

Numerical analyses and experiment investigations of an annular micro gas turbine power system using fuels with low heating values

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This study investigates the effects of using fuels with low heating values on the performance of an annular micro gas turbine (MGT) experimentally and numerically. The MGT used in this study is MW-54, whose original fuel is liquid (Jet A1). Its fuel supply system is re-designed to use biogas fuel with low heating value (LHV). The purpose is to reduce the size of a biogas distributed power supply system and to enhance its popularization. This study assesses the practicability of using fuels with LHVs by using various mixing ratios of methane (CH₄) and carbon dioxide (CO₂). Prior to experiments, the corresponding simulations, aided by the commercial code CFD-ACE+, were carried out to investigate the cooling effect in a perforated combustion chamber and combustion behavior in an annular MGT when LHV gas was used. The main purposes are to confirm that there are no hot spots occurring in the liners and the exhaust temperatures of combustor are lower than 700°C when MGT is operated under different conditions. In experiments, fuel pressure and mass flow rate, turbine rotational speed, generator power output, and temperature distribution were measured to analyze MGT performance. Experimental results indicate that the presented MGT system operates successfully under each tested condition when the minimum heating value of the simulated fuel is approximately 50% of pure methane. The power output is around 170 W at 85000 r/min as 90% CH₄ with 10% CO₂ is used and 70 W at 60000 r/min as 70% CH₄ with 30% CO₂ is used. When a critical limit of 60% CH₄ is used, the power output is extremely low. Furthermore, the best theoretical Brayton cycle efficiency for such MGT is calculated as 23% according to the experimental data while LHV fuel is used. Finally, the numerical results and experiment results reveal that MGT performance can be improved further and the possible solutions for performance improvement are suggested for the future studies.

numerical analyses, experiment investigations, annular micro gas turbine, low heating values

1 Introduction

The greenhouse effect is an international problem. Controlling emissions of greenhouse gases is an important environmental goal. In response to it, Taiwan has prioritized utilization of renewable energies and the development of new energy sources. As a large amount of biogas can be derived from bio-waste, this fuel source is currently favored in Taiwan. Marsh gas is composed of approximately 50% methane. Although methane is lighter than both air and natural gas, it has similar ther-

mal and physical properties. Methane generated from garbage, which is an air pollutant, can be used as fuel. Notably, generating electrical power using methane gas can reduce the emission of greenhouse gas and simultaneously produce electrical power. Peirs et al.^[1, 2] utilized a micro turbine with a rotor of 10 mm in diameter to generate electrical power using liquid fuel.

Received January 5, 2009; accepted June 15, 2009

doi: 10.1007/s11431-009-0312-0

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Supported by the 'National' Science Council of Taiwan, China (Grant No. NSC 96-2218-E-009-002) and Ted Knoy is appreciated for his editorial assistance

The tolerable maximal speed of ball bearings was the primary constraint on both power generated and efficiency. When attached to a small generator, the turbine generated 16 W of electrical power; overall system efficiency was 10.5%. The rotor was tested at speeds up to 130000 r/min with compressor air at 330°C. The micro turbine was small enough that it could be widely applied; however, some researchers have questioned whether it can use gaseous fuel rather than liquid fuel. Yamashita et al.^[3] examined micro gas turbine (MGT) under experimental conditions using fuels with low heating values (LHVs). Their simulations used liquefied petroleum gas (LPG) diluted with N₂. The efficiencies of all system components were based on temperature and pressure measurements. The proposed MGT system was successfully operated using a fuel with a heating value being 43% of that of LPG, indicating that, when fuels with LHVs are used, the combustor requires no modifications. However, the MGT system in this study was large. Kousuke et al.^[4–6] and Shuji et al.^[7] analyzed a micro-machining gas turbine developed at Tohoku University. In the MGT, most heat was transferred from high-temperature components, such as the combustor and turbine, to low-temperature components, such as the compressor, as the distance separating high-temperature and low-temperature components was close to the width of the turbine. Hence, the temperature gradient in the micro turbine was large when the highest and lowest temperatures in both turbines were equal. Hence, a MGT could not directly reduce the scale while maintaining the same structure as a macro gas turbine because thermal efficiency would be excessively low. Wang and Xi^[8] examined the predicted accuracy of flow structure in CFD analysis. Three perfect gas models and one real gas model were respectively considered to implement the numerical simulation of a centrifugal refrigeration compressor model. The results show that the real gas effects should be considered for the precise prediction of basic fluid flow parameters and flow structure. Zhao et al.^[9] devoted to revealing the hot streaks migration influence on a vaneless counter-rotating turbine. The simulation results show that the secondary flow and buoyancy have minor influence on the turbine stator in the high pressure condition. The authors found that there have been scarce published researches on using the LHV fuel in an annular MGT turbine either numerically or experimentally. In this study, the authors were interested in the thermo-flow

field distribution caused by the combustion behaviors of the renewable LHV fuel in an annular MGT system. A reliable steady state simulation result could help researchers to avoid inappropriate chamber design, which could cause hot spots on the chamber liner or could make the temperature of exhaust gas exceed the maximum allowable temperature that the turbine wheels can tolerate. The second part of this study was experimental work that evaluated the combustion efficiency of an annular MGT while applying the LHV fuel. The LHV gas used in this experiment was CH₄ mixed with CO₂. CO₂ was mixed in the LHV gas because in reality the biogas usually contains some impurity gases due to the limitation of present biochemical technique that cannot obtain very pure methane gas from the purified marsh gas. The experiment results were used to observe and calculate the efficiency of the MGT under the different combinations of parameters, including the mixing ratio of CO₂, rotational speed of compressor wheel, pressure of fuel, and mass flow rate of the fuel. It is well known that the impurities of gas will dilute the concentration of pure methane. This study tries to find the optimal operating condition for our proposed MGT to reach maximum combustion efficiency.

2 Experimental

2.1 A briefing of MGT system

Figures 1 and 2 display the experimental layouts and the major parts of the MGT. The MGT shown is basically composed of compressor (radius: 27 mm) (Figure 2 (a)); combustion chamber (length: 52.4 mm, radius: 38 mm) (Figure 2 (b)); and turbine wheel (radius: 25 mm) (Figure 2 (c)). The compressor compresses the incoming air into high pressure. The combustion chamber burns the fuel as well as produces high-pressure and high-velocity production gases. The turbine is energized by the high-pressure and high-velocity gas flowing from the combustion chamber. The most common shaft design in a MGT is the single shaft design that a radial centrifugal compressor and turbine are attached to a shaft which was also adopted in this study. Figure 3 shows the schematic view of the inner MGT in 60° section cut. The inverse injection fuel system was adopted. After the performance of MGT while using LHV fuel was tested, the capability of extending the use of MGT to building an electric power system was examined. Figure 4 shows the transmission gearbox and generator, which were placed

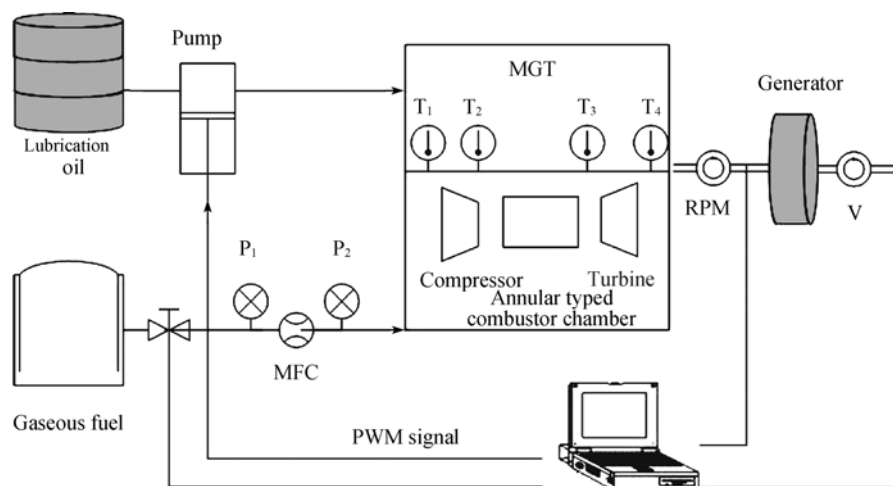


Figure 1 Experiment setup.

