of burner length, as shown in Figure 3-8

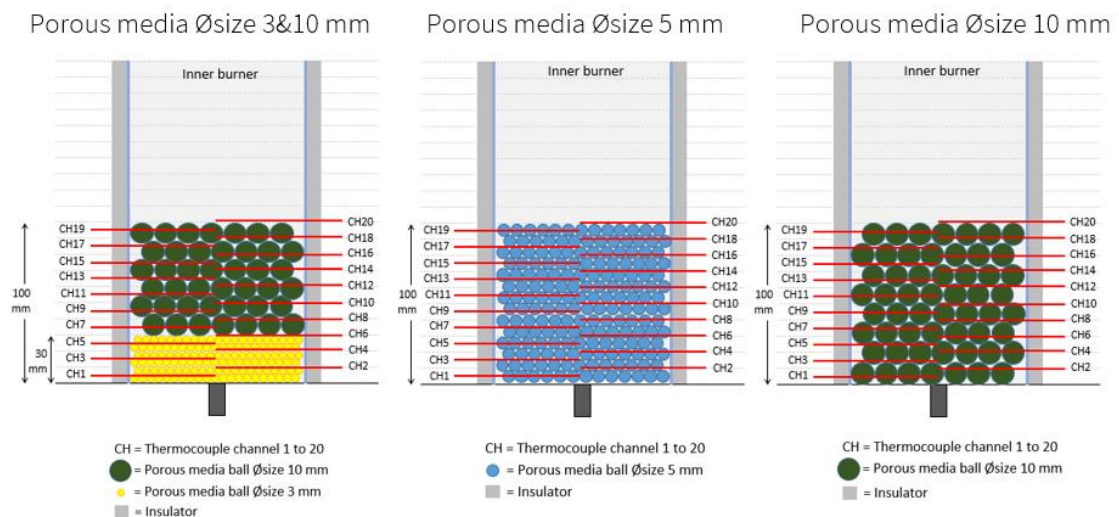


Figure 3-8 Summary of inserted porous media ball in the burner conditions

3.6.5 Parameter in experiment

In this research, the effects of various parameter changes are studied in the fuel flow rate is in the range 175-525 cc/min and the air flow rate is in the range 15,000-25,000 cc/min. Figure 3-9, show overall of mixture combustion between LPG and air when

burning in an experiment. A horizontal axis of the graph is the value of air flow rate, a vertical axis is the value of LPG flow rate. In the green box is a condition of porous media diameter size 3&10mm in the burner, in the blue box is a condition of porous media diameter size 5mm in the burner and in the orange box is a condition of porous media diameter size 10mm in burner. The meaning of green circle mark is the condition that can collect the temperature of combustion data in steady state, the 2 colors white and red cross mark is the condition that the fire when combustion was blackout, red cross mark is the condition that the temperature of combustion data is over 1200°C, as shown in Figure 3-9

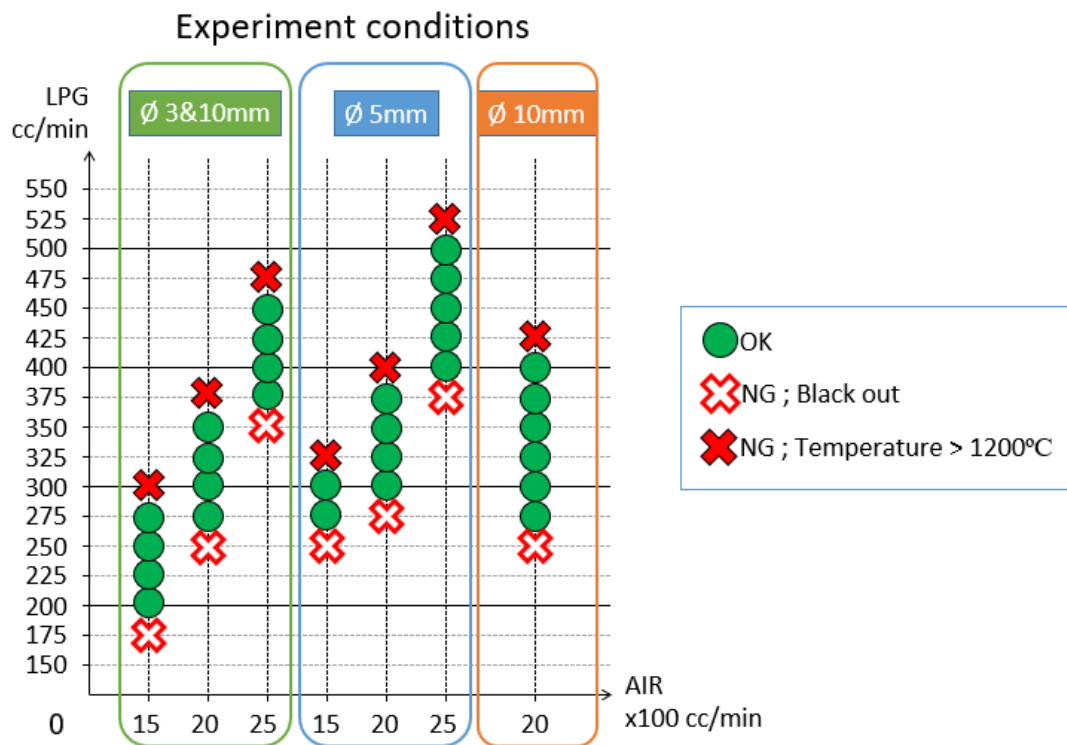


Figure 3-9 Summary of experiment combustion conditions

3.6.6 Procedure

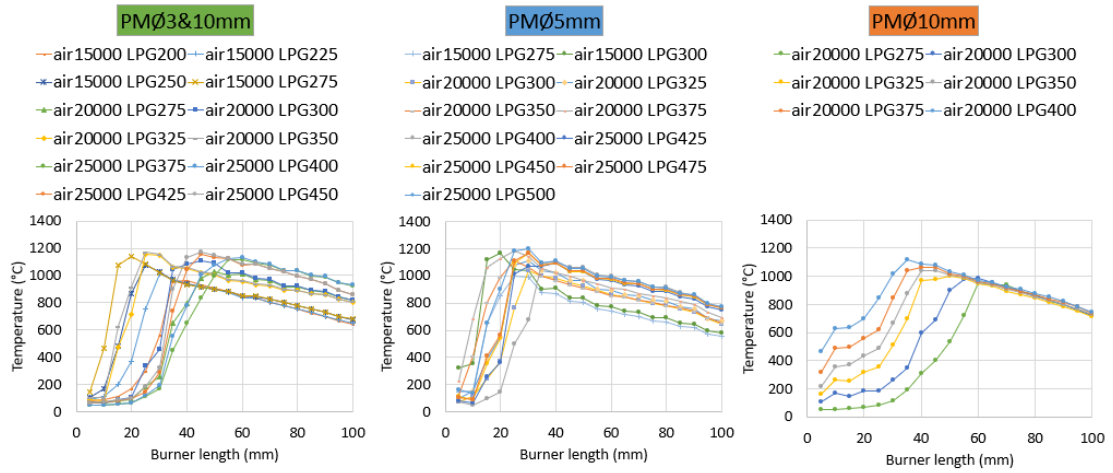
1. Input the air and fuel into the burner at a low flow rate and ignite and warm the burner until the combustion chamber has increased temperature.
2. Increasing air volume and fuel flow rate slowly. Adjust the experimental parameters, such as flow rate of air and fuel.
3. Let the burner continue until the temperature inside the burner enters a steady state. And, checking the above parameters throughout the experiment.

CHAPTER 4

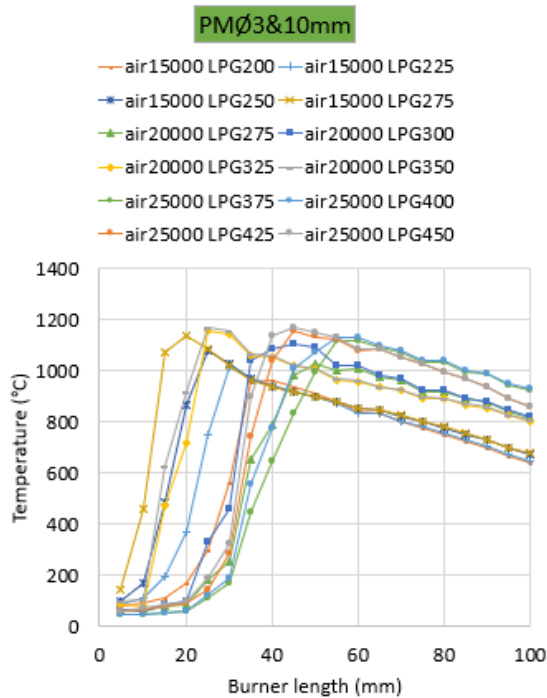
RESULTS AND DISCUSSIONS

4.1 Results

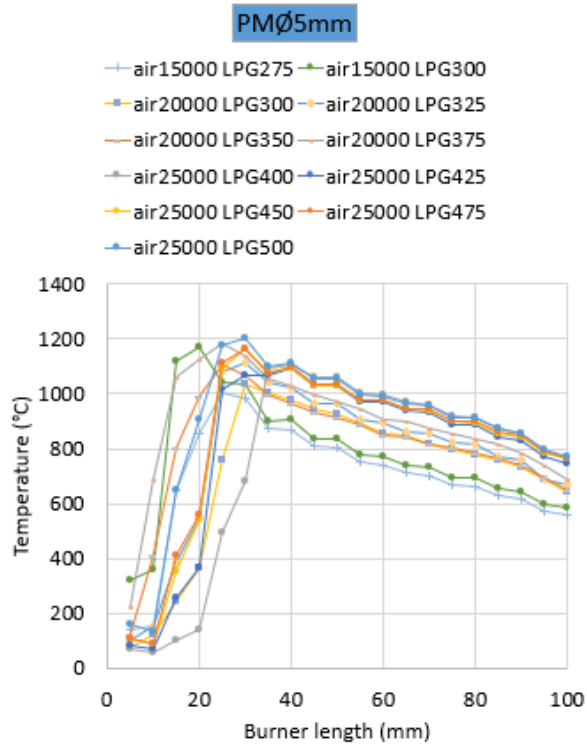
All combustion results were shown by the graph of burner length and the distribution temperature. The results were divided into three graphs by pattern of porous media in burner and lines in the graph are each combustion conditions, as shown in Figure 4-1



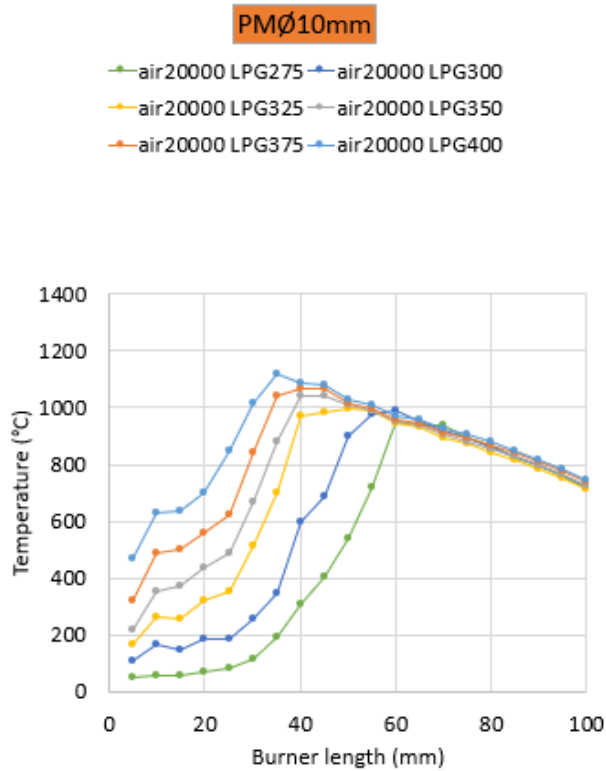
(a)



(b)



(c)



(d)

Figure 4-1 (a)Summary of all result in three patterns (b)Summary of all result pattern 1st porous media diameter size 3&10mm (c)Summary of all result pattern 2nd porous media diameter size 5mm (d)Summary of all result pattern 3rd porous media diameter size 10mm

