Design Optimization of Air Staged Combustor Geometry for Low-Grade Biomass Producer Gas Combustion

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Abstract- Since industrial revolution, the supply of the needed energy depended on fossil fuel burning. Nowadays, renewable energies are getting attention for replacing fossil fuels to mitigate the raising crisis of global warming caused by greenhouse emissions. Biomass energy is one of the carbon-neutral renewable energies. However, low heating value of producer gas from biomass gasification requires longer residence time to burn. Long residence time of fuel burning brings increment of the NOx formation. Thus, staged combustion is applied to control the NO_x emissions of the combustor. In this study, combustion characteristics of air staged combustor were studied. Effects of geometry manipulative variables, such as combustor diameter (D), diameter ratio (DR) of air distributor to total combustor diameters and length ratio (LR) of air distributor to total combustor lengths were investigated. The results showed combustor with smaller diameter and larger DR released less NO_x gases, but it resulted in higher CO emissions and lower mean outlet temperature. With the increment of LR, CO emissions raised, and mean outlet temperature decreased but NO_x emissions did not change much with the LR variation. Therefore, optimum chamber geometry was found to be 250 mm diameter, DR of 0.6 and LR of 0.2 to maximize the temperature at the chamber outlet while achieving acceptable CO and NO_x emissions from the staged combustor.

Keywords Air staged chamber, Combustion, Gasification, Biomass, Producer gas.

1. Introduction

Energy utilization for the whole world is increasing drastically since industrial revolution. Most of the energy usages are generated through fossil fuels burning. According to the study of Energy Information Administration (EIA), percentage of fossil fuel consumptions in United States reached peak of 94% in 1966 and still maintained about 80% in 2019. Oil, natural gas and coal are the main energy sources in Indonesia. In 2015, the renewable energy consumption of Indonesia is still less than 9.5% [1]. Large amounts of fossil fuel burning bring air pollution, climate change and global warming. Climate change is a serious problem, where number of deaths increase when temperature surpasses a certain limit [2]. Global warming causes extreme weather, increase of sea level, ice melt and damages to the animals. Water temperature increases under the influence of global warming, this will affect the growing of seaweed bed and in turn damage the habitat of marine herbivorous [3]. Recently, world governments were highly concerned about the environmental problems. Paris agreement was signed globally by many countries in 2016. By complying with Paris agreement, all countries aim to reach carbon neutral status as soon as possible. To achieve the carbon neutral target, many countries actively promote the implementation of renewable energies. South Africa government executes Renewable Energy Independent Power Producer Procurement Programme (REIPPPP) to attract private sectors in utility scale renewable energy investment [4]. Korea government implements strategy to enhance the growth of green technologies, such as propose carbon trading system and offer tax reduction for green technology users [5]. Besides that, few intelligent techniques are investigated by the researchers to ensure smooth connection of renewable energy sources to the grid [6]. Renewable energy is the energy which is converted from renewable resources, such as solar, wind, hydroelectric, biomass, geothermal and ocean energy. With the increment of renewable energy usages, the portion of fossil fuel usages will decrease, which in turn reduce the carbon emissions. Biomass energy is one of the renewable energies. Biomass is environmental friendly and easy to get [7]. Biomass is suitable to be used as an alternative energy in Malaysia due to the abundance of plantation and forest. Malaysia government also give incentives to promote the growth of biomass development [8]. Besides anaerobic digestion, biomass can be transformed into gas through combustion process. However, direct burning of biomass will produce some impurities which may damage the engine or turbine. To generate a cleaner fuel, biomass can be converted into producer gas (PG) through gasification process. However, the heating value of PG is lower compared to fossil fuel. More residence time is required for PG to burn completely. High residence time of burning will result in high NO_x formation. NO_x gas is poisonous gas which causes acid rain, smog and damage to human health. To mitigate NO_x emissions, several combustion methods are proposed, such as staged combustion, high temperature air combustion (HiTAC), moderate or intense low-oxygen dilution (MILD) combustion and porous media combustion (PMC).