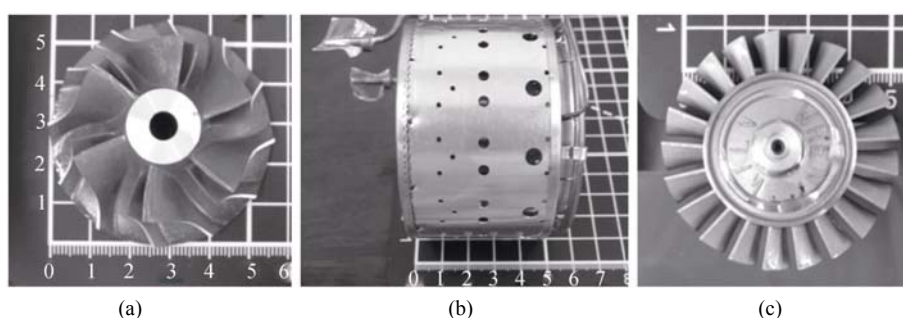


Figure 2 Pictures of the major parts of a MGT. (a) Compressor; (b) combustion chamber; (c) turbine wheel.

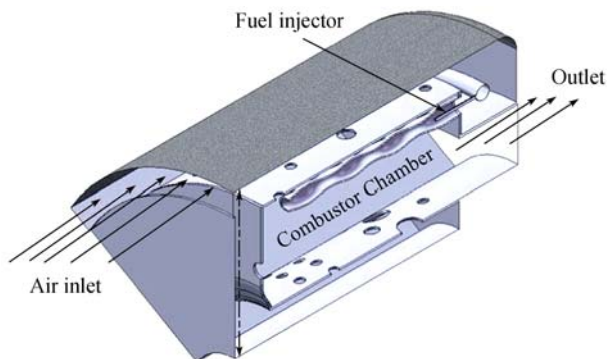


Figure 3 Schematic view of a 60° section cut of the inner MGT.

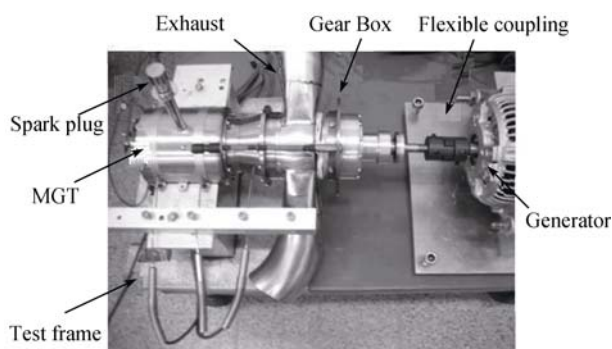


Figure 4 Picture of the MGT with an attached gearbox and generator.

on a test frame made of 6061 aluminum alloy. The generator is connected to the MGT via a transmission gear-box, which transforms the power generated by the MGT into axial work. This axial work drives the generator to produce electrical power. The MGT thermal system is very similar to heavy-duty turbines. However, due to the lower inertia of the compressor-turbine-generator shaft, it easily gets a high rotational speed and may reach 100000 r/min. Consequently, the MGT usually uses gear speed reducers as transmission gearbox to diminish the rotational speed to match the AC power grid frequency of the power generator. The generator used in this study is a commercial product which is taken from a motorcycle. The generated current is rectified to direct one for commercial use and the load is a set of high-power headlight, which is about 200 W. The blower used to drive the MGT is connected to the MGT air inlet via a rubber tube. When the MGT reaches stable rotational speed, the rubber tube is removed and the turbine wheel keeps rotating by absorbing power from the air combustor. Fuel pressure and flow rate at inlet nozzles of the combustion chamber can be adjusted by using a

pressure valve and flow meter. A mass flow controller (MFC) is a closed-loop device that sets, measures, and controls the flow of a particular gas or liquid. In this study, a TC-1350 MFC produced by Tokyo Keiso Company was adopted. The rotational speed of the MGT was measured using an induction tachometer installed close to the compression fans. A K-typed Chromel (chromium-nickel alloy)-alumel (aluminium-nickel alloy) thermocouple was used to measure temperature in the range of -200°C to 1370°C , with an accuracy of $\pm 2.2^{\circ}\text{C}$ or 0.75% of the measurement. The temperatures at the compressor inlet and outlet, turbine inlet and outlet and exhaust were measured by using thermocouples installed on the MGT. All the measurement data were stored on disk via a laptop-controlled data-acquisition system CompactRIO-9072. Since the experimental setup was introduced above, the corresponding testing procedure now is given in the next section. The descriptions for the uncertainty and repeatability are also followed.

2.2 Experimental procedure

2.2.1 Turning on the MGT

(1) Connect the blower to the MGT air inlet using the rubber tube and connect the fuel pipe.

(2) Connect the lubricating oil pipe. The oil pump is used to fill the pipe with lubricating oil, ensuring that no air enters the lubricating oil pipe as air would plug the pipe and stop the pump. Then, connect the spark plug and ensure that the spark plug ignites.

(3) Place all measurement sensors for rotational speed, temperature, pressure, flow rate and power output, and connect these sensors to the data-acquisition system using low resistance line.

(4) Turn on the lubrication pump. Carefully measure temperature, and maintain temperature at $<700^{\circ}\text{C}$.

(5) When the MGT reaches 40000 r/min, the blower and rubber tube can be removed.

(6) Adjust the rotation rate from 40000 r/min to maximum, and measure power outputs (voltage and current), temperatures, pressures and flow rates.

2.2.2 End of the test

(1) Slowly turn off the flow rate control valve and lubrication pump.

(2) Connect the blower to the MGT air inlet via the rubber tube, and use the air that is sucked by the blower to cool the MGT down to 50°C .

(3) Turn off valves on fuel supply bottles.

2.3 Experimental uncertainty

The apparatuses must be corrected by other standard instruments to make sure that they can normally operate and let the inaccuracy of the experimental results be reduced to minimum. In this study, the major sensor in the experiment was the mass flow controller (MFC). The measurement range of the TC-1350 MFC adopted in this study was $0.6\text{--}100\text{ L/min} \pm 0.2\%$. The authors also used a different type of MFC, series TC-3100, which had wider measurement range as the standard correction apparatus to correct the TC-1350 MFC. All the uncertainties in different flow rates were between -0.05% and 0.12% .

