Numerical Design of Cylindrical Combustor for Micro Gas Turbine Application

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Abstract

Nowadays, electrical energy is important in daily life and industry. The micro gas turbine is one type of technology used in generating power, with a small scale and reliably distributed power system. Advantages of the micro gas turbine are that it is cost effective, reliable and low maintenance. Fuels for producing electrical energy via the micro gas turbine are the subject of this study because the combustion efficiency of the optimal cylindrical combustor will change as it is used with different fuels. Therefore, the re-designed configuration of the combustor is necessary for improving the efficiency of the micro gas turbine. A computational design is implemented because of the accuracy in calculation and ease of modelling. This research mainly focuses on the efficiency of the combustor in the micro gas turbine system as the main fuel supplied is changed from LPG to natural gas (methane). The geometry of the cylindrical combustor was constructed by SolidWorks and then numerically solved by Fluent. The cylindrical combustor has a 50 mm flame holder, 600 mm chamber height, and four holes of 6, 8, 10 mm for the dead zone, combustion zone and dilution zone, respectively. Different fuels were used in simulations with a non-premix combustion model, consisting of the standard k-ε model for turbulent flow, energy equation, continuity equation and P-1 radiation model. All of the equations were solved by a finite volume method. The results were validated with the experimental data of (Enagi et al., 2017) and show that both cases provide similar temperature contours. The average temperatures at the outlet of

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LPG and methane fuels were 1318 K and 1312 K, respectively. In contrast, the mass fraction distributions of the product gases, i.e. H_2O and CO_2, in both cases were different. This resulted in different efficiencies of combustion, in which the combustor with methane was shown to provide higher combustion efficiency.

Keywords: Combustor, Combustion efficiency, Micro gas turbine, Non-premix combustion

1. Introduction

A micro gas turbine engine is a compact size technology for generating power, which has reliability, high efficiency and low maintenance. Micro gas turbines operate using the Brayton cycle, which is comprised of a gas turbine, combustor and compressor. The Brayton cycle has three procedures: first, air input into the compressor increases pressure and is feed to the combustor, second, fuel and air enter the combustor for combustion and transfer heat energy to the gas turbine, and finally, the gas turbine receives heat energy and changes it to mechanical energy for generating power.

The combustor is important for the Brayton cycle because this component is where combustion takes place, which affects the enthalpy input to the gas turbine, and if the combustor is highly efficient, the overall system is also. Fuels are the main element of a combustor because fuels have diverse properties after combustion and different heating values, which affects efficiency. Therefore, combustors are interesting to study for improvement of their efficiency.

There are two main approaches to the combustor study: experimental and simulation. The experimental method has accurate results but high costs and is time-consuming due to the need to construct a prototype and the set-up process. The simulation approach is convenient in terms of the calculations for design and development. This approach can construct the geometry and calculate complex governing equations by computer. However, the results from simulations are less accurate than experiments because the simulation technique might be incorrect due to limitations of the computer. Nevertheless, this approach can predict the performance of the system before design and save time. For these reasons, simulation techniques are utilised in studies for low cost and time constraint cases.

Many researchers have studied micro gas turbine technology; (Pilavachi, 2002) reviewed micro gas turbine technology and found that micro gas turbines usually produce between 25 and 500 kW of electrical power, have low emissions, low maintenance costs and reliable power, and can be operated with various kinds of fuel (Enagi et al., 2017) simulated a combustor of the micro gas turbine for optimisation of the dimensions of the chamber. The best chamber geometry adopted after optimisation was a 50 mm holder diameter and 60 cm height, having four holes of 6, 8 and 10 mm with a dead zone between the combustion zone and dilution zone. Then, data from the simulations were used for construction of the chamber. The results of the experimental test of the chamber with LPG fuel were stable (Al-attab and Zainal, 2014) evaluated performance of the micro gas turbine process with experiments using LPG and PG fuels. The system combining heat and power achieved an overall efficiency of approx. 58% with 35 kWth hot air production as the thermal output system (Ahn et al., 2001) constructed a simulation model and compared it with field testing for determination of combustion efficiency factors. The results demonstrated that relative aerodynamic pressure loss, position of the gas temperature maximum in the radial direction, and position of maximum linear wall temperature can affect the combustor efficiency (Shah and Banerjee, 2016) conducted an experimental investigation on a can combustor to study the effects of the swirler vane angle and air-fuel ratio. The experiment used methane for the main fuel. They reported a peak temperature at the can combustor outlet, when increasing the air-fuel ratio and that the temperature gradient near the wall decreases as the swirler vane angle is increased (Ghenai, 2010) conducted numerical investigation of combustion of a syngas fuel mixture in a gas turbine can combustor. The results from this study showed the change in gas turbine can combustor performance with the same power generation when natural gas or methane fuel is replaced by syngas fuel.