Staged combustion is good at minimizing the NO_x formation. It is divided into air staged and fuel staged combustion. In staged combustion, air or fuel enter the combustor by few stages to prevent the hot spots and long residence time of hot gas in the combustor. A combustor with continuous air staging (COSTAIR) concept released the NO_x emissions below 28 ppm [9]. Based on the study of Goh et al., NO emissions reached 1 ppm by using the fuel staged combustor [10]. Another design of staged combustor contained primary, secondary and dilution ports. Primary port increased fuel-air mixing and flame stabilization, secondary port was used to control NO_x formation and dilution port was used for gas cooling and NO_x reduction [11]. A triple air-staged combustor achieved better NO_x reduction efficiency compared to single air-staged combustor, where the triple air-staged combustor emitted 534.4 mg/m^3 of NO_x gases while single air-staged combustor released 729.8 mg/m^3 of NO_x gases [12]. Air staged combustion is commonly used in gas turbines since it provides higher tolerance for the use of higher excessive air supply without affecting flame stability [13]. It is also essential for the cooling of exhaust gas prior to the turbine using the final air stage known as dilution zone [14].

In HiTAC, the temperature of inlet air is higher than the fuel auto-ignition temperature and the combustion occurs under low oxygen concentration condition [15]. Exhaust gases transfer heat to the inlet air through heat exchanger, thus higher inlet temperature of combustion air can be achieved. The increment of combustion temperature enhances the combustion of low heating value fuels. Differences between HiTAC and ordinary combustion were investigated [16]. Fuel and air were injected into furnace close together during ordinary combustion while preheated air and fuel were injected separately during HiTAC. In ordinary combustion, air directly is mixed with fuel and the flame is concentrated in certain region. Thus, high temperature region is formed, and this would increase the NO_x formation. In HiTAC, preheated air, fuel and recycled exhaust gas were mixed, and concentrated flame was avoided. Since the fuel was distributed evenly, hot spots were minimized in the furnace, thus less NO_x molecules were formed. Several advantages of using HiTAC were discussed in literature [17]. HiTAC technology provides energy savings since thermal energy is extracted from waste gases through regenerative type heat exchangers. CO_2 and NO_x emissions can be reduced through implementation of HiTAC technology. Higher thermodynamic efficiency is achieved due to heat and gas recirculation.

MILD combustion occurs when inlet temperature of reactant mixture is higher than self-ignition temperature and the maximum allowable elevation in temperature with respect to inlet temperature is lower than self-ignition temperature [18]. The operation of MILD combustion is almost same as HiTAC. The main difference between MILD combustion and HiTAC is maximum allowable temperature increase with respect to inlet temperature, where in HiTAC it is higher than self-ignition temperature while for MILD combustion case is lower than self-ignition temperature. According to the study of Noor et al. [19], main requirements of achieving MILD combustion are high temperature of preheating air and high speed injection of air and fuel. MILD combustion normally occurs under condition less than 5-10% oxygen concentration. NO_x formation can be reduced due to low percentage of oxygen concentration. Thermal efficiency of MILD combustion increases through the recycle of waste heat from flue gases. Based on the study of Kuang et al., high inlet velocity strengthens the flow recirculation and makes the reaction zone of furnace more uniform [20].

In PMC, a porous medium is inserted into the combustor to improve the combustion efficiency and reduce emissions. Porous medium is an object with connected voids, and it is normally made of heat-resistant material. Several benefits of PMC were described in literatures [21]. When the combustion air passes through the solid porous medium, small vortices are formed and the flow turns into turbulent. High thermal conductivity of solid porous medium enhances the heat conduction from flame zone to non-flame zones, which in turn reduces the maximum flame temperature. Thus, NO_x formation is mitigated. Although flame temperature decreases, solid porous medium has higher radiative heat transfer compared to the gases, thus PMC can release low NO_x emissions without sacrificing the combustor efficiency. Based on the study of Chen et al., Porous media reduces the temperature of reaction zone which in turn decreases the NO_x formation of the combustor [22].

Only few studies investigated the use of air staging for the combustion of low-grade PG from air gasification. Factors such as the required space after the mixing zone compared to the mixing zone length to achieve stable PG flame were not investigated in literatures. Current study focuses on the geometry design of staged combustor. To investigate the combustion characteristics of staged combustor, effects of combustor diameter, air distributor to total combustor diameter ratio (DR) and air distributor to total combustor length ratio (LR) are studied. The optimum design will be chosen based on the level of outlet temperature, CO and NO_x emissions.

2. Methodology

Low-grade biomass wood producer gas (PG) was used as fuel source in current study. The combustor geometry was drawn using SOLIDWORKS 2020 software and the combustion characteristics of combustor were simulated through ANSYS-FLUENT 19.2 software. Standard k-ε model and non-premixed combustion model were applied to the simulation.