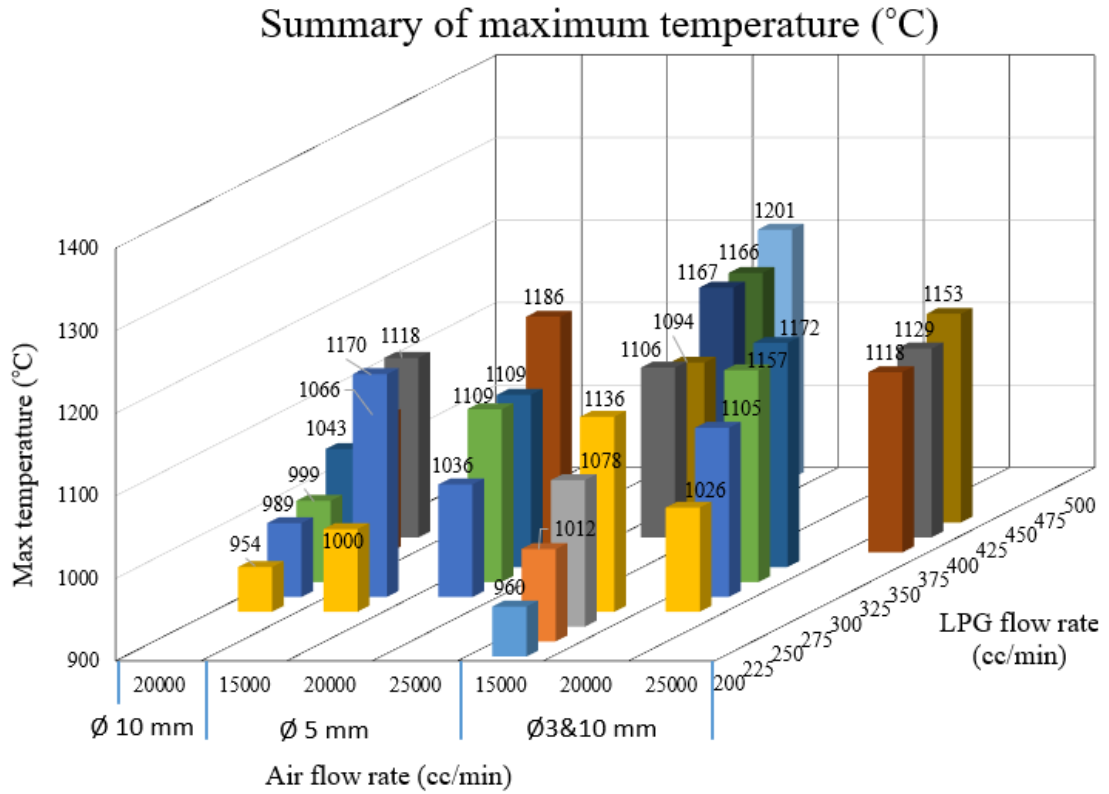


Figure 4-2 Summary of all result by maximum temperature and combustion condition

Figure 4-2 Summary of maximum temperature in all experiment combustion condition with x axis is air flow rate, y axis is LPG flow rate, and z axis is maximum temperature. The results from the experiment to studied impact of porous media size, arranging porous media ball size, fuel flow rate, air flow rate, and equivalence ratio

4.2 Impact of porous media ball size

This experiment was compared temperatures of combustion between porous media ball size 5 mm and size 10 mm in compact burner 0-200 mm. Both were wrapped with fiber paper insulating in 0-200 mm. The conditions of the experiment were using a constant flow rate of air at 20,000 cc/min and a different flow rate of LPG at 300, 325 and 350 cc/min, as shown in Figure 4-3

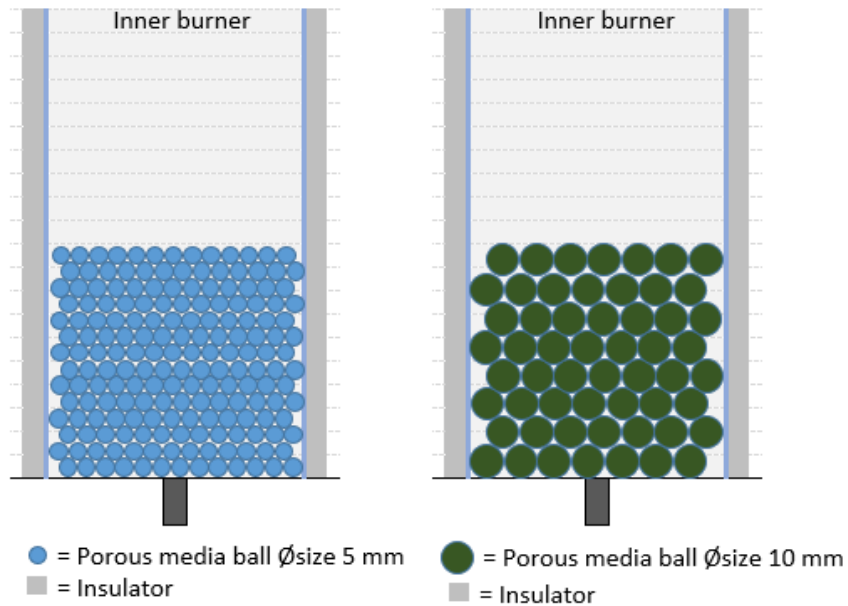


Figure 4-3 Schematic of the compact burner between porous media ball size 5 mm and size 10 mm

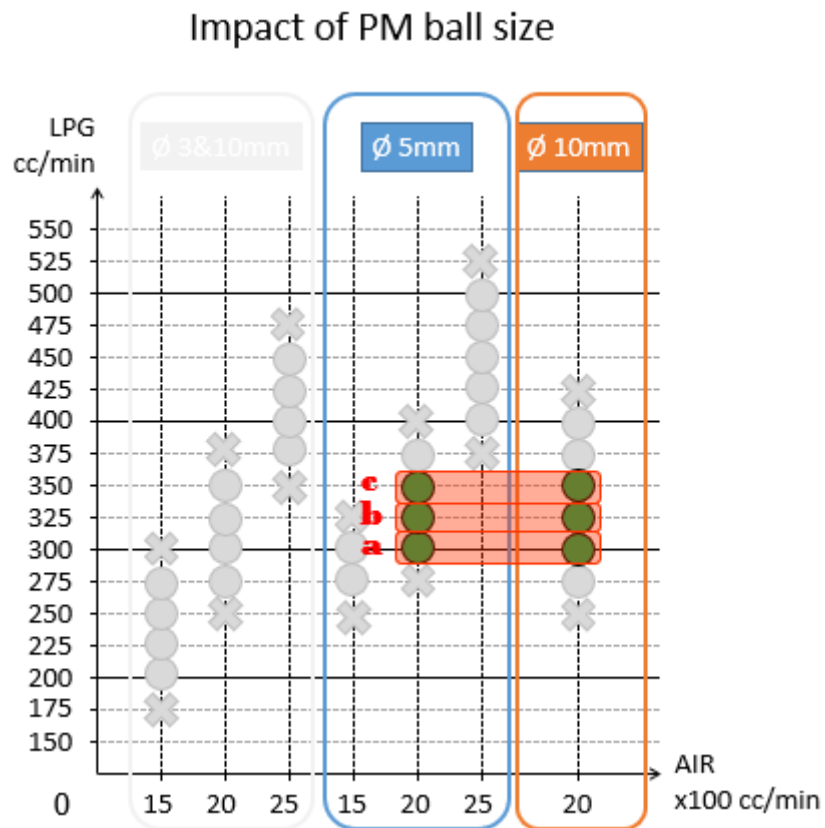
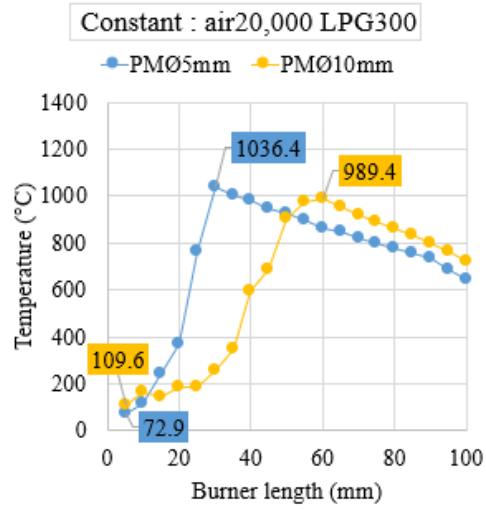


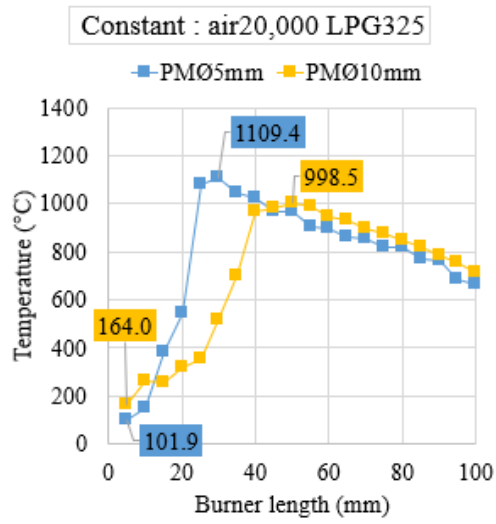
Figure 4-4 The comparison conditions graph of a porous media ball diameter size

Figure 4-4 This graph shows the condition of the comparison between porous media ball diameter size 5 mm and 10 mm in same condition combustion. The condition (a) is combustion in LPG 300 cc/min with air 20,000 cc/min condition. The condition (b)

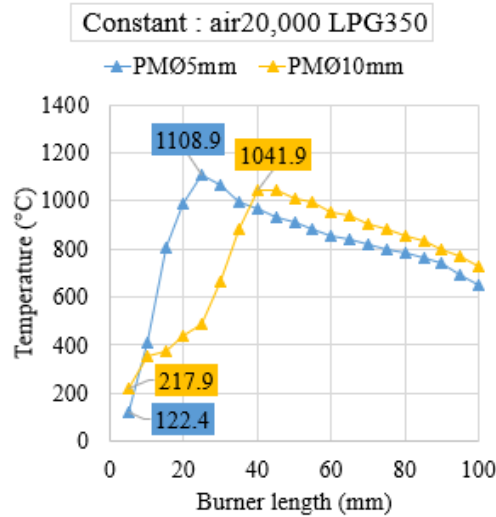
is combustion in LPG 325 cc/min with air 20,000 cc/min condition. The condition (c) is combustion in LPG 350 cc/min with air 20,000 cc/min condition



(a)



(b)



(c)

Figure 4-5 The comparison result graph of temperatures and burner length between porous media ball size 5 mm and porous media ball size 10 mm in compact burner 0-200 mm in (a) LPG 300 cc/min with air 20,000 cc/min condition (b) LPG 325 cc/min with air 20,000 cc/min condition (c) LPG 350 cc/min with air 20,000 cc/min condition

Figure 4-5, when the size of the porous media ball is bigger, the maximum temperature at the same condition was decreased, and the point where the maximum temperature occurs is farther from the inlet as well. And, the temperature at thermocouple channel 1st, the bigger size of the porous media ball has a higher temperature.

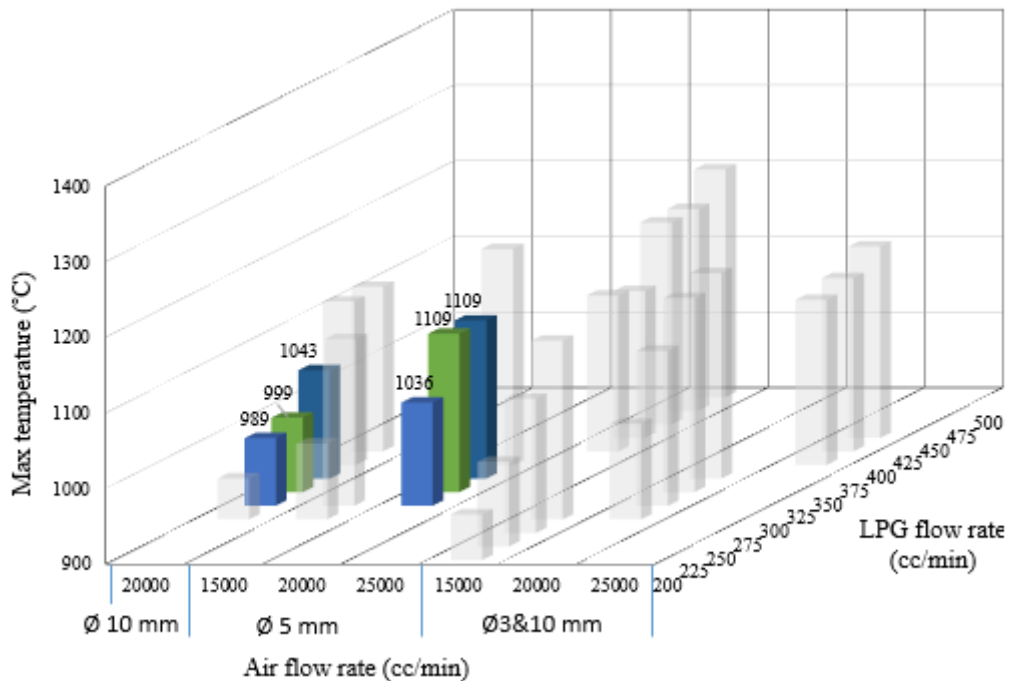


Figure 4-6 Summary of maximum temperature in impact of porous media ball size condition

Figure 4-6 from the graph, the bigger size of porous media ball has a higher maximum temperature in all conditions because when decreasing porous media diameter size, the area/volumes ratio is increased. The heat feedback from downstream was absorbed by porous media in upstream better. Also, porous media in upstream make high heat transfer from porous media to fuel in preheat combustion thus the pre-heating is high efficiency. The bigger size of porous media ball cannot absorb heat feedback not good as smaller size, so the temperature of thermocouple channel 1st is higher than smaller size.