2.4 Experimental repeatability

To verify experimental accuracy, perform one test using the specified mixed fuel at the specified pressure and flow rate; perform each test three times to ensure experimental repeatability. The following examples demonstrate experimental repeatability. The volume flow rates for four LHV fuels at different rotational speeds were chosen to demonstrate experimental repeatability. The evaluation included three measurements for volume flow rate; the average value for each test was used. Standard deviation was defined as the absolute difference among the three volume flow rates. Table 1 and Figure 5 show the coefficients of variation (CVs) and standard deviations. The CV is defined as the ratio of standard deviation s to mean $\bar{X}$, where s is derived by

$$s = \sqrt{\frac{1}{N} \sum_{i=1}^N (X_i - \bar{X})^2} . \quad (2.1)$$

As the experiments were conducted outdoors, it was difficult to control environmental conditions, for safety reason. As a consequence, the errors ($<3\%$) in these experiments are expected to be higher than general experiment errors, but they should be acceptable. Experimental repeatability is apparently very high (Table 1).

3 Mathematical model

In this paper, we did the simulation of steady and three-dimensional thermo-flow field inside the annular combustion chamber in advance. The regions of the simulation included combustion chamber, fuel pipe, air inlet, and air outlet. In order to make the problem more tractable, several assumptions were made as the flow was Newtonian fluid, the flow was steady, compressible, and

Table 1 Experimental repeatability for volume flow rate for various fuel

60% Methane						
r/min	1st (L/min)	2nd (L/min)	3rd (L/min)	average	s	CV(%)
40000	45.9382	45.6556	44.3682	45.32	0.68	1.51
42000	48.5758	49.5492	48.2461	48.79	0.55	1.13
45000	51.1349	52.752	54.5575	52.82	1.40	2.65
46000	53.4271	54.3377	55.421	54.40	0.82	1.50
47500	55.3582	55.6879	57.5562	56.20	0.97	1.72
70% Methane						
r/min	1st (L/min)	2nd (L/min)	3rd (L/min)	average	s	CV(%)
45000	42.0912	41.9728	40.8332	41.63	0.57	1.36
50000	46.8568	47.9964	44.8588	46.57	1.30	2.78
55000	51.874	51.2672	51.2228	51.45	0.30	0.57
60000	54.7452	51.4448	54.1828	53.46	1.44	2.70
80% Methane						
r/min	1st (L/min)	2nd (L/min)	3rd (L/min)	average	s	CV(%)
45000	35.0908	35.7124	35.3572	35.39	0.25	0.72
50000	38.8056	40.3004	39.9304	39.688	0.64	1.60
55000	43.1568	42.7276	44.2372	43.37	0.64	1.46
60000	46.4572	46.1612	47.4044	46.67	0.53	1.14
65000	51.9036	51.2376	51.3412	51.49	0.29	0.57
70000	55.87	53.8276	55.8848	55.19	0.97	1.75
90% Methane						
r/min	1st (L/min)	2nd (L/min)	3rd (L/min)	average	s	CV(%)
45000	31.5388	31.302	30.906	31.25	0.26	0.83
50000	35.3128	35.2536	34.1955	34.92	0.51	1.47
55000	39.6788	38.6724	37.3473	38.57	0.95	2.48
60000	43.3344	42.5796	40.545	42.15	1.18	2.79
65000	47.138	46.4276	44.4006	45.99	1.16	2.52
70000	49.9056	50.246	47.6748	49.28	1.14	2.31
75000	54.094	53.9608	51.7905	53.28	1.06	1.98
80000	57.5128	57.3944	55.5237	56.81	0.91	1.60
85000	58.904	58.3268	59.9607	59.06	0.68	1.15

turbulent, properties in solid were constant, and gaseous behaviors obeyed the ideal gas law. The corresponding simulated mass flow rate of fuel could be divided into four mass-fraction conditions, that is, the highest one was 90% of methane and the lowest was 60%. According to the usage of different fuel concentrations, the range of the related compressor speed was different, too.

3.1 Steady state governing equations

The flow in this research was three-dimensional and turbulent flow. The k - ε model, SIMPLEC algorithm as well as hybrid scheme were utilized to solve these problems. The major governing equations were shown as follows:

continuity equation:

$$\frac{\partial}{\partial x_j}(\rho u_j) = 0, \quad (3.1)$$

momentum equation:

$$\frac{\partial}{\partial x_j}(\rho u_j u_i - \tau_{ij}) = -\frac{\partial p}{\partial x_i}, \quad (3.2)$$

where suffixes i and j represented Cartesian coordinates ($i = 1, 2, 3$), u_i and u_j are the absolute fluid velocity components in x_i and x_j directions, ρ is density, t is time, τ_{ij} is stress tensor component and p is pressure.

Energy conservation. Heat transfer processes were computed by energy equation in the form known as the total enthalpy equation:

$$\begin{aligned} \frac{\partial}{\partial t}(\rho h_0) + \nabla \cdot (\rho \bar{V} h_0) = \nabla \cdot (k_{\text{eff}} \nabla T) + \frac{\partial P}{\partial t} \\ + \left[\frac{\partial(u\tau_{xx})}{\partial x} + \frac{\partial(u\tau_{yx})}{\partial y} + \frac{\partial(u\tau_{zx})}{\partial z} \right] \\ + \left[\frac{\partial(v\tau_{xy})}{\partial x} + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{zy})}{\partial z} \right] \\ + \left[\frac{\partial(w\tau_{xz})}{\partial x} + \frac{\partial(w\tau_{yz})}{\partial y} + \frac{\partial(w\tau_{zz})}{\partial z} \right] + S_h, \end{aligned} \quad (3.3)$$

where S_h are the energy sources, h_0 is the total enthalpy.

Species equation:

$$\frac{\partial}{\partial x_j}(\rho u_j Y_i) = \frac{\partial}{\partial x_j} \left(\rho D_i \frac{\partial Y_i}{\partial x_j} \right) + M_i \omega_i, \quad (3.4)$$

where Y_i represents mass fraction for mixture gas of CH_4 , H , CO , H_2 , O_2 , CO_2 , H_2O and inert gas N_2 . The term $M_i \omega_i$ is mass production rate of species. M_i is the molecular weight of the i -th species. ω is the reaction rate.

$$\omega = k_r Y_{\text{fuel}} Y_{\text{oxidizer}}, \quad (3.5)$$

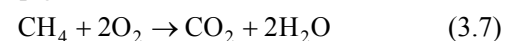
where k_r is the reaction rate constant that can be written in the Arrhenius form as

$$k = AT^n \exp(-E/RT), \quad (3.6)$$

where A is pre-exponential factor, n is temperature exponent, (E/R) is activation temperature.