2.1. Fuel

Combustion characteristics of low-grade producer gas (PG) burning in staged combustor was investigated in current study. Low heating value (LHV) of PG through gasification is normally around 4-6 MJ/m^3 [23]. To study the characteristics of low-grade PG, a PG with LHV 3.99 MJ/m^3 was chosen from the experimental work in literature [24]. The low-grade PG was converted through air gasification of biomass wood. The composition of PG is shown in Table 1.

Table 1. Composition values of low-grade PG [24]

Species	Composition (vol %)
H_2	5.0
CO_2	2.1
O_2	0.9
N_2	62.5
CH_4	0.3
CO	29.2

2.2. Geometry of staged combustor

Figure 1 shows the 3D geometry of conventional staged combustor which was drawn using SOLIDWORKS 2020 software. The combustor operates based on continuous air staging method. An air distributor is installed into the staged combustor. Total 16 openings are distributed around the surface of air distributor. Fuel is injected into the combustor through the inner tube of air distributor while air is continuously supplied into the chamber through the openings of the air distributor. The staged combustor is divided into two zones, which are air distribution zone and reaction zone. In air distribution zone, the air is distributed evenly to mix with fuel so that hot spots are avoided. In reaction zone, fuel will combust and release heat. NO_x formation is highly dependent of flame temperature and residence time. With uniform fuel-air mixing in staged combustor, the high temperature flame would not concentrate at certain part of combustor. In contrast, the flame temperature would decrease and distribute across the combustor. Besides that, the staged feature of combustor reduces the reaction time between nitrogen gas with the high temperature gas, thus less NO_x molecules are formed. The geometry is used to investigate the combustion characteristics of staged combustor. Effects of geometry manipulative variables, such as combustor diameter, air distributor to total combustor diameter ratio (DR) and air distributor to total combustor length ratio (LR) are studied. Combustor diameter is varied from 150 mm to 350 mm, DR is varied from 0.4 to 0.8 and LR is varied from 0.2 to 0.5.

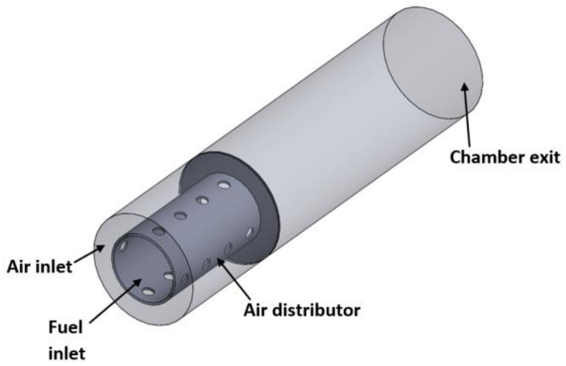


Fig. 1. Geometry of staged combustor.

2.2.1 Geometry parameters

Main geometry parameters in staged combustor are chamber diameter, air distributor to total combustor diameter ratio (DR) and air distributor to total combustor length ratio (LR). For variation of DR_{staged}, diameter of air distributor (D_{dis}) was varied with the constant value of total combustor diameter (D_t). For variation of LR, length of air distributor (L_{dis}) was varied with the constant value of total combustor length (L_t). Both D_{dis}, D_t, L_{dis} and L_t were illustrated in Figure 2 while calculation formulas of DR and LR were shown in Eq.1 and Eq.2 respectively.

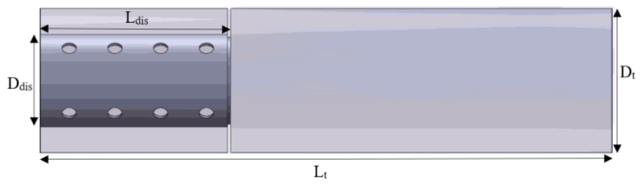


Fig. 2. Schematic diagram for DR and LR formula.

$$\text{Air distributor to total combustor diameter ratio (DR)} = \frac{D_{dis}}{D_t} \quad (1)$$

$$\text{Air distributor to total combustor length ratio (LR)} = \frac{L_{dis}}{L_t} \quad (2)$$

2.3. CFD simulation

Boundary conditions, turbulence and combustion models for CFD simulation were set. Standard k-ε model was applied to investigate the turbulent flow within the combustor while non-premixed combustion model was used to study the combustion characteristics of the fuel burning in the combustor.