4.3 Impact of arranging porous media ball size in the burner

This experiment was compared temperatures of combustion between porous media ball size 10 mm in compact burner 0-200 mm and size 3&10 mm in compact burner 0-30&30-200 mm. Both were wrapped with fiber paper insulating in 0-200 mm. The conditions of the experiment were using a constant flow rate of air at 20,000 cc/min and different flow rate of LPG at 275, 300, 325 and 350 cc/min, as shown in Figure 4-7

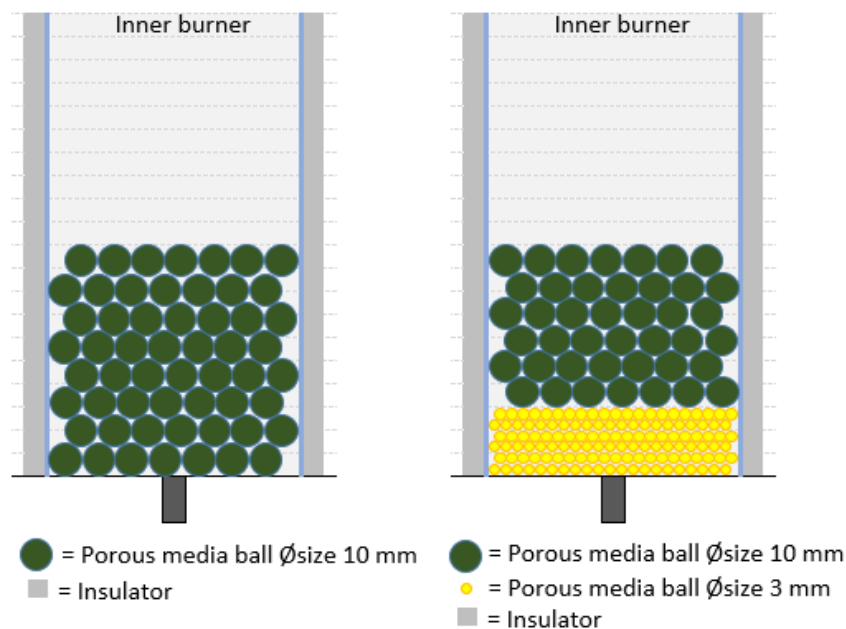


Figure 4-7 Schematic of the compact burner between porous media ball size 10 mm and size 3&10 mm

Impact of arrange PM ball size in burner

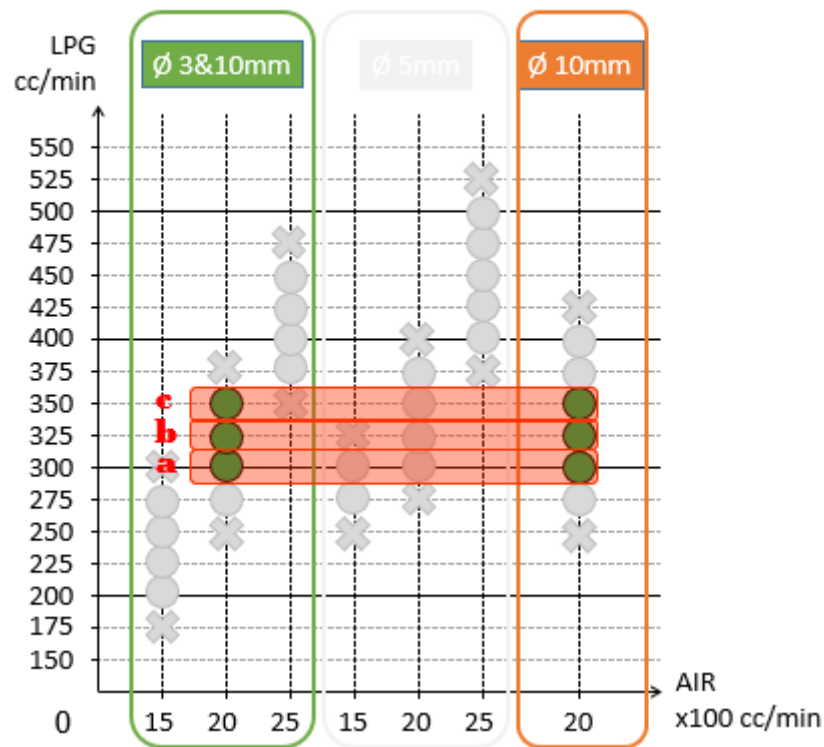
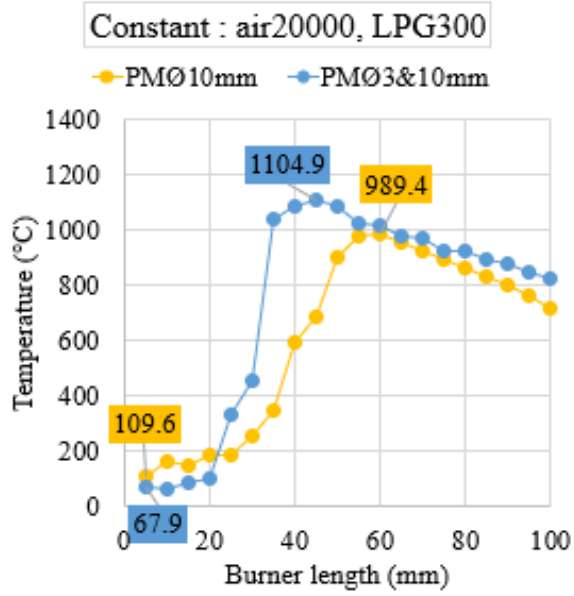
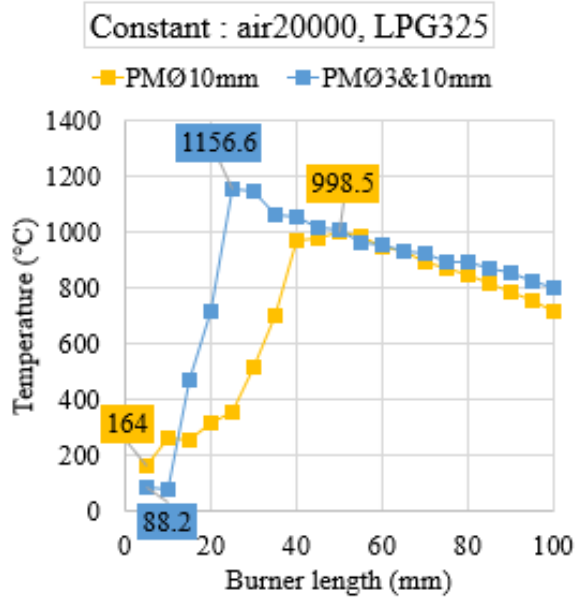


Figure 4-8 The comparison conditions graph of an arrange porous media ball size in burner

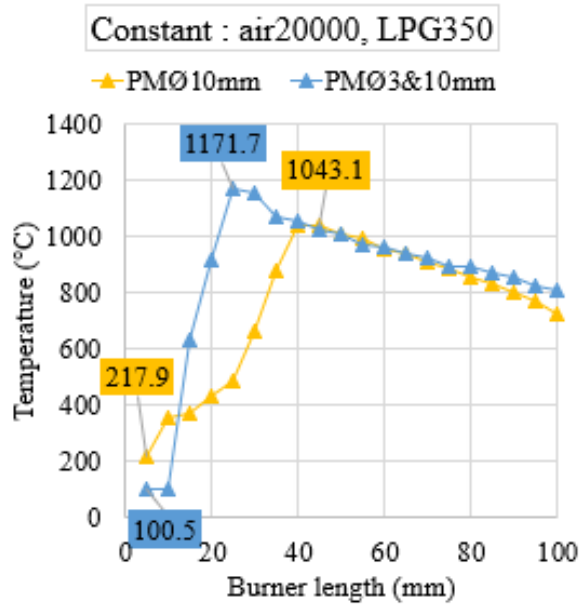
Figure 4-8 This graph show condition of the comparison between porous media ball diameter size 3&10 mm and 10 mm in same condition combustion. The condition (a) is combustion in LPG 300 cc/min with air 20,000 cc/min condition. The condition (b) is combustion in LPG 325 cc/min with air 20,000 cc/min condition. The condition (c) is combustion in LPG 350 cc/min with air 20,000 cc/min condition



(a)



(b)



(c)

Figure 4-9 The comparison graph of temperatures and burner length between porous media ball size 3&10 mm in compact burner 0-30&30-200 mm and porous media ball size 10 mm in compact burner 0-200 mm in a constant flow rate of air at 20,000 cc/min with (a) LPG 300 cc/min (b) LPG 325 cc/min (c) LPG 350 cc/min

Figure 4-9, when one size of porous media ball in the burner, the maximum temperature at the same condition with two sizes of porous media in the burner was decreased, and the point where the maximum temperature occurs is farther from the inlet as well. And, the temperature at thermocouple channel 1st, the one size of porous media ball in the burner has a higher temperature.

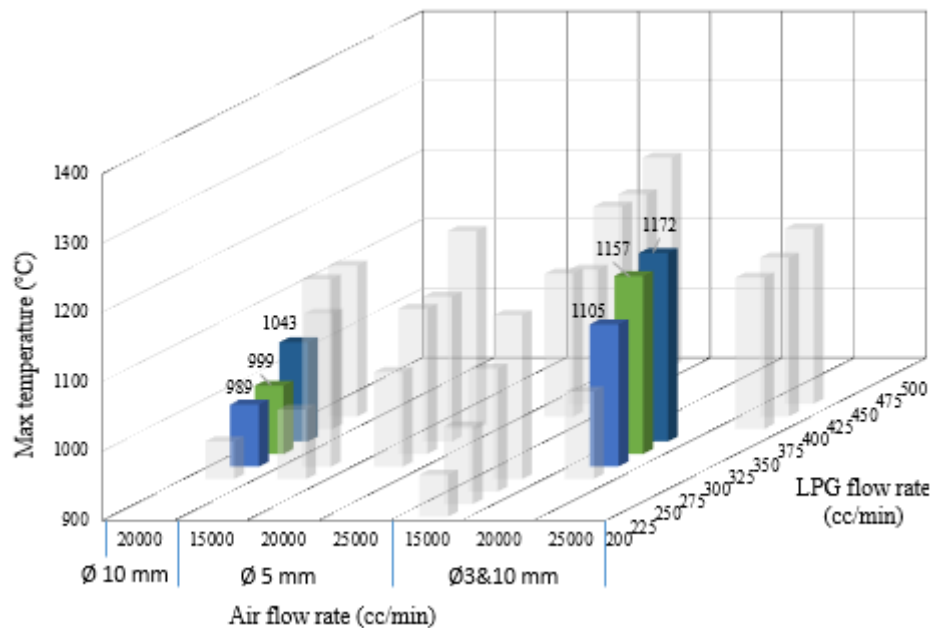


Figure 4-10 Summary of maximum temperature in the impact of arranging porous media ball size in the burner

Figure 4-10 from the graph, the two sizes of porous media in the burner has a higher maximum temperature in all conditions because when decreasing porous media diameter size in upstream, the area/volumes ratio is increased. The heat feedback from downstream was absorbed by porous media in upstream better. And porous media in upstream make high heat transfer from porous media to fuel in preheat combustion thus the pre-heating is high efficiency. the bigger size of porous media ball in upstream cannot absorb heat feedback not good as smaller size so the temperature of thermocouple channel 1st is higher than smaller size.