Here one-step global reaction mechanism is adopted, the pre-exponential factor, temperature exponent and activation temperature are described as follows:

(1) One-step global reaction mechanism:



with the reaction rate expression as^[10]

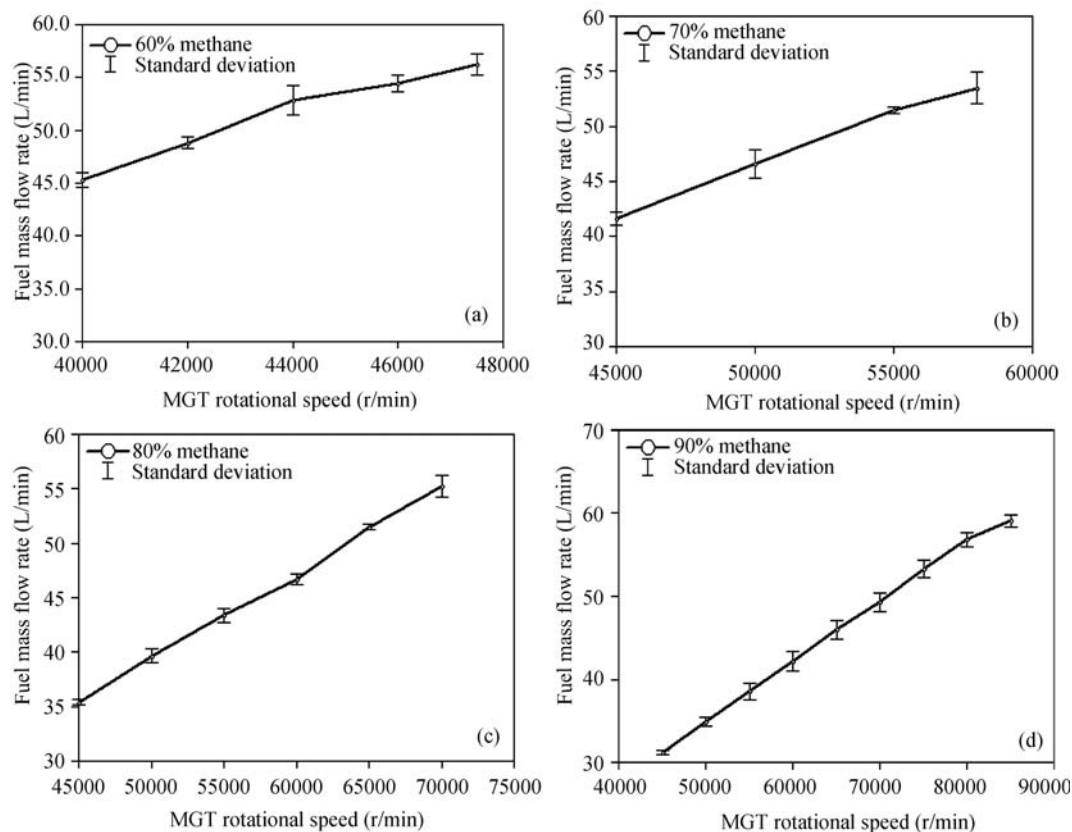


Figure 5 Experimental error bars for CH₄: CO₂ mixing ratios. (a) 60% CH₄ to 40% CO₂; (b) 70% CH₄ to 30% CO₂; (c) 80% CH₄ to 20% CO₂; (d) 90% CH₄ to 10% CO₂.

$$\frac{d[\text{CH}_4]}{dt} = -(2.5 \times 10^{12}) e^{-\left(\frac{24355}{T}\right)} [\text{CH}_4]^{0.2} [\text{O}_2]^{1.3}. \quad (3.8)$$

Turbulent model. The Reynolds averaged Navier-Stokes simulation adopts k - ε model which involves solutions of transport equations for turbulent kinetic energy and its rate of dissipation. The model adopted here is based on the research result of Launder and Spalding which was referred in the sample model of a micro gas turbine engine in CFD-ACE+ Manual^[10]. The transport equations for the standard k - ε model are as follows:

$$\begin{aligned} \frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho u_j k) \\ = \rho P - \rho \varepsilon + \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right], \end{aligned} \quad (3.9)$$

$$\begin{aligned} \frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho u_j \varepsilon) \\ = C_{\varepsilon 1} \frac{\rho P \varepsilon}{k} - C_{\varepsilon 2} \frac{\rho \varepsilon^2}{k} + \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right], \end{aligned} \quad (3.10)$$

where $C_\mu=0.09$; $C_{\varepsilon 1}=1.44$; $C_{\varepsilon 2}=1.92$; $\sigma_k=1.0$; $\sigma_\varepsilon=1.3$. Note that the standard k - ε model is a high Reynolds model and is not intended to be used in the near-wall regions where viscous effects dominate the effects of turbulence. Instead, standard wall functions are used in cells adjacent to walls.

3.2 Boundary conditions

The settings of boundary conditions are specified within the operating range of components that are confirmed by means of the size specification through the classically thermodynamic theory. The boundary values are corresponding to different operating conditions. In the model domain, it consists of two inlets, one outlet, two physical symmetric surfaces (including one cyclic symmetric surface), interfaces between two different phases (solid and gas), and the remaining wall. The non-slip boundary conditions are applied on the solid walls. The outlet locates at the rear of combustion chamber whose function is to guide the hot gas to generate thrust power. The outlet conditions of combustion chamber are mainly determined by what is happened upstream. The parameters,

such as temperature, velocity, and pressure, at outlet boundary are assumed zero gradients there, and the flow rate in outlet boundary must be satisfied with mass balance. The air and fuel inlet boundary conditions are summarized in Table 2.

3.3 Numerical method

In solving the Navier-Stoke Equation, the Semi-Implicit Pressure Linked Equation Correlation (SIMPLEC) algorithm was a widely used numerical method. Different from other algorithms adopting finite difference method, this SIMPLEC adopted finite volume method. Figure 6(a) shows the model of annular combustion chamber used in the CFD-ACE+. It was developed by Solid-

Works; then, imported to the CFD-ACE+ as IGS file. Afterwards, the mesh arrangement was accomplished and shown in Figure 6(b) by the CFD-ACE+ pre-processing, also called CFD-GEOM.

3.4 Grid-independence test

For obtaining the acceptable numerical solution, this research applied the unstructured grids produced from geometry models to carrying out grid-independence test. The grid-independence test comprised combustion case. Because the over-heating problem of liners is one of concerns in this research, the grid densities are increased especially near the liners, whose thicknesses are only 0.4 mm, and the fuel tube, which is about 0.5 mm in

Table 2 Boundary conditions of combustion chamber

Table 2 Boundary conditions of combustion chamber							
Air inlet				Fuel inlet			
Conditions	hydraulic diameter $(D_h)=0.012\text{ m}$ turbulence intensity $(I)=0.04$ $Y_{N_2}=0.768, Y_{O_2}=0.232$			60% methane	70% methane	80% methane	90% methane
				$D_h=0.01\text{ m}$	$D_h=0.01\text{ m}$	$D_h=0.01\text{ m}$	$D_h=0.01\text{ m}$
				$I=0.04$	$I=0.04$	$I=0.04$	$I=0.04$
				$Y_{CH_4}=0.353$	$Y_{CH_4}=0.459$	$Y_{CH_4}=0.593$	$Y_{CH_4}=0.766$
			$Y_{CO_2}=0.647$	$Y_{CO_2}=0.541$	$Y_{CO_2}=0.407$	$Y_{CO_2}=0.234$	
			$T=297\text{ K}$	$T=296\text{ K}$	$T=295\text{ K}$	$T=295\text{ K}$	
Rotational speed (r/min)	CR	$\dot{m}$ (kg/s)	T_2 (K)	Flow velocity of fuel (m/s)			
40000	1.050	0.001575	306.0	80.14	×	×	×
42000	1.063	0.001654	306.7	86.28	×	×	×
45000	1.081	0.001765	307.8	94.80	73.60	62.58	55.30
46000	1.088	0.001800	308.5	96.20	×	×	×
47500	1.100	0.001890	308.9	99.00	×	×	×
50000	1.140	0.002040	309.0	×	82.36	70.17	61.75
55000	1.145	0.002360	310.5	×	91.00	76.70	68.20
60000	1.182	0.002500	313.0	×	95.50	82.53	74.57
65000	1.209	0.002680	315.89	×	×	91.00	81.33
70000	1.245	0.003300	318.80	×	×	97.60	87.00
75000	1.294	0.004100	322.69	×	×	102.60	94.20
80000	1.355	0.004400	326.19	×	×	×	100.00
85000	1.400	0.004700	332.50	×	×	×	104.00