2.3.1. Turbulence and combustion model

Standard k-ε model was applied to simulate the flow within the combustor. Turbulent behavior of flow and heat transfer of gas can be well predicted through standard k-ε model. Basic operation of standard k-ε model mainly depends on model transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε) [25]. The transport equations are shown in Eq.3 & Eq.4.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(u + \frac{u_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k \quad (3)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(u + \frac{u_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} + S_\varepsilon \quad (4)$$

where, G_k refers to turbulence kinetic energy generation due to the mean velocity gradients; G_b refers to turbulence kinetic energy generation due to buoyancy; Y_M refers to fluctuating dilatation contribution in compressible turbulence to the overall dissipation rate; $C_{1\varepsilon}, C_{2\varepsilon}, C_{3\varepsilon}$ refer to constants; $\rho k, \rho \varepsilon$ refer to the turbulent Prandtl numbers for k, ε; and S_k, S_ε refer to user-defined source terms.

Non-premixed combustion model was used in the current study. In non-premixed combustion, fuel and air are injected into the combustor separately. For non-premixed combustion model, fuel compositions can be predefined through probability density function (PDF). This allows the composition of PG to be defined accurately before running the simulation.

2.3.2. Boundary conditions

Boundary condition settings are important in CFD simulation. By setting the boundary conditions, unnecessary works of simulation can be avoided and it can guide the simulation into the correct direction. The fuel inlet was set at a relatively low flow rate, which was 0.003 kg/s. To run the operation in stoichiometric condition, air inlet flow rate was set at 0.00256 kg/s through calculation. Initial gauge pressure was set at 0.1 atm to ensure the flow will head to the outlet. By considering the increment of temperature through air compressor, temperature of air inlet was set at 323 K. The temperature of fuel inlet was set at ambient temperature (300 K). k-ε model was used to simulate the turbulent flow within the combustor. The boundary conditions of the staged combustor are shown in Table 2.

Table 2. Boundary conditions of staged combustor

Air inlet	Mass flow rates = 0.00256 kg/s Initial gauge pressure = 0.1atm Temperature = 323K Mean mixture fraction = 0 Turbulence specification method = K and Epsilon
Fuel inlet	Mass flow rate = 0.003 kg/s Initial gauge pressure = 0.1atm Temperature = 300K Mean mixture fraction = 1 Turbulence specification method = K and Epsilon
Pressure outlet	Back flow temperature = 1000K Turbulence specification method = K and Epsilon

3. Model Validation

Study of Dattarajan, Kaluri and Sridhar was used to validate the setup of the current model [26]. Standard k- ε model was used in the current model validation and its results were compared with the study of Dattarajan, Kaluri and Sridhar. A cyclone combustor was used in the reference study. The reference study operated in non-premixed combustion model which was same as current simulation study. Fuel used in the reference study was biomass producer gas. The air and fuel flow rate used in the reference study were 75.4 kg/hr and 52.2 kg/hr, respectively, the equivalence ratio was 0.826. To obtain the model validation, a new geometry which similar to the geometry used in the reference study was drawn and tested with current simulation setting. Comparison of temperature values within the center region of combustor along the combustor axis were shown in Figure 3. The results of CFD modelling predicted were slightly higher than experiment results but lower than CFD results of the reference work. The higher temperature of CFD modelling predicted compared to experiment results of reference work was attributed to heat losses in the experiment. The percentage of temperature difference between CFD modelling predicted with experiment results was only 2.5%. Since the percentage difference was still in acceptable range, this model was applied to the current study.

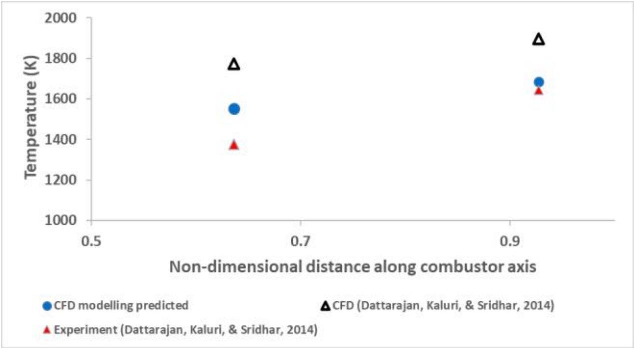


Fig. 3. Model validation.