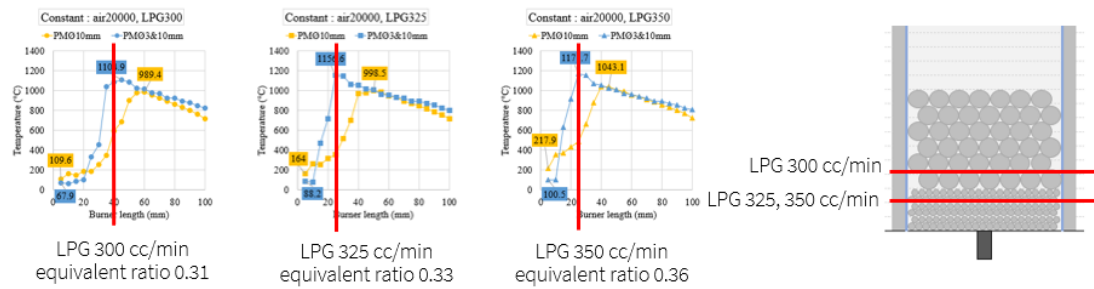


Figure 4-11 The comparison graph of temperatures and burner length between porous media ball size 10mm, 3&10 mm in a constant flow rate of air at 20,000 cc/min with (LH) LPG 300 cc/min (CTR) LPG 325 cc/min (RH) LPG 350 cc/min and Schematic of flame stabilization position

Figure 4-11 Flame stabilization position depends on the balance of flame speed and flow velocity. In LPG300 cc/min, the flow velocity is higher than flame speed, so flame stabilization position occurs on 40mm height from bottom in porous media size 10 mm zone. In LPG325, 350 cc/min flame speed is higher than flow velocity, but flame stabilization position cannot occur in porous media size 3 mm zone because absorbing porous media size 3 mm.

4.4 Impact of fuel flow rate

This experiment was compared temperatures of combustion in a different flow rate of LPG at 275-400 cc/min and a constant flow rate of air at 20,000 cc/min using porous media ball size 5 mm and size 10 mm in compact burner 0-200 mm. Both were wrapped with fiber paper insulating, as shown in Figure 4-12

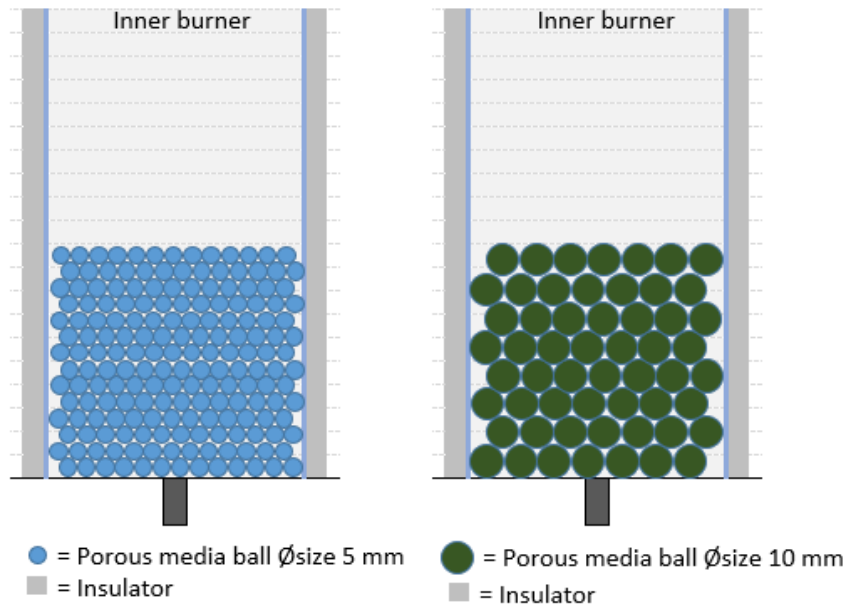


Figure 4-12 Schematic of the compact burner between porous media ball size 5 mm and size 10 mm

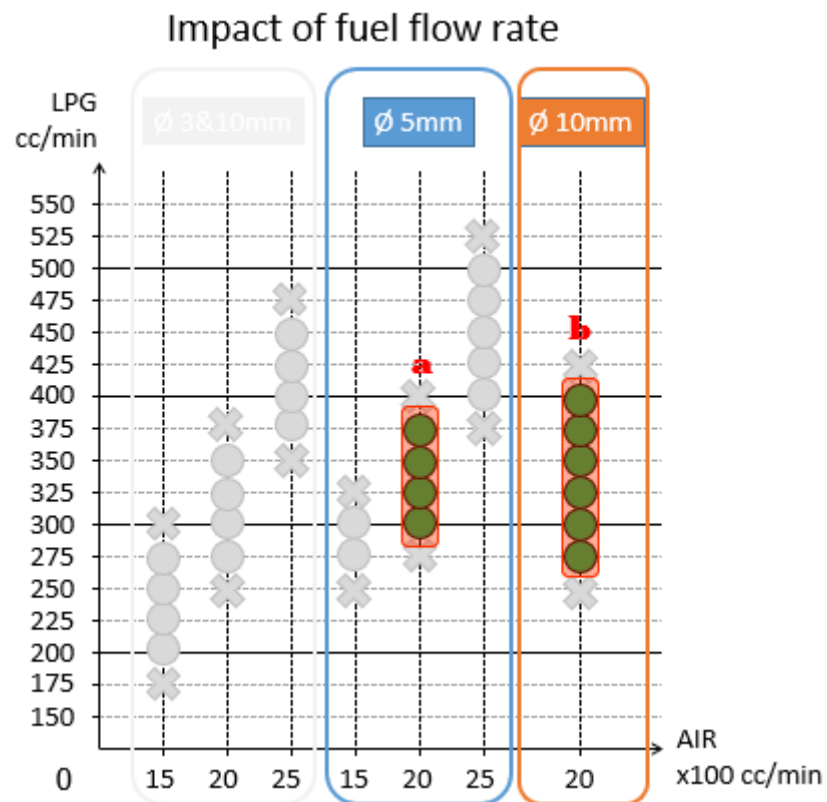
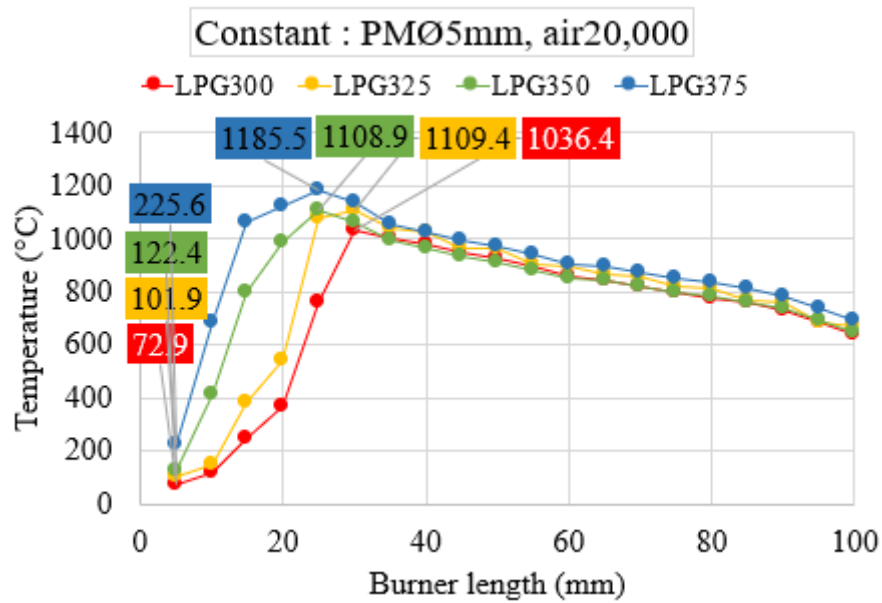


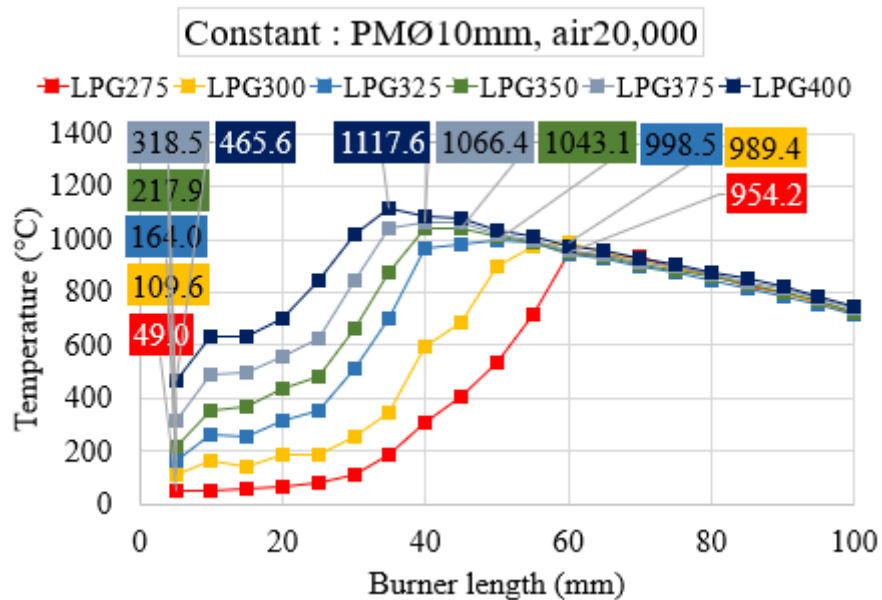
Figure 4-13 The comparison conditions graph of a fuel flow rate

Figure 4-13 This graph shows the condition of the comparison of different LPG flow rate with constant air flow rate condition using porous media ball diameter size 5 mm and 10 mm. The condition (a) is combustion in LPG 300-375 cc/min with air 20,000

cc/min condition. The condition (b) is combustion in LPG 275-400 cc/min with air 20,000 cc/min condition.



(a)



(b)

Figure 4-14 The comparison graph of temperatures and burner length between (a) LPG 300-375 cc/min with air 20,000 cc/min condition using porous media ball size 5mm in the burner 0-200mm (b) LPG 275-400 cc/min with air 20,000 cc/min condition using porous media ball size 10mm in the burner 0-200mm

Figure 4-14, when increase LPG flow rate but constant air flow rate, the maximum temperature is moving toward the inlet of the burner and the maximum temperature increases is higher as well.

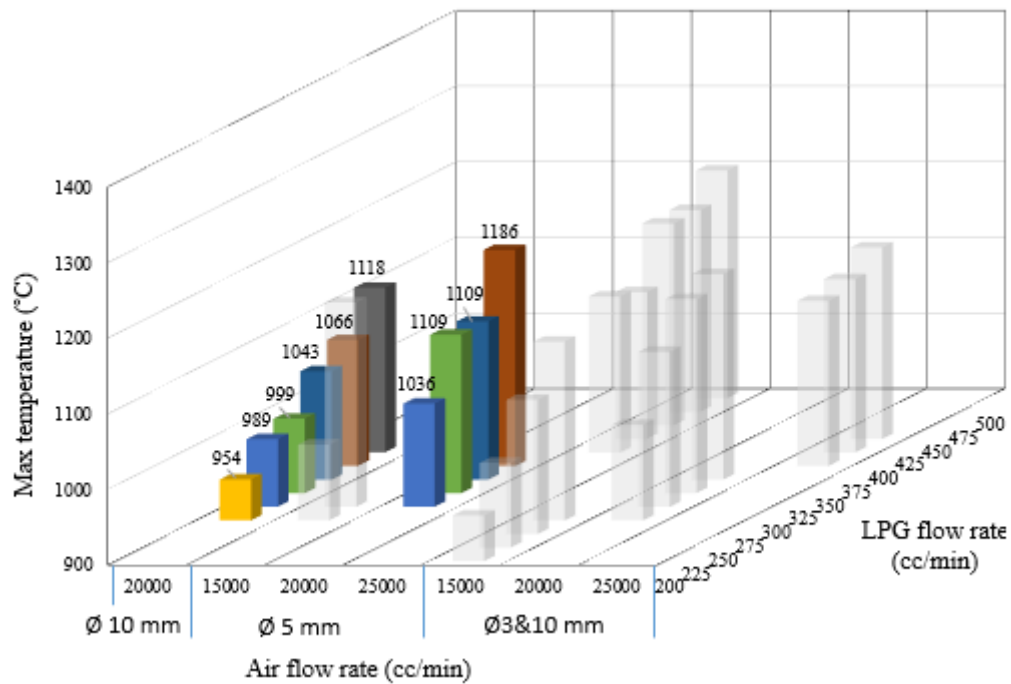


Figure 4-15 Summary of maximum temperature in the impact of fuel flow rate

Figure 4-15 from the graph, the maximum temperatures increase, when LPG flow rate was increased because it means increasing fuel mass in combustion then equivalence ratio and combustion ratio was increased.