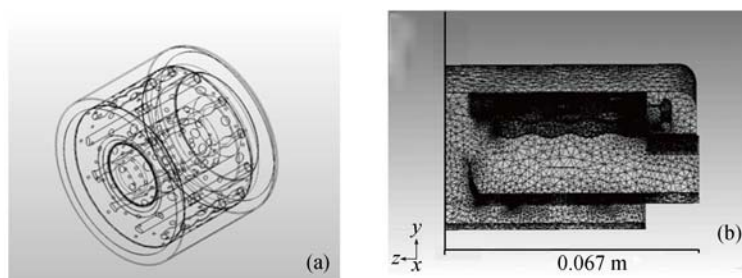


Figure 6 (a) Configurations of annular combustion chamber; (b) grids generation for numerical computation.

thickness. Owing to adoption of upwind numerical difference scheme, the grid amount should be dense enough to avoid the false diffusion phenomenon. For grid test, the operating conditions are specified as that the air inflow of $\dot{m}=0.0044$ kg/s is generated by compressor with a compression ratio (CR) 1.35 under the rotation rate 80000 r/min, and the fuel tube supplies a mixing fuel of methane (mass fraction 0.766) and CO₂ (mass fraction 0.234) under a fixed mass flow rate ($\dot{m}=3.325\times10^{-5}$ kg/s). From the above discussion, the mass flow rate, velocity and temperature of gas mixture ejected from the rear of combustion chamber are the emphasized data. The grid numbers adopted for the grid-independence tests in this case are 477181, 717974, and 1484378, respectively. The convergence criterion is selected as 10^{-3} , whose value is applied for all of simulations afterward. The test results are listed in Table 3. From the information given by the table, it can be seen that the maximum relative errors of various physical quantities are all less than 3%.

Under such circumstance, it is natural to select the grid number of 477181 to compromise the computational time and acceptable accuracy. PC of Pentium 4 with CPU 3.0 GHz, 2GB RAM is used to carry out the computation. The computational time for a typical simulation needs about 60 h.

4 Results and discussion

4.1 Numerical simulation results and discussions

Since a series of parametric studies were carried out, a reference case of steady state of methane concentration, 60%, at idle speed of MGT (45000 r/min) was chosen to serve as a detailed illustration case (Figures 7(a)—7(e)). Table 4 presents some important data of the simulation results. Figure 7(a) shows the CH₄ mass fraction distribution on the symmetric plane (0 degree cross-section) in combustor chamber for the case of inlet methane concentration 60% at 45000 r/min. The combustor chamber can be divided into three zones: primary, in-

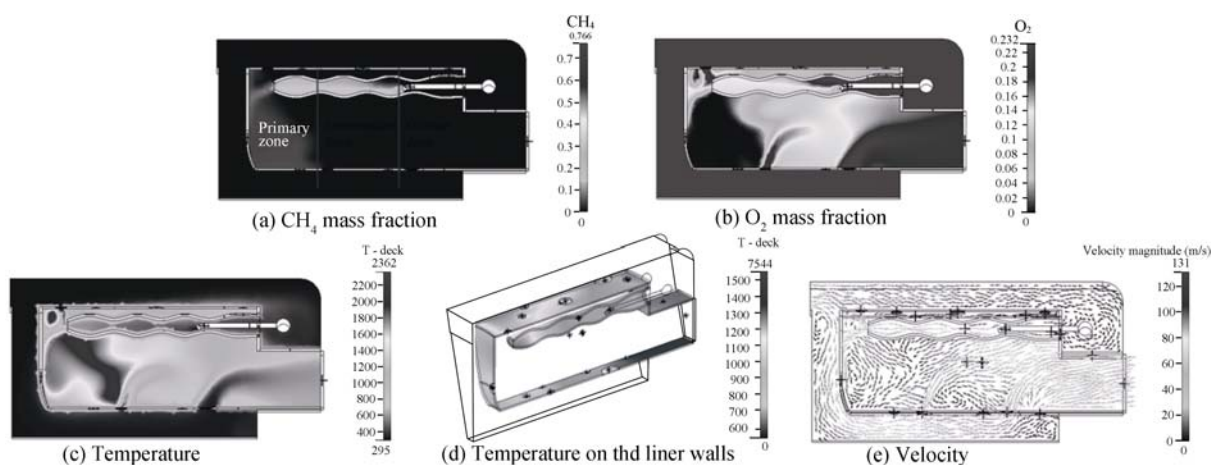
termediate, and dilution zones. They are marked according to the temperature distribution shown in Figure 7(c) at the same plane. The fuel mixture of CH₄ and CO₂ is injected into a mixing tube, where it mixes with the concurrent air flow from the back of combustor. Then, the premixed fuel/air is exhausted from the mixing tube and immediately met with the air supplied from the hole of liner above (see Figure 3) to enter the primary zone. In the meantime, air is also supplied via the hole on the bottom surface of liner. Since such hole in the primary zone is not located on the symmetric plane (see Figure 3), it cannot be observed in this figure. Because the air and fuel are mixed and burnt in the primary zone, the methane concentration ($Y_{CH_4}=0.6$) decreases gradually from fuel tube exit to primary zone. Eventually, the methane concentration is near zero in the intermediate zone due to the strong dilution by air. Figure 7(b) shows the O₂ mass fraction distribution at symmetric plane. There are 14 air inlets in the computational domain, which can be referred in Figure 3. Only four of them are located on the symmetric plane. In this figure, each zone has one air inlet, and the last one is behind the fuel tube. At these air discharge holes, their oxygen mass fractions are 0.232 because they come from fresh air. The intense combustion occurs in primary zone, the oxygen mass fraction is almost zero here. It increases gradually toward the intermediate zone and finally becomes 0.232 in the dilution zone since about 76% of total supply air entering the combustor flows into the two zones via the holes on the liner to dilute and cool the combustion products. Figure 7(c) is the temperature distribution. There is a high temperature area occurring in primary zone, indicating that the combustion is concentrated there. The maximum temperature (2190 K) does not exceed the adiabatic one, 2170 K, owing to the application of finite-rate chemical mechanism. Behind the combustion zone, the methane concentration in the intermediate zone is too low to sustain the combustion. In