4. Result and Discussions

The effects of combustor diameter, DR and LR on combustion performance of staged combustor were investigated. The responses of the study were outlet temperature, CO and NO_x emissions.

4.1. Effect of combustor diameter

Figure 4 (a) shows the effect of the increase in combustor diameter in the range of 150-250 mm on mean temperature at chamber outlet, CO and NO_x emissions. The effect of diameter increase on the outlet temperature was not significant in the range of 200-300mm. On the other hand, the extreeme reduction of diamter below 200mm resulted in noticible disturbemce in flame as it was in direct contact with the chamber walls in the reaction zone. This flame disturbence showed negative effect on combustion completion which resulted in elevation in CO emissions at the smallest diamter. Larger cross sectional area of combustor reduced the axial velocity of flow particles through the combustor. Residence time of flow particles increased due to decrement of axial velocity. Higher residence time enhanced the fuel-air mixing level and the fuel molecules were decomposed more completely. Thus, more CO molecules were converted into CO_2 molecules. Mean outlet temperature rised about 3% when combustor diameter increased from 150mm to 350mm. Figure 4 (b-f) clearly shows that stable high temperature flame was distributed uniformly at the combustor outlet with larger diameter compared to smaller diameter. This is due to the longer residence time given for fuel molecules to mix with air molecules and combust in larger combustor diameter. Hence, more heat energy was released. NO_x emissions increased from 12 ppm to 51 ppm when combustor diameter increased from 150mm to 350mm. NO_x formation is highly influenced by residence time and temperature. Higher residence time and temperature of large diameter combustor promoted the formation of NO_x molecules. Thus, more NO_x molecules were released.

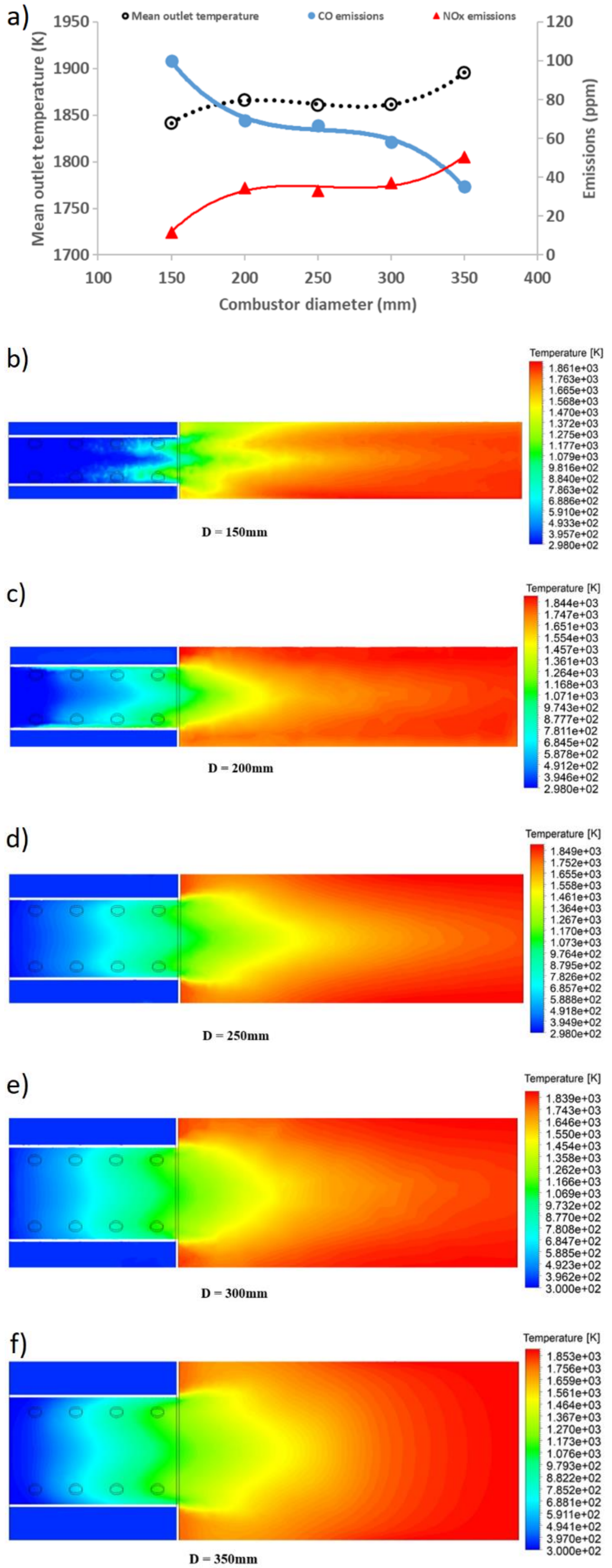
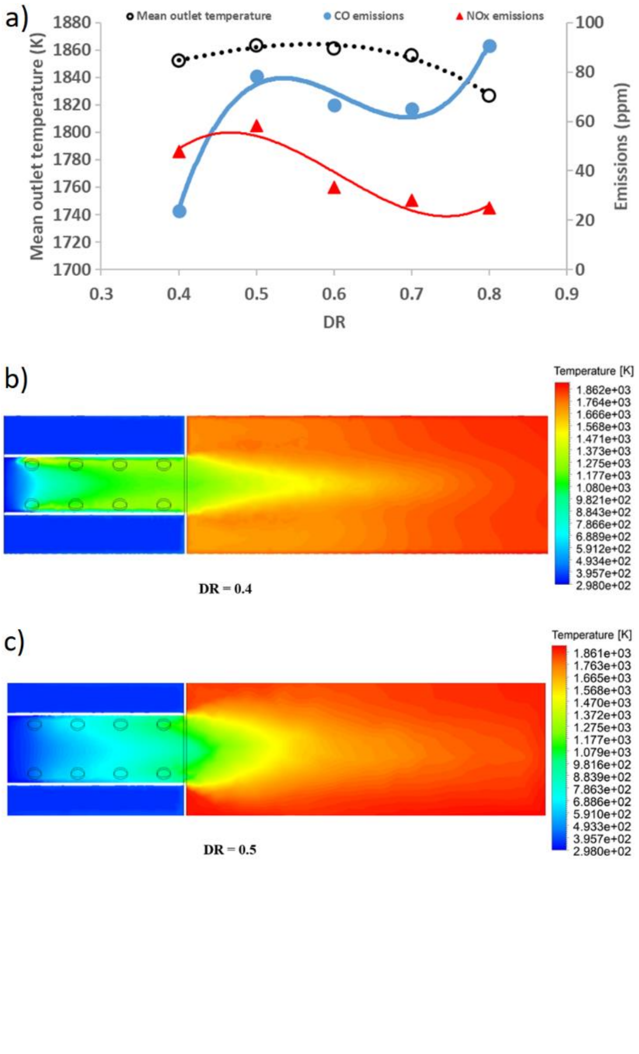