The present work focuses on the efficiency of the combustor in the micro gas turbine system as the main fuel supplied is changed from LPG to natural gas (methane). Both combustor geometry and boundary conditions of this study were performed in previous research (Enagi et al., 2017) and the results of this research were validated by Enagi et al. (2017). .

2. Theory

Computation of fluid dynamics is the main procedure for prediction of combustion effects; thus, mathematical models are necessary. In this study, a non-premix equation, momentum equation, energy equation, radiation P-1 equation, and standard k-ε model were used. All of the equation details are shown below.

2.1. Continuity equation

$$\frac{\partial \overline{\rho u_i}}{\partial x_i} = 0, \tag{1}$$

where ρ is the density of fluids, u_i is the flow fields of all fluids in the system, and x_i is a direction axis for fluid fields.

2.2. Momentum equation

$$\frac{\partial\left(\overline{\rho u_i u_j}\right)}{\partial x_j} = -\frac{\partial \overline{P}}{\partial x_i} + \frac{\partial\left(\overline{\tau_{ij}} + \overline{t_{ij}}\right)}{\partial x_j}, \tag{2}$$

where $\overline{t_{ij}}$ is the viscous stress tensor, and $\overline{\tau_{ij}}$ is the average Reynolds stress tensor.

2.3. Turbulent kinetic energy equation

$$\frac{\partial\left(\overline{\rho k u_j}\right)}{\partial x_j} = \frac{\partial\left[\left(\mu + \mu_t/\sigma_k\right)\left(\partial \overline{k}/\partial x_j\right)\right]}{\partial x_j} + G_k - \overline{\rho \varepsilon}, \tag{3}$$

where $\sigma_k = 1, G_k$ is the production of turbulent kinetic energy, and ε is the dissipation rate of turbulent kinetic energy.

2.4. Dissipation of kinetic energy

$$\frac{\partial\left(\overline{\rho k u_j}\right)}{\partial x_j}=C_{\varepsilon 1}\frac{\bar{\varepsilon}}{k}G_k+\frac{\partial\left[\left(\mu+\mu_k/\sigma_{\varepsilon}\right)\left(\partial\bar{\varepsilon}/\partial x_j\right)\right]}{\partial x_j}-C_{\varepsilon 2}\bar{\rho}\frac{\overline{\varepsilon^2}}{k}, \tag{4}$$

where $C_{\varepsilon 1}=1.44$, $C_{\varepsilon 2}=1.92$, and $\sigma_{\varepsilon}=1.3$.

2.5. Mixture fraction equation

$$\frac{\partial\left(\overline{\rho f u_j}\right)}{\partial x_j}=\frac{\partial\left[\left(\mu_t/\sigma_t\right)\left(\partial\bar{f}/\partial x_j\right)\right]}{\partial x_j}+S_m, \tag{5}$$

where f is the mixture fraction, which can be written in terms of the elemental mass fraction as:

$$f=\frac{Z_k-Z_{k,O}}{Z_{k,F}-Z_{k,O}}, \tag{6}$$

where Z_k is the element mass fraction of an amount of element k. Subscripts F and O denote the fuel and oxidizer inlet stream, respectively. For the mixture fraction method, the equilibrium chemistry PDF model is used.