4.5 Impact of air flow rate

This experiment was compared temperatures of combustion in different flow rate of air at 15,000 and 20,000 cc/min, constant flow rate of LPG 300 cc/min and 15,000 and 20,000 cc/min, constant flow rate of LPG 275 cc/min using porous media ball size 5 mm and size 3&10 mm in compact burner 0-30&30-200 mm. respectively. Both were wrapped with fiber paper insulating in 0-200 mm, as shown in Figure 4-16

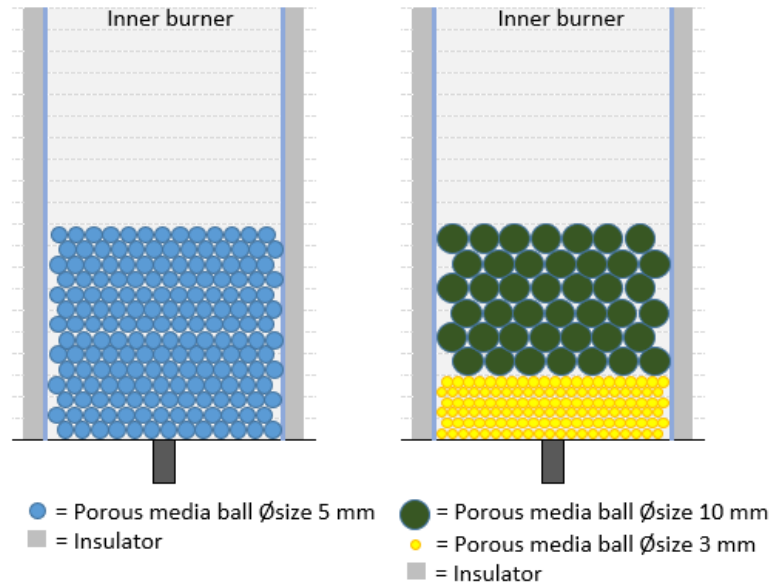


Figure 4-16 Schematic of the compact burner between porous media ball size 5 mm and size 3&10 mm

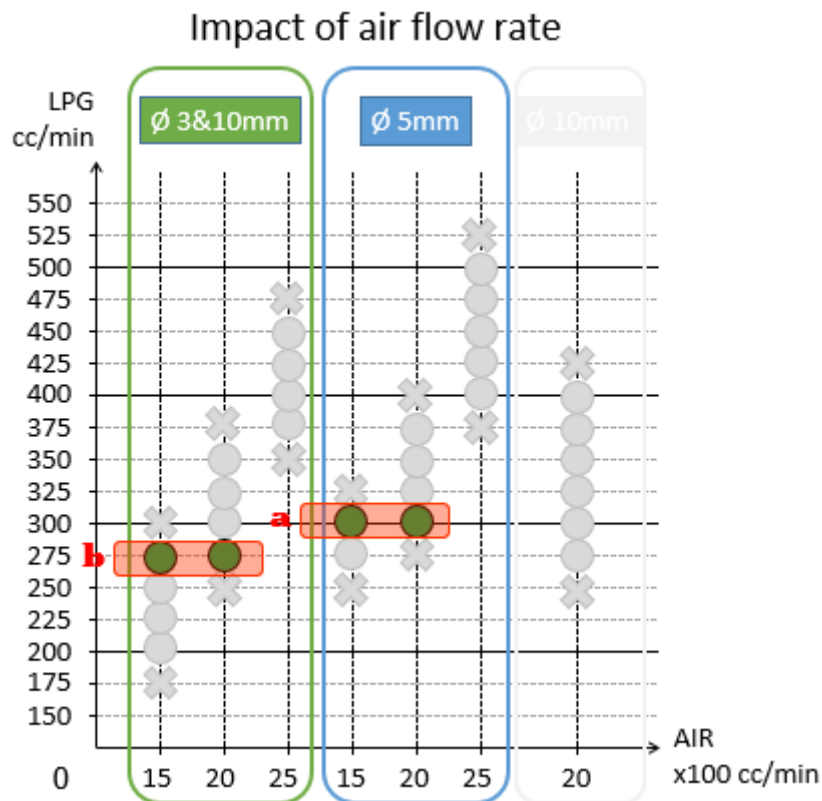


Figure 4-17 The comparison conditions graph of an air flow rate

Figure 4-17 This graph show condition of the comparison of different air flow rate with constant LPG flow rate condition using porous media ball diameter size 5 mm and 3&10 mm. The condition (a) is using porous media ball diameter size 5 mm combustion in LPG 300 cc/min with air 15,000 and 20,000 cc/min condition. The condition (b) is

using porous media ball diameter size 3&10 mm combustion in LPG 275 cc/min with air 15,000 and 20,000 cc/min condition.

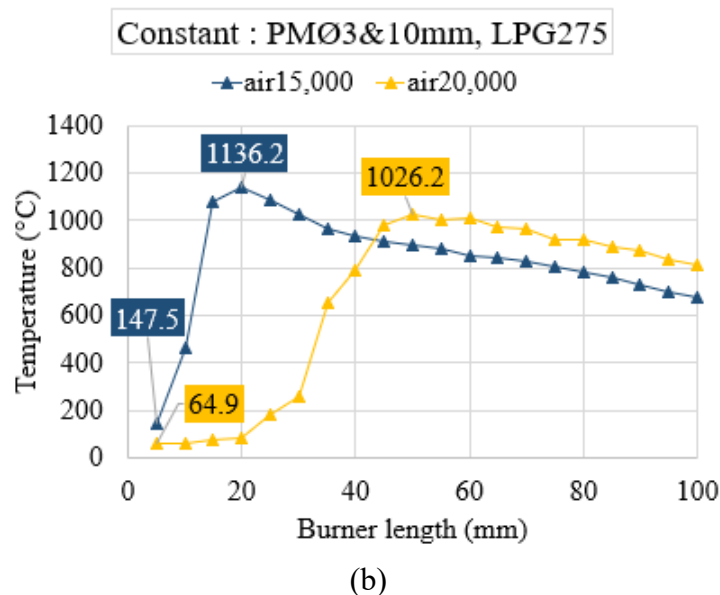
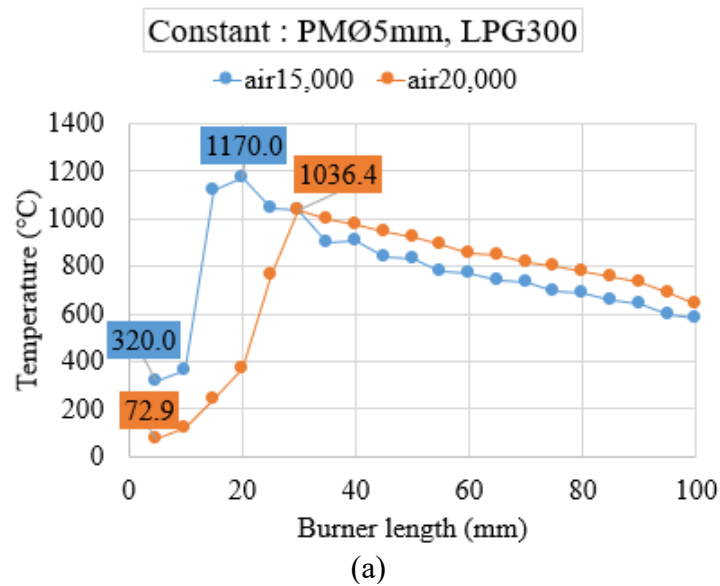


Figure 4-18 The comparison graph of temperatures and burner length between (a) LPG 300 cc/min with air 15,000 and 20,000 cc/min condition using porous media ball size 5mm in compact burner 0-200mm. (b) LPG 275 cc/min with air 15,000 and 20,000 cc/min condition using porous media ball size 3&10mm in compact burner 0-30&30-200mm.

Figure 4-18, when increase air flow rate but constant LPG flow rate, the maximum temperature is moving toward the outlet of the burner and the maximum temperature decreases is lower as well.

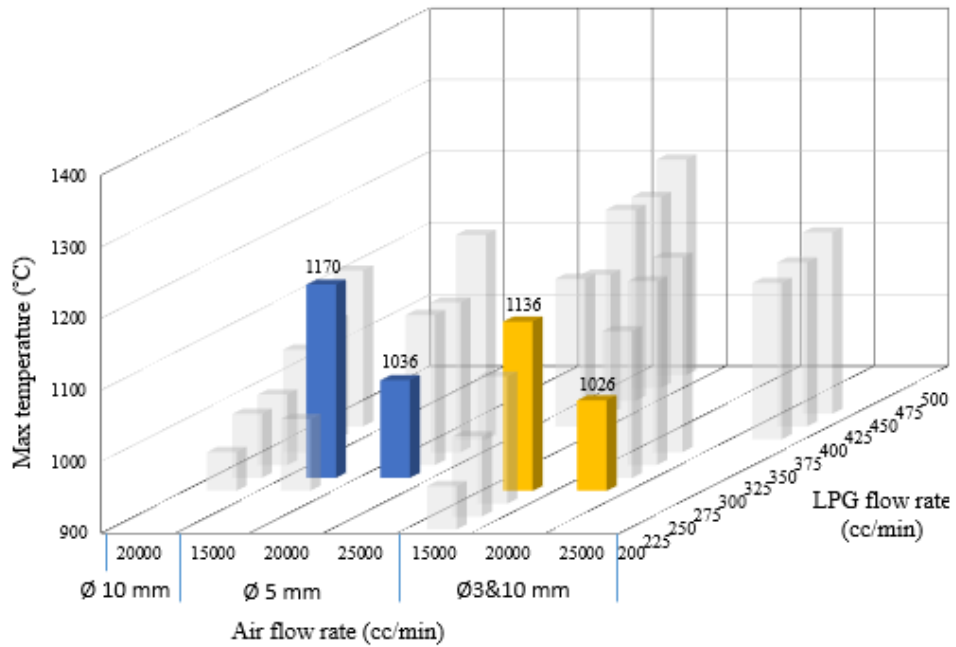


Figure 4-19 Summary of maximum temperature in the impact of air flow rate

Figure 4-19 from the graph, the maximum temperatures decrease, when air flow rate was increased because the equivalence ratio and combustion ratio was decreased.

4.6 Impact of equivalence ratio

This experiment was compared to the maximum temperatures of all combustion conditions but divided compare by pattern using porous media, as shown in Figure 4-20.

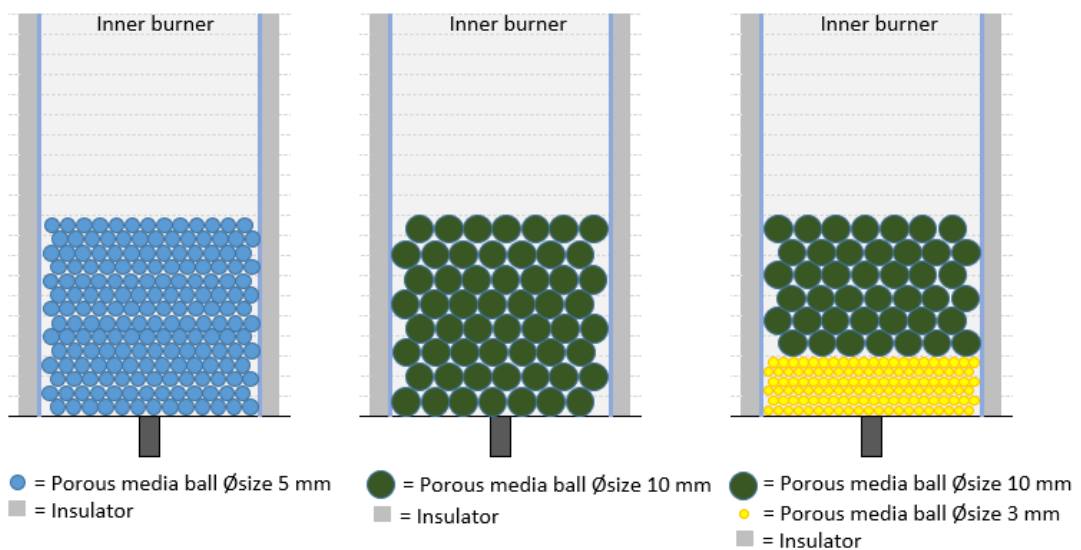


Figure 4-20 Schematic of the compact burner between porous media ball size 5 mm, size 10 mm and size 3&10 mm