Fig. 4. Effect of combustor diameter on temperature, CO and NO_x emissions.

4.2. Effect of air distributor to total combustor diameter (DR)

Figure 5 (a) shows the effect of DR in the range of 0.4-0.8 on the mean outlet temperature, CO and NO_x emissions. CO emissions increased from 24 ppm to 91 ppm when DR increased from 0.4 to 0.8. This could be attributed to the less efficient merge of air and fuel at centre region when the diameter of air distributor is enlarged. The poor mixing of fuel and air reduced the combustion reaction completion, thus, less CO molecules were converted into CO_2 molecules in combustor with larger DR. Moreover, DR has shown considerable effect on the flame shape in the reaction zone. Sharp nerrow flame with high extention through the reaction zone was achieved at the lowest DR of 0.4. While inclreasing DR resulted in gradual reduction in the flame extention while the flame diameter is increased untill it touches the chamber walls at the largest DR of 0.8. Mean outlet temperature difference between 0.4 with 0.6 DR was only 9 K while mean outlet temperature decreased from 1861 K to 1827 K when further increased to 0.8 DR. The decrement of temperature was due to lower mixing rate of fuel and air in the centre region of larger diameter air distributor, which in turn reduced the reaction rate of the combustion. Based on Figure 5 (b-f), fuel and air mixed rapidly and start released heat at early stage of 0.4 DR combustor which had a big contrast compared to 0.8 DR combustor. Hence, lower combustion rate and outlet temperature existed in 0.8 DR combustor. The temperature drop in larger DR combustor resulted in less formation of NO_x molecules. Thus, NO_x emissions decreased with the increment of DR.