Table 3 Grid test results of different grid densities for numerical simulation

	477181 grids	717974 grids	1484378 grids	Maximum relative error (%)
Outlet mass flow rates (kg/s)	4.18×10^{-3}	4.17×10^{-3}	4.25×10^{-3}	1.9
Outlet max. temperature (K)	645	641	655.2	2.2
Outlet max. velocity (z-direction) (m/s)	108.9	106	106.4	2.3
Average outlet velocity (z-direction) (m/s)	80.6	81.5	82.9	2.5

Table 4 Simulation results for different concentrations of methane

Rotational speed (r/min)	40000	42000	45000	46000	47500	50000	55000	60000	65000	70000	75000	80000	85000
60% Methane													
Max. temp. in combustor (K)	2176	2185	2190	2192	2198	×	×	×	×	×	×	×	×
Max. temp. on liners (K)	1231	1254	1261	1271	1276	×	×	×	×	×	×	×	×
Max. temp. at turbine inlet (K)	1161	1158	1108	1101	1079	×	×	×	×	×	×	×	×
70% Methane													
Max. temp. in combustor (K)	×	×	2227	×	×	2245	2255	2258	2245	×	×	×	×
Max. temp. on liners (K)	×	×	1304	×	×	1307	1350	1363	1307	×	×	×	×
Max. temp. at turbine inlet (K)	×	×	1055	×	×	1041	1025	1020	1041	×	×	×	×
80% Methane													
Max. temp. in combustor (K)	×	×	2251	×	×	2265	2270	2271	2276	2309	2337	2265	2270
Max. temp. on liners (K)	×	×	1371	×	×	1384	1396	1402	1406	1447	1481	1384	1396
Max. temp. at turbine inlet (K)	×	×	1046	×	×	1028	1019	1014	1011	1009	981.0	1028	1019
90% Methane													
Max. temp. in combustor (K)	×	×	2265	×	×	2281	2282	2304	2316	2320	2358	2362	2265
Max. temp. on liners (K)	×	×	1389	×	×	1406	1423	1440	1450	1470	1491	1544	1547
Max. temp. at turbine inlet (K)	×	×	1036	×	×	1023	1016	1010	999.7	960.5	955.2	948.5	934.0

**Figure 7** Simulation results of CFD analysis.

addition, the cooling air is continuously supplied into this zone via 5 holes. Thus, the temperature starts to drop. After that, all of the air which discharges into the dilution zone is used to further cool down the flow to exit. This purpose of cooling is to avoid the injury to the turbine blade by a high temperature stream from the combustor outlet. Because the turbine blade is made of inconel, whose allowable temperature is about 1336°C (=1609 K). From Table 4, the turbine inlet maximum temperature, 1108 K, is well below the allowable temperature, but exceeds the experimental measurement, 913 K, which will be presented in Figure 12 and discussed later. These results show that the dilution holes

are fully functional and the jet flows discharged from dilution holes can effectively cool down the exhaust gas temperature. In addition, the chemical mechanism used in this case is a reduction one (one-step) that the simplified chemical reactions are expected to be faster than the actual chemical reactions. Thus, the predicted temperature is higher than the actually measured value. Figure 7(d) demonstrates the temperature distribution on the liner walls in the computational domain. The temperature distributions on liner walls are affected not only by heat transfer of the combustion flow but also by the cooling air flowing between casing and liners. Therefore, the flowing air adjacent to the outer liner walls is heated

by heat conduction that can aid the cooling effect of liner wall. Of course, the air temperature rises continuously toward downstream. Therefore, the heated air mixed with fuel mixture can preheat the premixed mixture to enhance the combustion efficiency in primary zone.

The highest temperature zone as expected happens on the surface above in primary zone. From Table 4, the maximum temperature on liner walls is 1261 K. The maximum allowable temperature of liner wall, made of inconel, is about 1336°C (=1609 K). Therefore, the simulation result shows that the maximum temperature on liner walls is tolerable. Figure 7(e) is the velocity distribution of flow field in combustor chamber. It clearly demonstrates that there exists a recirculation in primary zone that can assist fuel mixing and hold the flame simultaneously. Another recirculation occurs in the area above the bottom liner wall between the two air inlets located at the primary and intermediate zones, respectively. It results from two different air flow velocities via the inlets. It also can be observed that the largest velocity occurs at the exit. The air flow velocities between the liner walls and casing are relatively smaller. The acceleration as air passes the high temperature area of liner (see Figures 7(d) and 7(e)) can be still observed. Also, the flow velocities at the air inlet are quite great because the area changes. The main purpose of the simulation work is to provide further thermo-fluid information inside the combustion chamber that cannot be obtained from the experiments discussed below. Also, the predictions can be compared with the available measurements to verify the creditability of the computational model to serve as a base for the future redesign of combustion chamber.

4.2 Experimental results and discussions

The original MGT, which was powered by liquid fuel, was modified to run on gaseous fuel. Therefore, the original pipes were changed accordingly. Additionally, the lubricant for bearings in the original MGT was pre-mixed with oil fuel. The supply system was separated from the fuel pipes after modification. In the experiments, stable rotational speed, 45000 r/min, was reached. To increase fuel efficiency, data were obtained at each step as the rotational speed was increased by 5000 r/min increments. Each step was maintained for approximately 10 s to ensure that the engine reached a stable condition such that the measured output data at

each step were meaningful; otherwise, stop-and-run experiments would consume a substantial amount of gas and the time required would be excessive.

4.2.1 Tests using various fuels with no loading. The proposed MGT was tested without loading. Various fuels with low LHVs with various mixture ratios of methane (CH₄) to carbon dioxide (CO₂) were used. The concentrations of CH₄ in the fuel were in the range of 50%–90%, and those of CO₂ increased from 10% to 50%. The temperatures of the main components of the MGT were measured at a specified fuel and its flow rate against cylinder gauge pressure. Thermal efficiency was calculated accordingly. Finally, the performances of the MGT under the various test conditions were compared.

4.2.1.1 Pressures and volume flow rates of fuel in the MGT. Fuel pressure was the pressure exerted by the cylinder (bottle); this pressure was adjusted using a control valve. Fuel pressure was determined by fuel volume flow rate. The volume flow rate of each fuel was measured as a function of pressure, and calculated from mass flow rate. (CH₄ density is 0.7168 g/L at 0°C, 1 atm). The volume flow rate of each fuel was roughly linearly proportional to fuel pressure. Figure 8 shows plot rotational speeds as functions of volume flow rates for the various fuels with LHV. In the test using fuel composed of 90% CH₄ and 10% CO₂, the rotational rate approached 85000 r/min as fuel pressure approached a maximum of 12 bar. Under the same pressure, a rate of only 47000 r/min was reached with 60% CH₄ and 40% CO₂. The condition of 50% methane could not generate power when loading was applied. More specifically, the power of MGT at 40000 r/min while applying 50% methane fuel was not able to drive the generator. The authors present the results of 50% methane here which were tending to offer more information for future research interest.