2.6. Energy equation

$$\frac{\partial\left(\overline{(\rho E+p)uj}\right)}{\partial x_j}=\frac{\partial\left[\left(k_{eff}\right)\left(\partial\bar{T}/\partial x_j\right)-\sum_j h_j J_j+\left(\overline{\tau_{eff}}u_j\right)\right]}{\partial x_j}+S_{h,} \tag{7}$$

where E is the total energy, k_{eff} is the effective conductivity, J_j is the diffusion flux of species, and S_h is the term heat source that includes the radiation and chemical reaction.

2.7. Equation for the P-1 radiation model

$$q_r = -\frac{1}{3(a+\sigma_s) - C\sigma_s}\nabla G, \tag{8}$$

where a is the absorption coefficient, σ_s is the scattering coefficient, G is the incident radiation, and C is the linear anisotropic phase function coefficient.

3. Results and Discussion

The three dimensions were designed by SolidWorks and the combustion process was indicated by Fluent. The combustor dimensions and procedures with the LPG fuel were as performed by Enagi et al. (2017).

3.1. Detail of a combustor drawing

The dimensions of the combustor of the MGT have been optimized by Enagi et al. (2017). The geometry comprises a 50 mm flame holder and 600 mm chamber height, having four holes of 6, 8, 10 mm for the dead zone, combustion zone and dilution zone, respectively. Figure 1 shows the combustor drawing and micro gas turbine system, which in this study is a two-stage turbine with a heat recovery unit for increasing the effectiveness of the system.