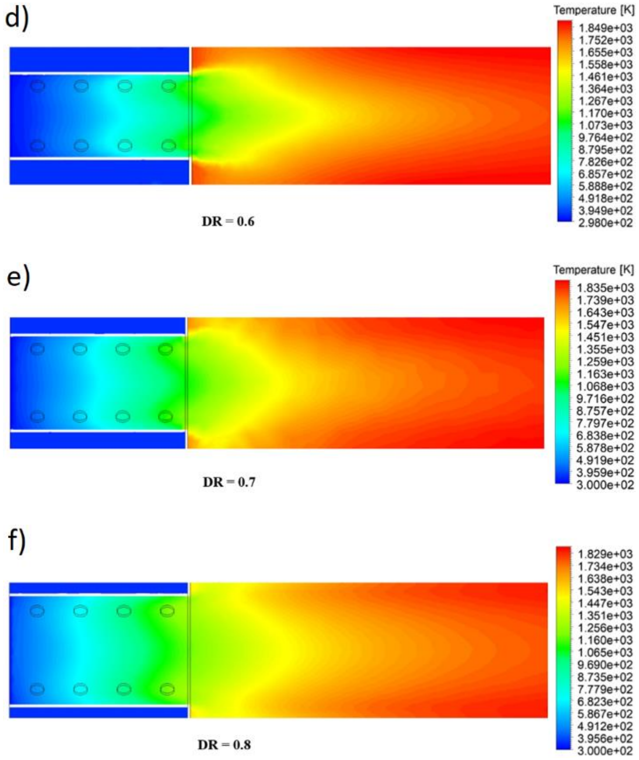


Fig. 5. Effect of DR on outlet temperature, CO and NO_x emissions.

4.3. Effect of air distributor to total combustor length (LR)

Figure 6 (a) shows the effect of LR in the range of 0.2-0.6 on mean outlet temperature, CO and NO_x emissions. CO emissions elevated with the increment of LR. With the extension of air distribution zone, length of reaction zone shortened, thus less residence time was given for combustion to occur and less fuel molecules were decomposed. Hence, less CO molecules were converted into CO_2 molecules. Mean outlet temperature decreased with the increment of LR. This could be explained based on Figure 6 (b-f). 0.2 LR combustor (Figure 6 b) consisted of shorter air distribution zone and longer reaction zone. Long reaction zone enabled the complete combustion of the majority of fuel molecules and stable high temperature flame appeared at the combustor outlet. Compared to 0.2 LR combustor, longer air distribution zone and shorter reaction zone existed in the 0.6 LR combustor (Figure 6 f). Although fuel and air could be mixed well in long air distribution zone, the combustion reaction did not have enough time to fully burn in 0.6 LR combustor due to the short length of the reaction zone. Another considerible effect of LR was noticed on the flame caracteristics at the exit of the air distributer. Shorter distributer resulted in more uniform conical shape. On the other hand, extending the distributer length resulted in early flame start inside the distributer. This resulted in multiple flame probagation fronts, breaking the single conical flame shape, and reducing flame stability. However, NO_x emissions did not change much with the variation of LR as the maximum temperature at the reaction zone was nearly identical for all cases.