Clearly, the combustible fuel concentration in the low-heating-value fuels influenced the performance of MGT. According to Figure 8, as expected, the fuel with the higher heating value performs better because it can supply more energy to MGT. Also, a higher pressure must be applied to the lower heating-value fuel to approach idleness at 45000 r/min. Since MGT performance is proportional to the CH₄ concentration in fuel and volume flow rate, increasing volume flow rate can increase MGT performance. Furthermore, using a fuel with a LHV in the MGT originally designed for oil fuel may result in choking of both the turbine wheel and fuel supply pipe. This choking can occur in the nozzle throat

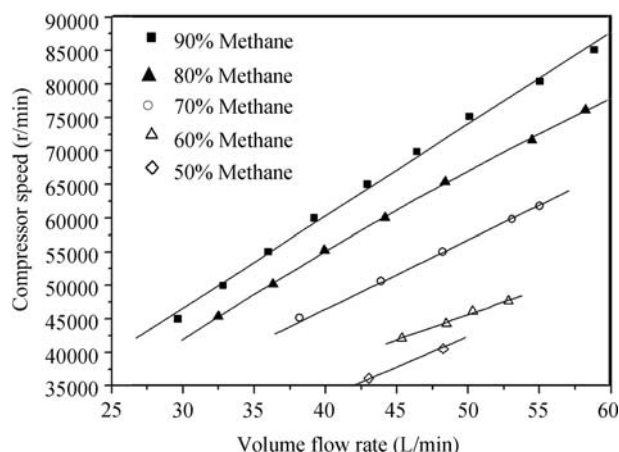
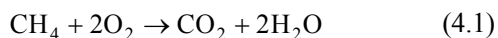


Figure 8 Volume flow rate for fuels with LHVs at various rotational speeds.

or in the annulus at the turbine outlet. As rotational speed increases to the one greater than 85000 r/min, choking may markedly limit the rotational rate of the turbine wheel because the rotational speed increases as flow rate increases; however, the maximum fuel pressure reached at 85000 r/min was 12 bar. Therefore, turbine choking did not occur during tests in this study. Compared to the turbine choking, the choking at fuel supply pipe needs more attention. The original pipe whose diameter was only 0.5 mm was designed to deliver 0.05 L liquid fuel per minute at 80000 r/min. However, the experiment results showed that the volume of the fuel was 58 L and the pressure was 12 bar when the maximum power was reached. Thus, in order to avoid choking condition, the fuel supply system should be modified to deliver more volume of fuel so that the pressure will be reduced and the security will be enhanced.

In this investigation, the equivalent ratio is given by



Air: Fuel (CH_4) = 18.125:1.

The air mass flow rate of 321.12 g/min in the primary zone was determined from the compressor map at 45000 r/min; the fuel mass flow rate was 26.87 g/min. The Air/Fuel ratio (A/F) was 16.7 less than the equivalent ratio of 18.1; however, at 80000 r/min, the A/F was 22.5, which is greater than the equivalent ratio of 18.1. Therefore, the A/F increases as rotational speed increases from 45000 to 80000 r/min. Since the volume flow rate is proportional to cylinder pressure, the volume flow rate of fuel is given by the following linear regression equation (CH_4 density

is 0.7168 g/L at 0°C, 1 atm):

$$\dot{m} = 5.54 + 4.2P_{\text{out}} \quad (4.2)$$

The volume flow rates of the fuel with a LHV were then replaced by those of CH_4 . Figure 9 presents the corresponding results. At a particular compressor rotational speed, the volume flow rates of CH_4 obtained for different fuels with LHVs were almost equal, indicating that combustion was stable in the proposed MGT, even when different fuels were used.

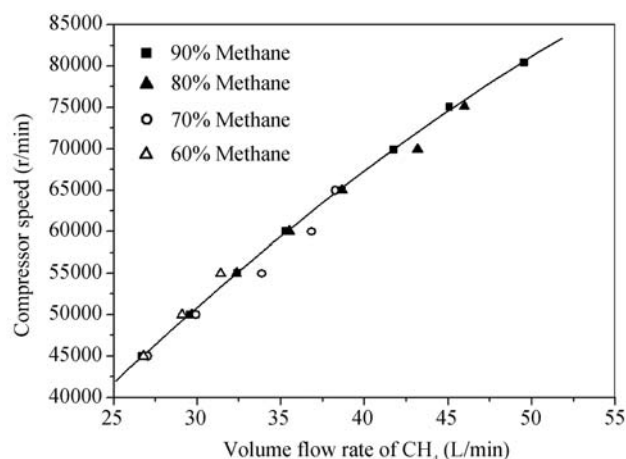


Figure 9 Compressor speed vs. volume flow rate of CH_4 with different concentrations of methane fuels.

4.2.1.2 Temperature of unloaded MGT. Temperatures were measured at the compressor inlet (T_1), compressor outlet (T_2), turbine inlet (T_3) and turbine outlet (T_4). Figures 10(a) and 10(b) plot the histories of all temperatures and the corresponding rotational speeds for two fuels, 90 CH_4 and 60% CH_4 . The inlet and outlet temperatures (T_3 and T_4) of the turbine decline as rotational speed increases because as compressor's rotational speed increases, an increased amount of air enters the combustion chamber, and the turbine is cooled by this additional air. Liou and Leong^[11], who experimented with a particular MGT using oil fuel, observed the same phenomenon at 40000–120000 r/min. However, when this MGT operated at greater than 120000 r/min, temperature increased as the rotational speed increased. T_3 and T_4 temperatures varied slightly as the concentration of CH_4 in fuels varied (Figures 10(a) and 10(b)). This trend could influence thermal efficiency, which is discussed in Section 4.2.3. Additionally, the difference between T_3 and T_4 varied minimally with the fuel with a LHV, because, at a given compressor rotational speed,

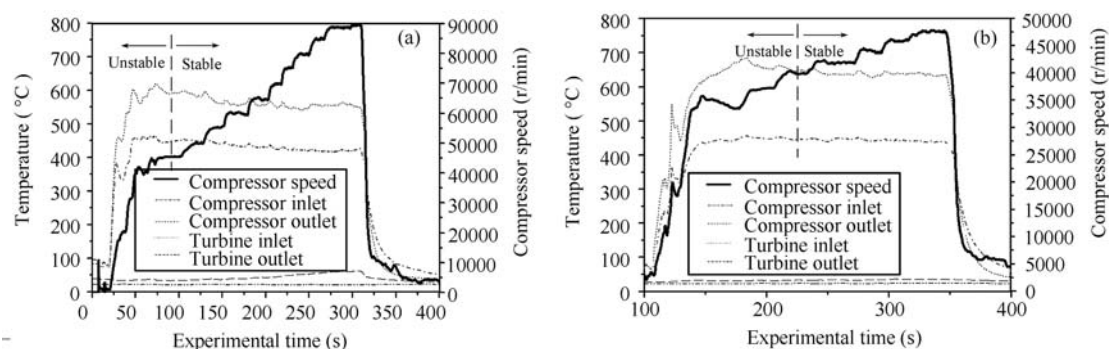


Figure 10 (a) Temperatures at different positions in the MGT and MGT rotational speed with 90% CH₄; (b) temperature measurements at different positions in the MGT and MGT rotational speed with 60% CH₄.

the amounts of CH₄ burned were the same for all fuels (Figure 10). T_3 and T_4 were markedly perturbed at the start of operation because the blower and fuel were manually controlled before the MGT reached stability. Thus, a dash line was added in Figure 10 to separate an unstable duration of the MGT from the stable one. Excessively increasing the air flow rate markedly reduced T_3 and T_4 . However, an excessive fuel flow rate rapidly increased T_3 and T_4 .