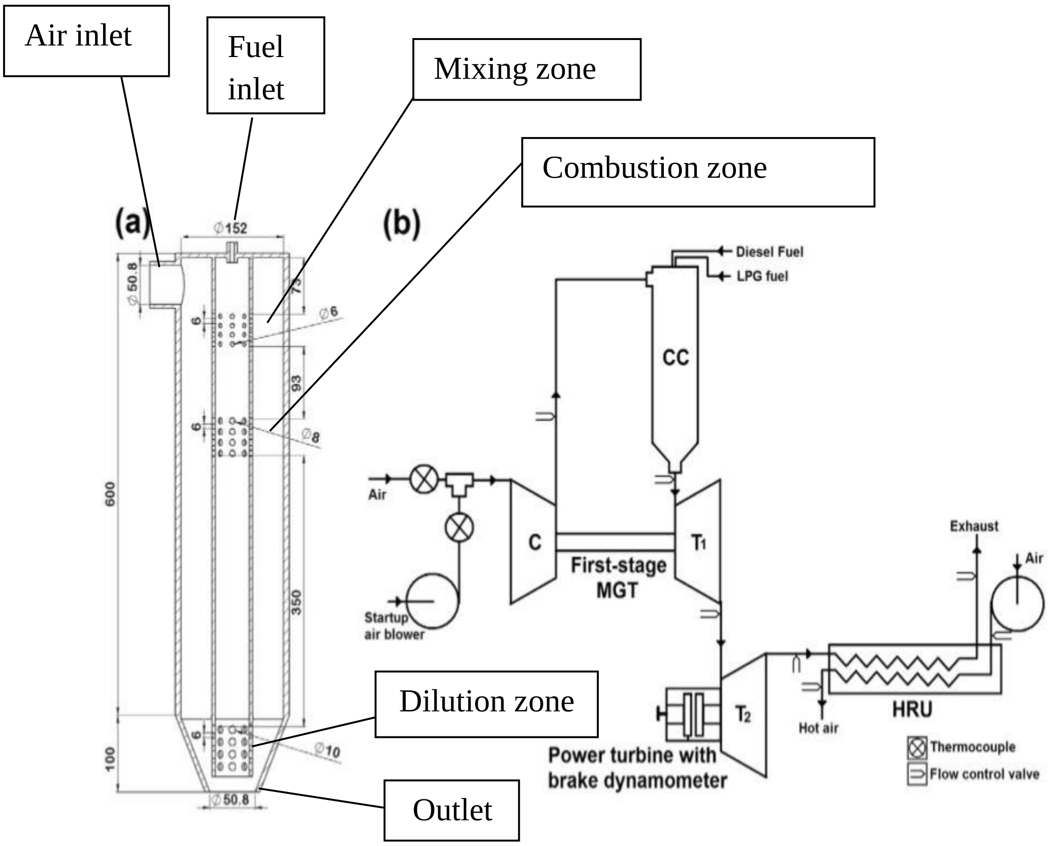


Figure 1. (a) Combustor drawing, (b) System description

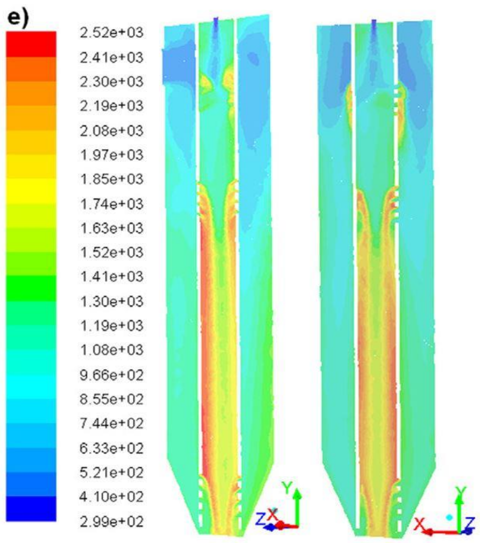
3.2. Validation

Before changing the main fuel, which is the main topic of this research, validation results are necessary for accuracy. This research utilised a simulated model from previous research (Enagi et al., 2017). Table 1 shows the experimental boundary conditions. The temperature and specie contours are compared in Fig. 2.

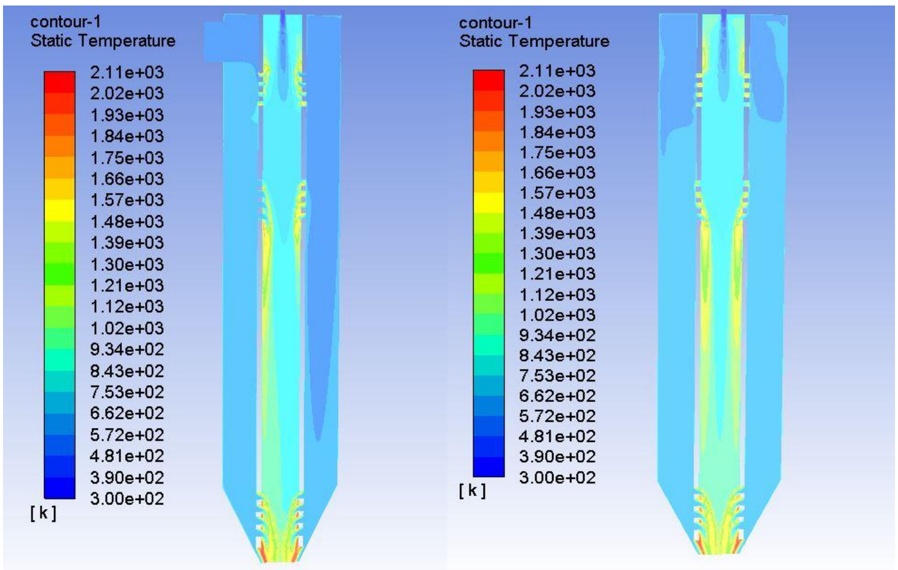
Table 1 Boundary conditions in experiment

Parameter	Value
Fuel inlet	
Temperature	300 K
Pressure	2 bar
Mass flow rate	0.0055 kg/s
Species LPG	Butane 70% Propane 30%
Air inlet	
Temperature	530 K
Pressure	0.7 bar
Mass flow rate	0.07 kg/s
Inner wall	
Emissivity	0.5
Outer wall	
Thickness	6 mm
Emissivity	0.5

In the combustor, the burning occurs in the mixing zone. When the system has burned the fuel, it results in the heat transfer effect. In the combustion zone, the temperature is more intense than that in the mixing zone due to the heat transfer effect and fuel mixing process. The dilution zone is where the flame propagation is found. Figure 1 shows the compared temperature contours of the previous research and this project.