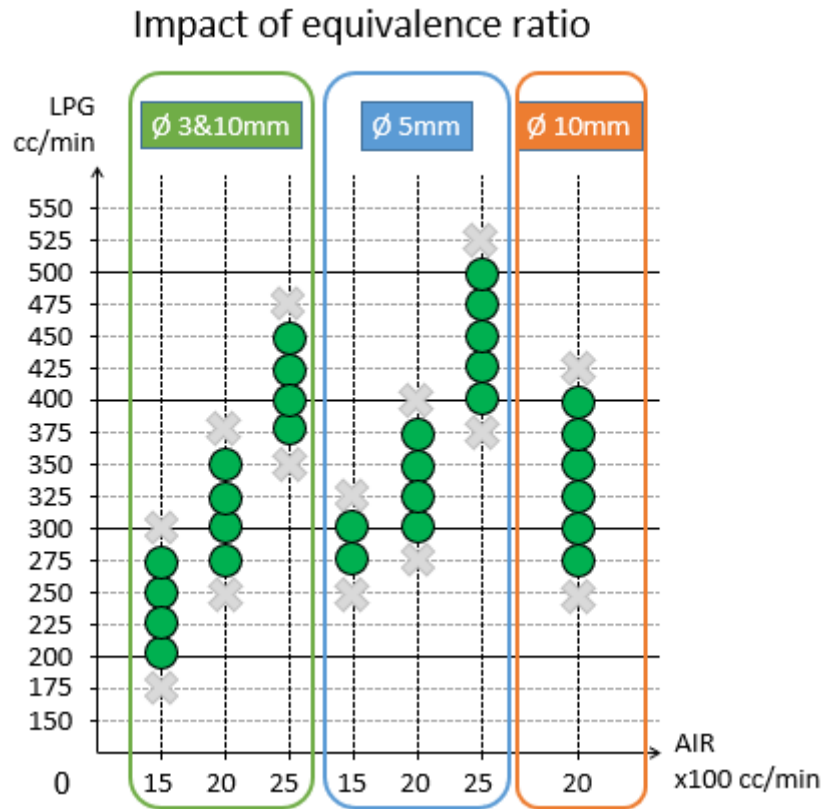


Figure 4-21 The comparison conditions graph of an equivalence ratio

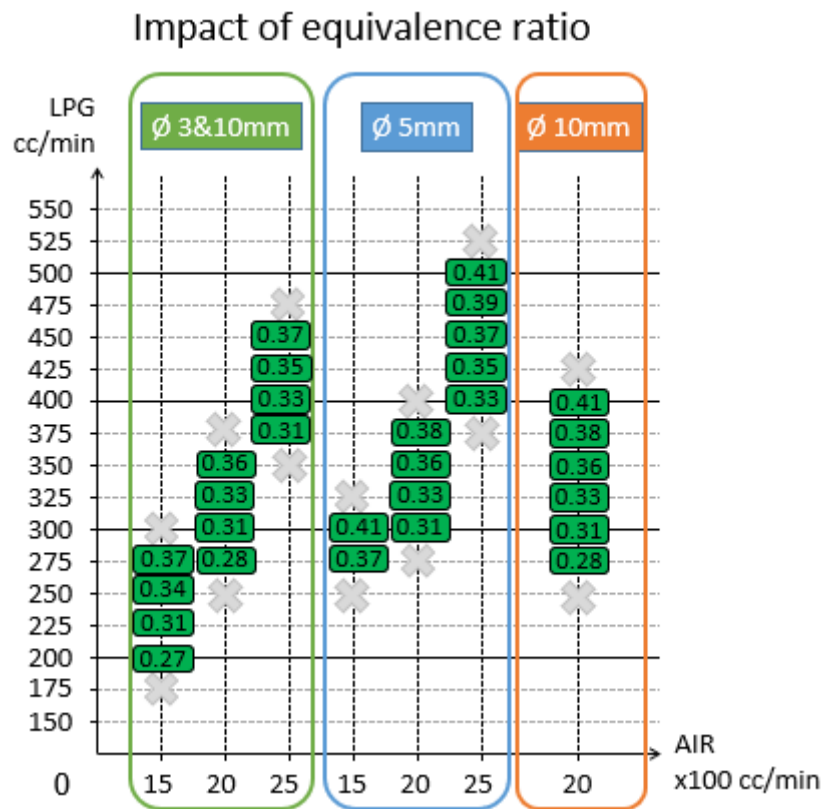
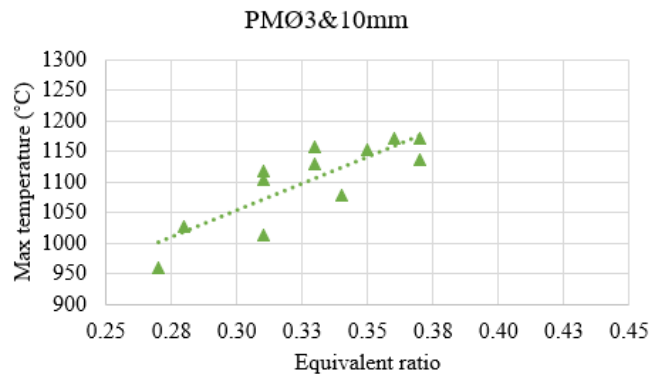
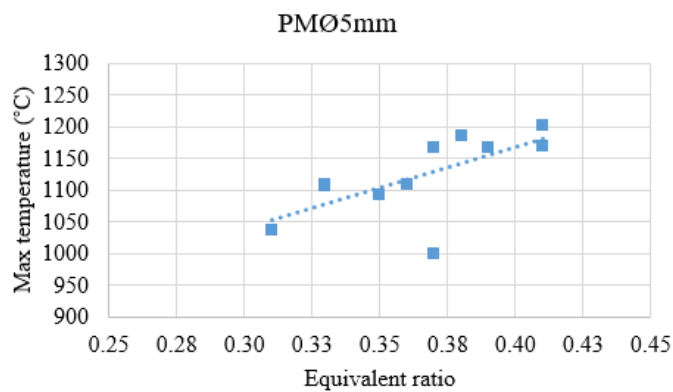


Figure 4-22 The graph of equivalence ratio value in each condition

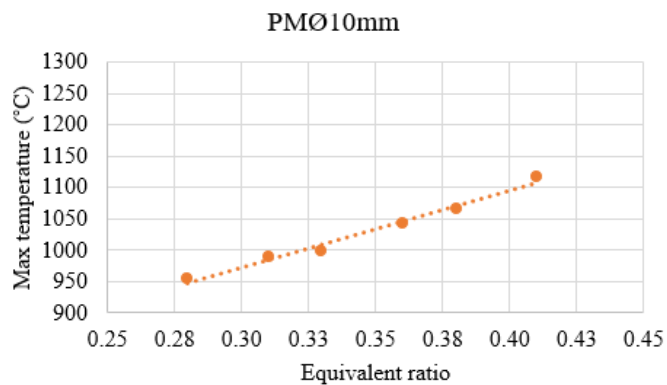
Figure 4-21, Figure 4-22 This graph show condition of the comparison equivalence ratio but divided compare by porous media ball diameter size 3&10 mm, 5 mm and 10 mm. The condition (a) is using porous media ball diameter size 3&10 mm equivalence ratio 0.27-0.37. The condition (b) is using porous media ball diameter size 5 mm equivalence ratio 0.31-0.41. The condition (c) is using porous media ball diameter size 10 mm equivalence ratio 0.28-0.41.



(a)



(b)



(c)

Figure 4-23 The comparison graph of maximum temperatures and equivalence ratio (a) porous media ball diameter size 3&10 mm (b) porous media ball diameter size 5 mm (c) porous media ball diameter size 10 mm

Error! Reference source not found., when equivalence ratio was increased, the maximum temperature increase.

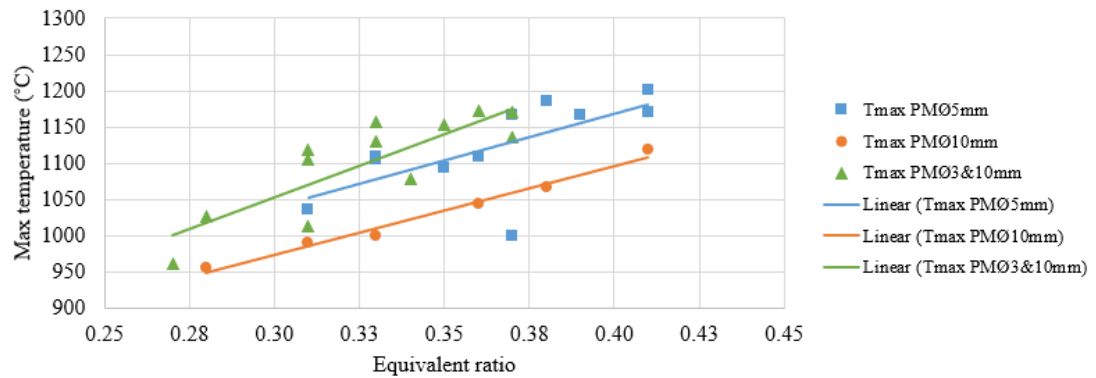


Figure 4-24 The summary of maximum temperature and equivalence ratio in all pattern porous media ball

Figure 4-24 from the graph, the maximum temperatures increase because equivalence ratio was increased. As the equivalence ratio approaches to the stoichiometric ratio 1.00, the amount of remaining oxygen will decrease, So the combustion will be complete combustion more. The maximum temperature is approaching an adiabatic flame temperature.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

Porous media combustion has a various factor in developing high-efficiency burner. The factors had been investigated: porous media size, porous media pattern, fuel, and air flow rate, and equivalence ratio. The factors that recommend using for increase efficacy of burner and be the careful point is

1. The decrease in porous media ball size makes the maximum temperature increase because the area/volumes ratio in the burner was increased. Beware point is timing to steady state take more time and hard to keep combustion point into the middle of the porous media zone.
2. The two porous media ball size in burner make the maximum temperature increase because the area/volumes ratio in the burner was increased.
3. The increase in LPG flow rate makes the maximum temperature increase because equivalence ratio was increased, but this research is not collected data about emission.
4. The decrease air flow rate makes the maximum temperature increase because equivalence ratio was decreased, but this research is not collected data about emission.
5. The increase equivalence ratio makes the maximum temperature increase.

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APPENDIX A

Calculation of porosity of porous materials

The porosity is defined as

$$\text{Porosity}(\phi) = \frac{V_{\text{cavity}}}{V_{\text{total}}} \quad (5-1)$$

Where, ϕ = the ratio of the void volume within a porous media to the total bulk volume of the porous media

V_{cavity} = a volume of cavity of porous media pack bed

V_{total} = a volume of cylindrical quartz tube

To calculate the porosity of porous materials, there is a calculation method as follows.

In case of alumina ball size 3 mm

1. Put as many alumina balls into the beaker but not above 200 mL line of the beaker.



Figure 1 Put alumina balls into the beaker

2. Add a water by syringe into beaker until the amount of flooded alumina ball and total volume after adding water equal to the 200 mL. Record the volume of water that is inserted.



Figure 2 Adding water equal to the 200 mL

Find the porosity is in the following the equation (5-1)

$$\text{Porosity}(\emptyset) = \frac{V_{cavity}}{V_{total}}$$

The volume of water that is inserted = 76 mL

Total volume after adding water = 200 mL

So,

$$\begin{aligned} \text{Porosity}(\emptyset) &= \frac{76}{200} \\ &= 0.38 \end{aligned}$$

In the same method, the porosity of 5 mm and 10 mm alumina balls can be calculated as 0.39 and 0.41, respectively.

Table Summary of alumina ball size and porosity.

Size	Porosity
3 mm	0.38
5 mm	0.39
10 mm	0.41

This method is cannot find the true value of porosity because the effect of the space between the beds and the beaker affects the volume of added water to find a volume of cavity

The first-time test combustion in compact burner by methane gas.



Figure 3 Methane tank

Before test combustion with porous media ball, porous media ceramic foam was used and combustion in various condition for find method to ignition.



Figure 4 The combustion appears on top of burner and use that flame to preheat ceramic foam

The experiment problem is the combustion isn't occurred in burner. It's always occurred at the end of burner even we input porous media in burner. So, we tried to preheat ceramic foam at room temperature to high temperature ceramic foam before input in burner because we think high temperature ceramic foam can absorb outside flame into inside burner. However, the preheat ceramic foam have a problem, too.