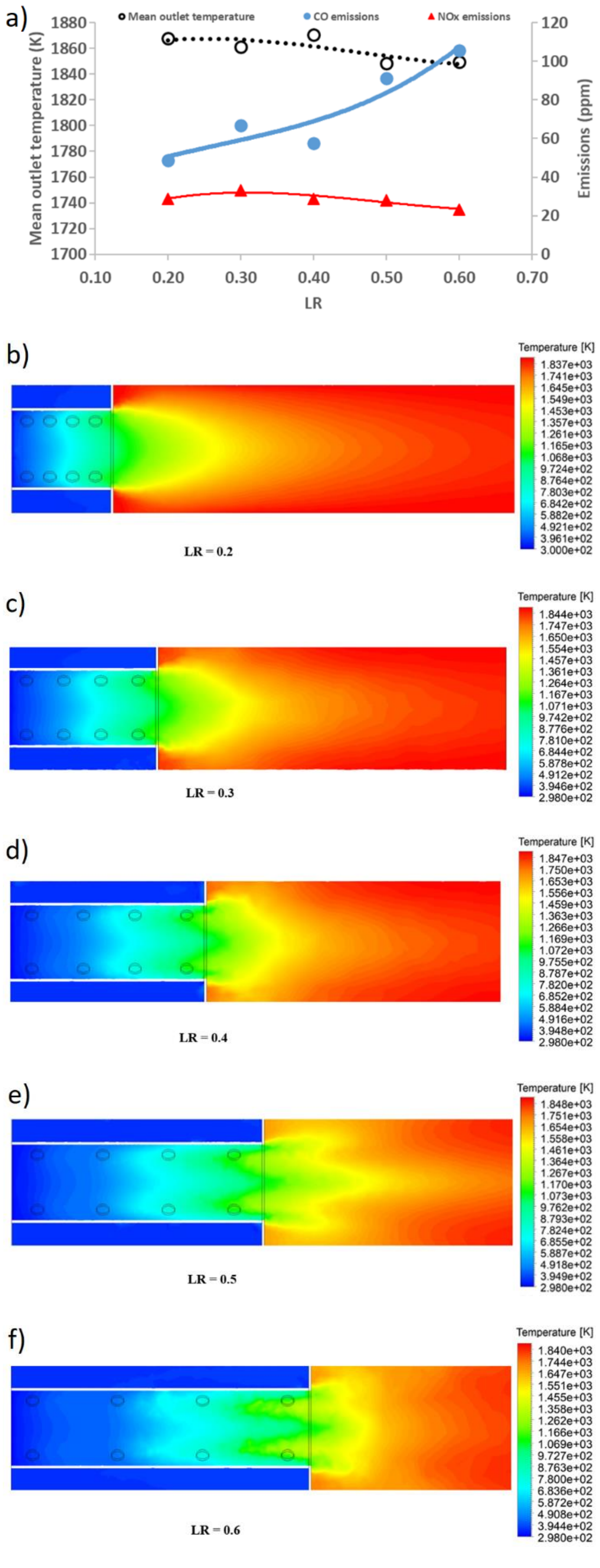


Fig. 6. Effect of LR on outlet temperature, CO and NO_x emissions.

4.4. Performance of combustor

By comparing the outlet temperature, CO and NO_x emissions between different geometry parameters of combustor, chamber geometry with 250 mm diameter, 0.6 DR and 0.2 LR was chosen as the optimum design. It achieved 1867 K outlet temperature, 48.77 ppm CO emissions and 28.56 ppm NO_x emissions. Table 3 shows the performance comparison between current study with the previous staged combustion works in literatures using different types of fuels. The use of flame tube for air staging is the dominant design for micro gas turbine combustors for all the reviewed studies here. However, the major difference for the combustor design used in the current study is the removal of the dilution zone that is replaced with secondary reaction zone. Also, fuel atomization is needed when using liquid fuels [11], [13]. NO_x emission is one of the most important parameters to evaluate the performance of staged combustor since it indicates the good control of maximum temperature and oxygen concentration inside the chamber. Based on Table 3, current staged combustor achieved lower NO_x emissions compared to the other air staged designs. It proved that the current staged combustor was effective in reducing NO_x emissions. However, having pressurized chamber in gas turbine application commonly enhances combustion intensity resulting in lower CO emissions compared to the current design with atmospheric chamber. Extreme low CO emissions below 5 ppm were achieved at 3 bar absolute pressure [11], while operating at lower pressures below 1.5 bar [13], [14] resulted in higher CO emissions comparable to the values in the current study.

Table 3. Performance comparison between current study with previous staged combustion works

Fuel	NO_x emissions (ppm)	CO emissions (ppm)	Reference
Producer gas	28	48	Current study
LPG	42	51	[13]
Diesel	31	95	[14]
Biodiesel	36	4	[11]
NH_3 / H_2	50	-	[27]
NH_3 / CH_4	30	10	[28]

5. Conclusions

Current combustor operated based on continuous air staging concept. Combustion characteristics of PG burning in staged combustor were investigated through ANSYS-FLUENT software. Effects of combustor diameter, DR and LR on combustion performance of staged combustor were studied. Combustor with smaller diameter and larger DR emitted less NO_x gases but it caused higher CO emissions and lower outlet temperature. With the increment of LR, CO emissions increased and outlet temperature reduced. However, NO_x formation did not change much with the variation of LR. Based on the responses of outlet temperature, CO and NO_x emissions, chamber geometry with

250 mm diameter, 0.6 DR and 0.2 LR was chosen as optimum design. The optimum geometry released 1867 K outlet temperature, 48.77 ppm CO emissions and 28.56 ppm NO_x emissions. It achieved high outlet temperature with the acceptable CO and NO_x emissions.

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