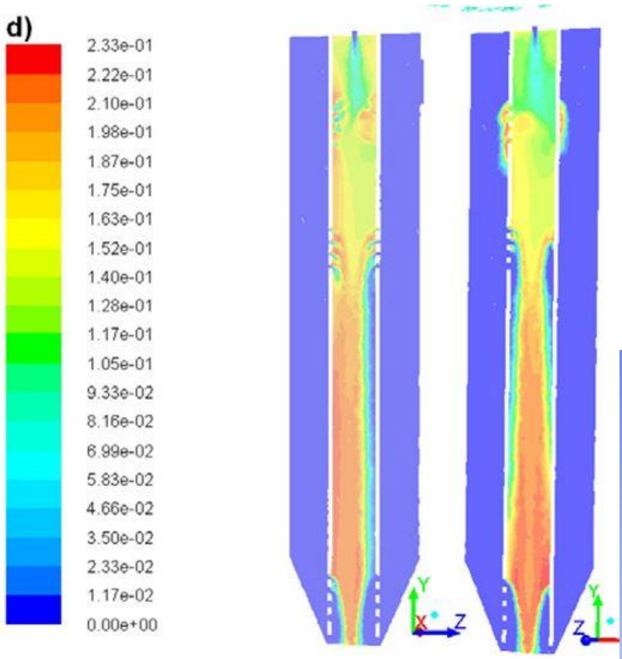
(a) Temperature contour of previous research (Enagi et al., 2017)



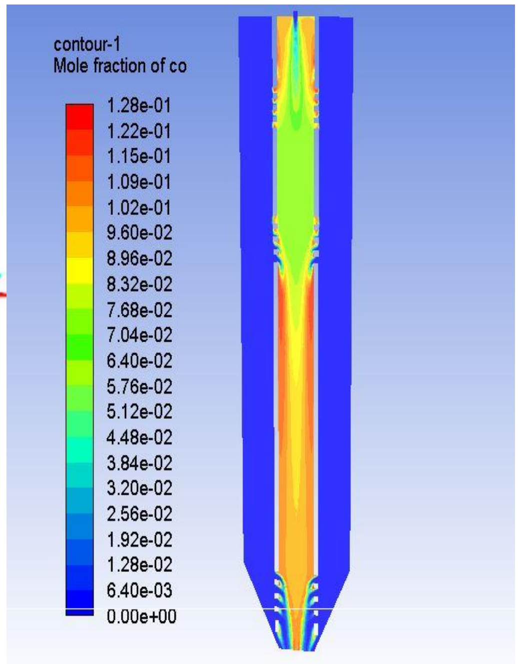
(b) Temperature contour of the current study

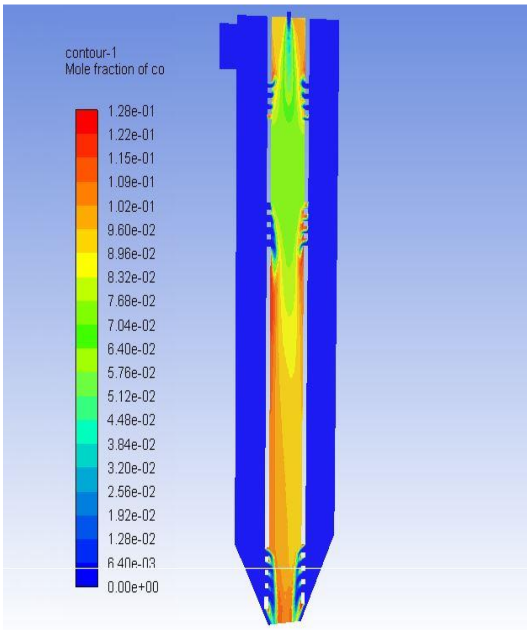
Figure 1. Temperature contours

The simulation model requires verification of the burning process, which is checked using emissions in the combustor. Fig. 2 shows the carbon monoxide and carbon dioxide contours.