Table 1 The comparison table of temperatures and burner length between porous media ball size 5mm in compact burner 0-200 mm and porous media ball size 10 mm in compact burner 0-200 mm in (a) LPG 300 cc/min with air 20,000 cc/min condition (b) LPG 325 cc/min with air 20,000 cc/min condition (c) LPG 350 cc/min with air 20,000 cc/min condition

Porous media ball size →		(a)		(b)		(c)		
		5mm	10mm	5mm	10mm	5mm	10mm	
Position (mm)	Channel of thermocouple	LPG (x100 cc/min)	3.00	3.00	3.25	3.25	3.50	3.50
		AIR (l/min)	20	20	20	20	20	20
5	CH1	72.9	109.6	101.9	164.0	122.4	217.9	
10	CH2	117.6	165.6	151.0	262.0	410.7	353.3	
15	CH3	244.0	144.6	384.8	257.5	800.9	371.3	
20	CH4	368.2	186.6	544.2	319.0	990.0	435.5	
25	CH5	761.3	184.7	1081.1	354.8	1108.9	485.1	
30	CH6	1036.4	258.6	1109.4	514.2	1064.3	665.8	
35	CH7	1000.1	349.2	1041.2	701.7	994.0	879.2	
40	CH8	978.6	594.7	1024.7	969.5	963.9	1041.9	
45	CH9	947.6	686.5	963.9	981.3	934.3	1043.1	
50	CH10	926.0	900.3	963.9	998.5	910.8	1008.4	
55	CH11	894.5	975.7	904.9	987.8	884.4	991.8	
60	CH12	857.9	989.4	894.1	946.0	851.5	952.7	
65	CH13	846.8	953.7	863.6	929.8	842.9	938.8	
70	CH14	819.3	920.8	856.8	894.9	818.6	905.3	
75	CH15	799.2	891.5	820.3	871.9	800.8	883.1	
80	CH16	778.4	862.9	815.8	844.5	781.9	857.1	
85	CH17	757.8	830.0	770.4	816.2	762.5	829.1	
90	CH18	733.1	797.9	758.6	785.8	739.6	799.1	
95	CH19	685.2	763.4	684.4	754.9	692.1	767.7	
100	CH20	640.9	718.2	666.6	713.4	650.2	726.4	

→ Max temperature

Table 2 The comparison table of temperatures and burner length between porous media ball size 3&10 mm in compact burner 0-30&30-200 mm and porous media ball size 10 mm in compact burner 0-200 mm in (a) LPG 275 cc/min with air 20,000 cc/min condition (b) LPG 300 cc/min with air 20 L/min condition (c) LPG 325 cc/min with air 20 L/min condition (d) LPG 350 cc/min with air 20 L/min condition

		(a)		(b)		(c)		(d)		
Porous media ball size →		3&10mm	10mm	3&10mm	10mm	3&10mm	10mm	3&10mm	10mm	
Position (mm)	Channel of thermocouple	LPG (x100 cc/min)	2.75	2.75	3.00	3.00	3.25	3.25	3.50	3.50
		AIR (l/min)	20	20	20	20	20	20	20	20
5	CH1	Temperature (°C)	64.9	49.0	67.9	109.6	88.2	164.0	100.5	217.9
10	CH2		61.4	53.8	63.3	165.6	82.4	262.0	104.1	353.3
15	CH3		77.8	57.8	88.3	144.6	471.2	257.5	629.3	371.3
20	CH4		88.1	69.6	100.7	186.6	718.1	319.0	918.2	435.5
25	CH5		182.9	83.1	332.8	184.7	1156.6	354.8	1171.7	485.1
30	CH6		257.2	111.2	459.2	258.6	1144.3	514.2	1158.0	665.8
35	CH7		651.1	190.8	1040.3	349.2	1059.3	701.7	1066.7	879.2
40	CH8		791.2	309.6	1085.3	594.7	1053.2	969.5	1058.4	1041.9
45	CH9		981.8	403.5	1104.9	686.5	1018.2	981.3	1023.8	1043.1
50	CH10		1026.2	536.3	1088.6	900.3	1006.3	998.5	1010.5	1008.4
55	CH11		1001.4	718.2	1021.2	975.7	963.2	987.8	968.0	991.8
60	CH12		1007.4	954.2	1018.0	989.4	956.9	946.0	960.3	952.7
65	CH13		972.4	937.8	981.3	953.7	934.1	929.8	937.7	938.8
70	CH14		961.4	936.7	969.4	920.8	924.1	894.9	927.2	905.3
75	CH15		919.0	892.5	925.2	891.5	893.7	871.9	897.1	883.1
80	CH16		918.1	870.9	924.2	862.9	890.2	844.5	893.0	857.1
85	CH17		888.9	823.5	894.3	830.0	867.7	816.2	870.5	829.1
90	CH18		876.2	799.2	881.5	797.9	855.5	785.8	858.2	799.1
95	CH19		839.2	761.1	844.5	763.4	824.8	754.9	827.6	767.7
100	CH20		816.8	720.8	822.2	718.2	803.9	713.4	806.6	726.4

→ Max temperature

Table 3 The comparison table of temperatures and burner length between (a) LPG 300-375 cc/min with air 20,000 cc/min condition that consist of porous media ball size 5mm in compact burner 0-200 mm (b) LPG 275-400 cc/min with air 20,000 cc/min condition that consist of porous media ball size 10 mm in compact burner 0-200 mm

Porous media ball size →		(a)				(b)						
		5mm				10mm						
Position (mm)	Channel of thermocouple	LPG (x100 cc/min)	3.00	3.25	3.50	3.75	2.75	3.00	3.25	3.50	3.75	4.00
		AIR (l/min)	20	20	20	20	20	20	20	20	20	20
5	CH1	Temperature (°C)	72.9	101.9	122.4	225.6	49.0	109.6	164.0	217.9	318.5	465.6
10	CH2		117.6	151.0	410.7	686.3	53.8	165.6	262.0	353.3	488.4	630.9
15	CH3		244.0	384.8	800.9	1064.0	57.8	144.6	257.5	371.3	498.7	633.0
20	CH4		368.2	544.2	990.0	1122.9	69.6	186.6	319.0	435.5	560.1	700.7
25	CH5		761.3	1081.1	1108.9	1185.5	83.1	184.7	354.8	485.1	622.4	848.3
30	CH6		1036.4	1109.4	1064.3	1138.9	111.2	258.6	514.2	665.8	844.0	1017.1
35	CH7		1000.1	1041.2	994.0	1057.0	190.8	349.2	701.7	879.2	1041.3	1117.6
40	CH8		978.6	1024.7	963.9	1025.6	309.6	594.7	969.5	1041.9	1066.4	1085.9
45	CH9		947.6	963.9	934.3	994.8	403.5	686.5	981.3	1043.1	1064.6	1081.8
50	CH10		926.0	963.9	910.8	969.4	536.3	900.3	998.5	1008.4	1017.3	1030.7
55	CH11		894.5	904.9	884.4	942.4	718.2	975.7	967.8	991.8	997.5	1010.4
60	CH12		857.9	894.1	851.5	907.5	954.2	989.4	946.0	952.7	960.3	971.9
65	CH13		846.8	863.6	842.9	898.8	937.8	953.7	929.8	938.8	946.7	958.5
70	CH14		819.3	856.8	818.6	872.8	936.7	920.8	894.9	905.3	915.7	926.9
75	CH15		799.2	820.3	800.8	855.0	892.5	891.5	871.9	883.1	894.4	905.1
80	CH16		778.4	815.8	781.9	833.8	870.9	862.9	844.5	857.1	868.5	878.6
85	CH17		757.8	770.4	762.5	813.6	823.5	830.0	816.2	829.1	841.2	850.2
90	CH18		733.1	758.6	739.6	787.0	799.2	797.9	785.8	799.1	811.2	819.1
95	CH19		685.2	684.4	692.1	735.8	761.1	763.4	754.9	767.7	780.2	786.6
100	CH20		640.9	666.6	650.2	690.5	720.8	718.2	713.4	726.4	738.8	743.7

→ Max temperature

Table 4 The comparison table of temperatures and burner length between (a) LPG 300 cc/min with air 15,000,20,000 cc/min condition consist of porous media ball size 5 mm in compact burner 0-200 mm (b) LPG 275 cc/min with air 15,000,20,000 cc/min condition consist of porous media ball size 3&10 mm in compact burner 0-30&30-200 mm.

Porous media ball size →		(a)		(b)	
		5mm		3&10mm	
Position (mm)	Channel of thermocouple	LPG (x100 cc/min)			
		AIR (l/min)	3.00	3.00	2.75
		15	20	15	20
5	CH1	320.0	72.9	147.5	64.9
10	CH2	358.5	117.6	462.3	61.4
15	CH3	1119.8	244.0	1075.0	77.8
20	CH4	1170.0	368.2	1136.2	88.1
25	CH5	1044.2	761.3	1083.7	182.9
30	CH6	1038.2	1036.4	1023.0	257.2
35	CH7	901.8	1000.1	963.4	651.1
40	CH8	907.6	978.6	934.3	791.2
45	CH9	836.9	947.6	915.3	981.8
50	CH10	834.0	926.0	899.8	1026.2
55	CH11	778.5	894.5	878.9	1001.4
60	CH12	769.6	857.9	854.4	1007.4
65	CH13	739.9	846.8	847.2	972.4
70	CH14	731.2	819.3	826.2	961.4
75	CH15	695.6	799.2	803.4	919.0
80	CH16	691.2	778.4	782.1	918.1
85	CH17	656.1	757.8	757.0	888.9
90	CH18	642.9	733.1	733.5	876.2
95	CH19	598.1	685.2	701.4	839.2
100	CH20	581.5	640.9	679.4	816.8

→ Max temperature

Table 5 The comparison table of temperatures and burner length between (a) equivalence ratio 0.33 LPG 325 cc/min with air 20,000 cc/min and equivalence ratio LPG 400 cc/min with air 25,000 cc/min condition consist of porous media ball size 5 mm in compact burner 0-200 mm (b) equivalence ratio 0.41 LPG 300 cc/min with air 15,000 cc/min and equivalence ratio 0.41 LPG 500 cc/min with air 25,000 cc/min condition consist of porous media ball size 5 mm in compact burner 0-200 mm.

Porous media ball size →		(a)		(b)		
		5mm		5mm		
Position (mm)	Channel of thermocouple	LPG (x100 cc/min)				
		AIR (l/min)	20.00	25.00	15.00	25.00
		Equivalent ratio	0.33	0.33	0.41	0.41
5	CH1	Temperature (°C)	101.9	70.5	320.0	159.6
10	CH2		151.0	53.7	358.5	136.4
15	CH3		384.8	100.2	1119.8	649.1
20	CH4		544.2	143.1	1170.0	904.1
25	CH5		1081.1	497.3	1044.2	1180.0
30	CH6		1109.4	678.5	1038.2	1200.9
35	CH7		1041.2	1090.8	901.8	1097.8
40	CH8		1024.7	1105.9	907.6	1111.7
45	CH9		963.9	1063.0	836.9	1052.7
50	CH10		963.9	1060.6	834.0	1051.4
55	CH11		904.9	1006.2	778.5	997.1
60	CH12		894.1	997.9	769.6	993.0
65	CH13		863.6	968.0	739.9	962.7
70	CH14		856.8	959.2	731.2	955.9
75	CH15		820.3	918.8	695.6	915.0
80	CH16		815.8	913.6	691.2	912.9
85	CH17		770.4	873.1	656.1	872.2
90	CH18		758.6	855.9	642.9	858.1
95	CH19		684.4	790.3	598.1	796.7
100	CH20		666.6	765.5	581.5	772.7

→ Max temperature

APPENDIX B

PUBLICATIONS

The 4th International Conference on Engineering, Applied Sciences and The International Conference on Engineering, Applied Sciences and Technology (ICEAST), July 4-7, 2018

Numerical Simulation of Porous Media Combustion for High Temperature Heat Exchanger

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Abstract. The purpose of this work is developing the numerical 1D model of porous media combustion for investigating porous media burner systems. The software is used to solve energy, mass transfer and chemical reaction equation of the combustion. The operating condition and property parameters, which mainly affect the functions and quality of the industrial burner design, such as the inlet velocity of the reactants, the equivalence ratio, the extinction coefficient and the thermal conductivity of porous media, will be investigated and validated with experimental data. For developing the procedure of experiment, three diameter sizes of porous media materials (5 mm, 10 mm, and 15 mm.) were used. As a result, the developed model will be used as a tool to explore temperature distribution of heat exchange to improve thermal performance and overall efficiency system. Moreover, this knowledge can be applied to design porous media burner systems for uniform temperature distribution operation.

1 Introduction

In the present, environmental concerns and global warming problems have an impact on our environment. Although, the sustainable energy such as biomass, hydropower, geothermal, solar, wind and marine energies has been progressively used, it still cannot supply enough for the total world energy. And the nuclear energy has a problem about safety. Furthermore, the world considers reducing the consumption of coal and petroleum to reduce greenhouse gases. Under these conditions, improve the efficiency of in the present combustion has important.

Currently, the heat exchanger is widely used in commercial industry such as power stations, chemical plants and petrochemical plants etc. Operating of the heat exchanger in commercial industry is transfer heat from combustion to use. high temperature heat exchanger is an improved efficiency of the heat exchanger in high temperature operating condition but they have problems about meltdown of material, how to design heat transfer in heat exchanger etc.

Accordingly, porous media materials in the combustion of heat exchanger have attracted attention for improving heat exchanger efficiency in this work because of their advantages such as higher burning rates than free flame, extended lean flammability limits, low emissions of pollutants, high radiant output, and increased power dynamic range.

This paper presents the development of the combustion modeling in porous media. The developing

code is based on "One-dimensional CFD combustion modeling in porous media" written by Wasinaron[1]. The model is governed by conservation of mass, fuel species, energy in fluid pore space, energy in a solid porous matrix. According to the 2 phases of Heat transfer between solid and fluid phase, local thermal non-equilibrium was been modeling via energy source term of the solid and fluid energy equation. Arrhenius law is used to calculate fuel consumption and heat transfer from oxidation. The code is developed in one dimension due to prevent fluid dynamics complication at this stage. And, velocity field was merely calculated by ideal gas law as the calorific equation of state with 1D models. One dimensional code is recommended at the early stage of the complex porous media with combustion development.

2 Methodology

2.1 Porous media combustion modeling (PMC modeling)

The porous media combustion modeling trends up to 1994[2] are mainly described 1D models only. Takeno, Echigo, and their coworkers[3] are the pioneers in PMC modeling to study the effects of mass flow rate and heat transfer coefficient on flame characteristics in excess enthalpy flames and suggested inserting a porous, highly conductive solid into the flame to conduct heat from the solid to the reactants. They found that increasing the mass flow rate above the laminar burning rate increased the heat

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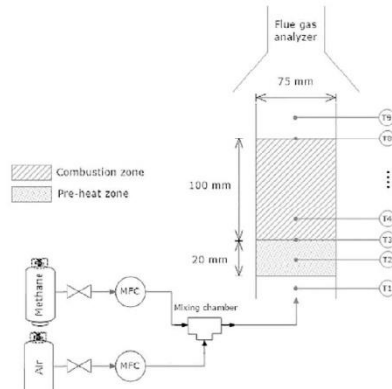


Fig. 1 Schematic of the experimental setup.

In the burner, a quartz tube was separated into two layers of porous media. The first layer was a preheat zone that was 20 mm. long and filled with Al_2O_3 pellets diameter size 3.0 mm. The second layer was the flame region that was 100 mm. long and filled with Al_2O_3 pellets diameter size 5.0 mm. in the 1st experiment, 10 mm. in the 2nd experiment, and 15 mm. in the 3rd experiment.

Thermocouples were set up as No.1 in the premixing zone and No.2 is in center of flashback protection zone. The No.3 is on the border between flashback protection zone and combustion zone. Thermocouples No.4-8 begin to start apart from No.3 in every 20mm until No.9 set depart of the combustion zone. The recorded data is temperature data and emission of the combustion exhausts such as CO, O₂, NO, C_xH_y and NO₂ at the burner outlet.

The air flow rate at inlet burner of methane/air mixture is kept constant between 0.09-0.15 m/s. And, equivalent ratio is developed between 0.5-1.0 that was suggested by Marker J. et al.[15].

3.2 Modeling equations

For simplify the problem, Model was followed under the below assumptions.

1. The porous media was setup as gray homogeneous media.
2. The wall is no slip, adiabatic, and radiative gray.
3. In high temperature, Potential catalytic effects of the solid are ignored.
4. Gas radiation and Body forces is neglected.
5. Reactants and Products setup as incompressible ideal gases.

A set of differential equations is in the following form.

Continuity equation,

$$\frac{\partial \epsilon \rho_g}{\partial t} + \nabla \cdot (\epsilon \rho_g u) = 0 \quad (1)$$

Where

ρ_g is the density of methane/air mixture

ϵ is the porosity, u is the velocity

Momentum equation,

$$\epsilon \rho_g \left(\frac{\partial u}{\partial t} + u \cdot \nabla u \right) = -\epsilon \nabla p + \mu \nabla^2 u + \frac{1}{3} \mu \nabla (\nabla \cdot u) \quad (2)$$

Where

μ is the viscosity of gas.

Species conservation equation,

$$\frac{\partial \epsilon \rho_g Y_i}{\partial t} + \nabla \cdot (\epsilon \rho_g Y_i) = -\nabla \cdot (\epsilon \rho_g Y_i V_i) + \epsilon \dot{c} \quad (3)$$

Where

Y_i is the molar fraction of substance

V_i is the diffuse velocity of substance

\dot{c} is the reaction rate of substance

W_i is the molar mass of substance

Energy equation,

For gas

$$\frac{\partial (\epsilon \rho_g C_p T_g)}{\partial t} + \nabla \cdot (\epsilon u (\rho_g C_p T_g + p)) = h_c (T_s - T_g) + \nabla \cdot (\epsilon k_g \nabla T_g) + \epsilon \dot{c} \quad (4)$$

For solid

$$\frac{\partial [(1-\epsilon) \rho_s C_s T_s]}{\partial t} = \nabla \cdot (k_s \nabla T_s) + h_c (T_s - T_g) \quad (5)$$

Where

\dot{c} is the heat from the chemical reaction

h_c is the convection of heat transfer coefficient

k_s is the heat transfer coefficient for solids

The inlet velocity is defined by

$$u = \frac{\dot{I}}{A} = \frac{\dot{I}}{\pi D^2} = \frac{\dot{I}_m}{\pi D^2} \quad (6)$$

Where

\dot{I} is the gas flow rate of methane/air mixture

A is the sectional area of the flow path

D is the diameter of the flow path

\dot{I}_m is the gas flow rates of methane

\dot{I}_a is the gas flow rates of air

The porosity is defined as

$$\text{Porosity}(\epsilon) = \frac{V_{\text{cavity}}}{V_{\text{total}}} \times 100 \quad (7)$$

Where

ϵ is the ratio of the void volume within a porous media to the total bulk volume of the porous media

V_{cavity} is a volume of cavity of porous media pack

bed

V_{total} is a volume of cylindrical quartz tube

release rate and the reaction zone became more concentrated.

Generally, all 1D models assume 1D flow conditions and no radial heat losses[4]. These two assumptions may become inaccurate if the porous media combustor is a commercial burner prototype with complex geometry. In such a situation 1D relations will no longer be valid and multidimensional models are imperative. It is, therefore, necessary to predict 3D combustion and emissions in complex geometrical burner configurations to help the design of commercial inert porous burners[5].

Mohamad et al.[6,7] modeled a PM burner with embedded coolant tubes. The 2D continuity, momentum, energy and fuel mass fraction equations were solved and the combustion was described as a one-step reaction. A numerical code has been developed by Bidi et al.[8] to evaluate the effects of different parameters of combustion in porous media. The Navier-Stokes, the solid and gas energy and the chemical species transport equations were solved using a multi-step reduced kinetic mechanism. The discrete ordinates method was used to solve the radiative transfer equation and a finite volume method (FVM) based on SIMPLE method was applied to discretize the conservation equations. Moraga et al.[9] who studied the convective heat transfer within a cylindrical inert porous media combustor have used a 2D, two temperature mathematical model, based on fluid mechanics, energy and chemical species governing equations. The FVM was used to solve the discrete model for methane combustion with air.

Hayashi and coworkers[10] had introduced the 3D modeling of a two-layer burner. The first layer was a perforated plate made of an insulating material (Al_2O_3) with the purpose of avoiding flash-back, while the second layer, a thin plate made of SiC foam to act as the reaction layer. They claimed that the proposed 3D model could facilitate a detailed study of the flow at the interface of the two solid layers, which is not possible utilizing one- or two-dimensional models, owing to the complex flow structure originated by the 3D jets from the perforated plate into the SiC foam.

For most of the cases, a 1D model with single-step reaction kinetics could yield the results with reasonable accuracy. In fact, the realistic prediction of pollutants formation necessitated detailed reaction kinetics to be incorporated in the model. And in 3D simulation, the inclusion of detailed reaction chemistry is still lacking, or 3D simulation is too small[11].

2.2 Porous media burner

Porous media combustion has three modes of heat transfer conduction, convection, and radiation. Advantages of Porous media burner[12] is a better homogenization of temperature across the porous media and the significant amount of radiation helps to preheat the incoming air-fuel mixture at upstream. The technique of premixed combustion within porous media has been studied and applied to steady combustion with great success. The porous media combustion has proved to be

one of the applicable options to solve the problems to a remarkable extent in both technical and economic perspectives. This technique has[12] been used for both gaseous and liquid fuels in steady or unsteady combustion. Flame stability in porous media with lean and rich mixtures, a significant reduction in pollutants and increasing combustion efficiency, was proven.

The material of porous media burner[13], aluminum oxide (Al_2O_3), silicon carbide (SiC), and zirconium dioxide (ZrO_2) proposed as suitable materials for application. Aluminum oxide and zirconium dioxide were recognized as high temperature resistant materials. SiC shows good thermal shock resistance, mechanical strength, and conductive heat transport. Silicon carbide also has a high melting point (3260 K), against cyclic thermal stress and strength retention at the peak regenerator temperature (1673 K), and excellent oxidation resistance. Metallic materials were found less suitable for porous media because of their inadequate thermal stability and high thermal inertia.

A comparison of the data for materials relevant for use in porous media burners is shown in Table 1.

Table 1 Most important material data for Al_2O_3 , SiC and ZrO_2 [14]

Property	Unit	Al_2O_3	SiC	ZrO_2
Maximum use temperature in air	$^{\circ}C$	1900	1650	1800
Thermal expansion coefficient (20-1000 $^{\circ}C$)	$10^{-6} K^{-1}$	8	4-5	10-13
Thermal conductivity at 20 $^{\circ}C$	$\frac{W}{m \cdot K}$	20-30	80-150	2-5
Thermal conductivity at 1000 $^{\circ}C$	$\frac{W}{m \cdot K}$	5-6	20-50	2-4
Specific thermal capacity	$J g^{-1} K^{-1}$	0.9-1	0.7-0.8	0.5-0.6
Thermal stress resistance parameter, hard shock, $R (\sigma/E\alpha)$	K	100	230	230
Thermal stress resistance parameter, mild thermal shock, $R^* (R\lambda)$	$\frac{J}{m^2 \cdot K}$	3	23	1

3 Experiment section

3.1 Experiment setup

Fig. 1 shows the schematic of the experimental setup. This system is composed of methane gas and air supply systems, a mixing chamber, four sizes of porous media, and a measurement system. Methane gas and air were controlled by MFC and then mixed in the mixing chamber. The methane/air mixture flowed through flashback protection and after that combustion in a quartz tube 75 mm. in diameter and 170 mm. long.

4 Results

In this study, the combustion position occurs at the rapidly changed temperature of the fluid. And the location of the combustion is depended upon the porosity. The porosity. At the porosity of 0.30 or 5 mm. of diameter size of porous media has the combustion position at approximately 1.9 cm., and porosity of 0.35 (diameter size 10mm) has the combustion position at 2.2 cm., and porosity of 0.40 (diameter size 15mm.) has the combustion position at 2.5 cm., respectively

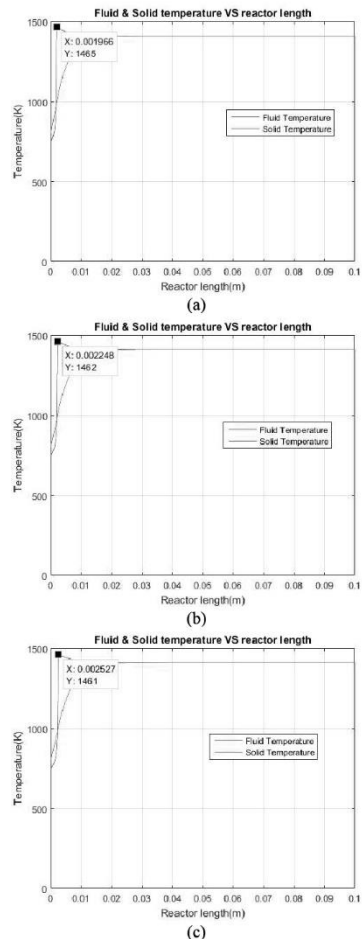


Fig. 2 Temperature distribution along the axial distance of modeling at porosity (a) 0.3, (b) 0.35, and (c) 0.4

5 Conclusions

The comparison three diameter size of porous media model results, the porosity of porous media affects the position of combustion that occurs in the model. The lower value of porosity shows the faster combustion will occurred.

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