(a) Carbon monoxide of the previous research (Enagi et al., 2017)





(b) Carbon monoxide contour of the current study

Figure 2. Carbon monoxide contours

The results of the current study found that the temperature level and mole fraction of the carbon monoxide is lower than Enagi et al. (2017). Flame propagation was observed by temperature and carbon monoxide contours. Both temperature and mole fraction contours performance Enagi et al. (2017) are similar to the current study.

Temperature at the outlet combustor must not exceed 900 $°C$. In addition, this temperature level is optimised for protection of the turbine blade. Table 2 shows the results of the experiment, and simulation results of the previous research (Enagi et al., 2017) and the current simulation. All of the data is compared with the experimental results.

Table 2 Comparison between experimental and CFD simulation results

Parameters	Experiment	Simulation with previous research (Enagi et al., 2017)	Current study	Error (%) Current study compared with experiment
Fuel composition (Vol%)	70% butane and 29.5% propane	70% butane and 30% propane	70% butane and 30% propane	-
Air inlet pressure (barg)	0.7	0.7	0.7	-
Air flow rate (kg/s)	0.07	0.07	0.07	-
Excess air (%)	37	37	37	
Temperature inlet to micro gas turbine ($°C$)	920-960	1,069	1,045.31	13.5-8.8
CO (ppm)	50-60	69	-	-
CO_2 mass fraction	-	-	0.0544	-
H_2O mass fraction	-	-	0.0420	-
Enthalpy of LPG (kJ/kg)			2,304.194	
LHV of LPG (kJ/kg)			44,150	

3.3 Combustor efficiency

The main objective of this research is to compare the combustor efficiency between LPG and methane fuel. Combustor efficiency was calculated by fraction of heat transfer from the outlet to LHV multiplied by work flow at the inlet. The important properties of the methane are found with the experimental boundary conditions. The products of the combustion reaction are also included. Table 3 shows the parameters of the methane for the combustor efficiency evaluation.

Table 3 The simulation results of the methane fuel

Parameter		Value
Temperature inlet to micro gas turbine	°C	1,045.31
CO_2 mass fraction		0.0459
H_2O mass fraction		0.0552
Enthalpy of methane (kJ/kg)		-4,664.4
LHV of methane (kJ/kg)		50,016
Specific heat with constant pressure of CO_2 (kJ/kg-K)		1.18
Specific heat with constant pressure of H_2O (kJ/kg-K)		2.38
Enthalpy of air (kJ/kg)		237.817
Enthalpy of formation CO_2 (kJ/kg)		-8,981.59
Enthalpy of formation H_2O (kJ/kg)		-13,844.7

Tables 3 and 4 lists the data for the finding combustor efficiency. In this case, Equation *(3.1)* was used for finding the enthalpy of the production and the combustor efficiency.

$$H_{product} = \chi_{product}\left[\left(c_p(\mathrm{T}_{outlet} - 298)\right) + (\mathrm{h}_{formation,\, product})\right] \tag{3.1}$$

where $\chi_{product}$ is the mass fraction of the combustion reaction product, and c_p is specific heat of the combustion reaction product.

$$\eta = -\left[\frac{((\frac{A}{F}+1)\times H_{product}) - (\frac{A}{F}\times \mathrm{h}_{air\,inlet}) - (\mathrm{h}_{fuel})}{LHV}\right] \tag{3.2}$$

After substitution of data from Table 3 and Table 4, it was found that the combustor efficiencies with methane and LPG are 23.84% and 29.69%, respectively.

4. Conclusion

In this research, the main objective was finding the combustor efficiency using different fuels. The results of the simulation model were validated by Enagi et al (2017). From the results of this research, it can be inferred that in the system, the burning occurred in the mixing zone, inducing combustion in the combustion zone, and resulting in heat transfer, Finally, the flame was diluted in the dilution zone. The simulation results show that both cases provide similar temperatures and carbon monoxide contours when compared with Enagi et al (2017). The average temperatures at the outlet of LPG and methane were 1318.47 K and 1312.45 K, respectively. The mass fraction distributions of the product gases, i.e. H_2O, CO_2 and LHV, in both cases are different. This led to the different efficiencies of combustion, in which the combustor with LPG was shown to be able to provide higher combustion efficiency.

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