Figure 11(a) presents the surface (case) conditions of a combustion chamber using oil fuel; Figure 11(b) presents those of a combustion chamber using gaseous fuel. In Figure 11(b), the surface condition (color changed) on both the primary and intermediate zones of combustion chamber have been affected by high temperature while applying the LHV fuel, whereas such phenomenon did not appear when using liquid fuel. Clearly, combustion chamber temperature with methane exceeds that with liquid fuel because the combustion chamber is designed to burn liquid fuel. When gaseous fuel was used, the cooling system was inadequate. Moreover, this experimental conclusion can also be deduced that it is easier to burn methane than oil fuel. Based on the schematic view of the MGT system (Figure 3), the fuel pipe in the combustion chamber is close to the liner wall. Thus, the gas fuel should be burnt immediately after it is ejected from the fuel pipe and the temperature near the liner wall should be higher than that using oil fuel. This phenomenon has been predicted from the numerical results of temperature distribution in the combustion chamber (see Figure 7(c)). Since the distributions of CH₄ mass fraction, O₂ mass fraction and velocity in the combustion chamber were unavailable due to the deficiency of experiment facilities in this study, temperature data were

the only information that could be used to make comparison between the numerical predictions and experiment measurements.

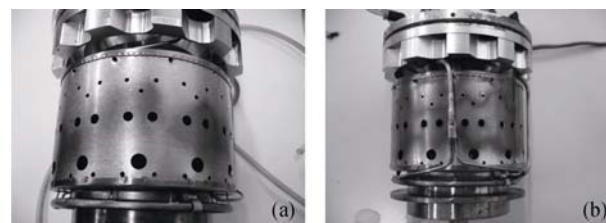


Figure 11 (a) Combustion chamber (oil); (b) combustion chamber using 70% CH₄ (gas).

Figures 12(a)–12(d) demonstrate the results of maximum temperature at turbine inlet (T_3) by using the global one-step simulation and the experimental data at different methane concentrations. From these figures, the predicted results of one-step mechanisms are more superior to the experimental measurements. In addition, the simplified chemical reaction used in this study is the reduced one. Thus, the one-step reaction mechanism over the predicted maximum temperature in combustor outlet with the experimental data is expected. This over prediction could be mended when four-step reaction mechanism is considered in future research^[12].

4.2.1.3 Thermal efficiency. The thermal efficiency of the proposed MGT system combined with the adiabatic efficiencies of the compressor and turbine were estimated thermodynamically by assuming a typical and simple Brayton cycle. The reason for selecting this simple Brayton cycle is that this equation is suitable for a single-shaft and low CR gas turbine that is suggested by Sonntag et al.^[13]. The efficiency of the air-standard Brayton cycle is

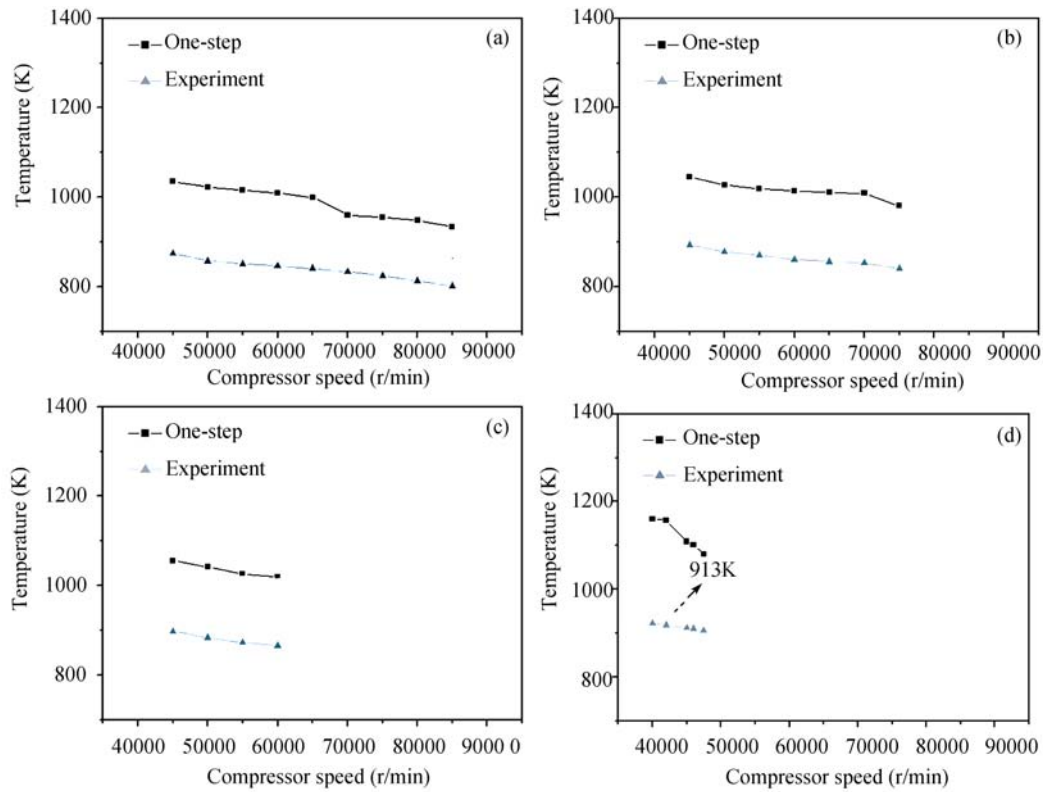


Figure 12 Maximum temperature at combustor outlet for different fuels. (a) 90% CH₄ to 10% CO₂; (b) 80% CH₄ to 20% CO₂; (c) 70% CH₄ to 30% CO₂; (d) 60% CH₄ to 40% CO₂.

$$\eta_{th} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{T_1 \left(\frac{T_4}{T_1} - 1 \right)}{T_2 \left(\frac{T_3}{T_2} - 1 \right)}. \quad (4.3)$$

According to Sonntag et al.^[13], because the working fluid does not complete a thermodynamic cycle in the MGT, the internal combustion engine operates on a so-called open cycle. However, when analyzing internal-combustion engines, devising closed cycles that closely approximate open cycle is advantageous. The thermal efficiency of the proposed MGT is derived from the efficiency of the air-standard Brayton cycle in eq. (4.3). Figure 13 shows thermal efficiency versus compressor rotational speed. For the fuels with 90%, 80% and 70% CH₄, efficiency decreases as rotational speed increases. Based on eq. (4.3), the thermal efficiency is affected by T_4 , T_3 , and T_2 ; that is, thermal efficiency decreases because the decrease in T_2 is larger than the decrease in T_3 and T_4 at high rotational speeds. Fuels with LHVs have more thermal efficiency than those with high heating values (Figure 13). Theoretically, thermal efficiency should not increase with fuels with LHV because the pressures and flow rates of fuels with LHVs fuels are

larger than those of fuels with high heating values, and the mismatch between the combustion chamber and fuel eject velocity causes the flame to approach the T_3 thermal couple. Hence, T_3 increased. This phenomenon also can be verified by the simulation result of this study and the conclusion of Kumar's research^[14] that the combustion zone would move linearly towards the combustor outlet with an increase of mixture flow rate.

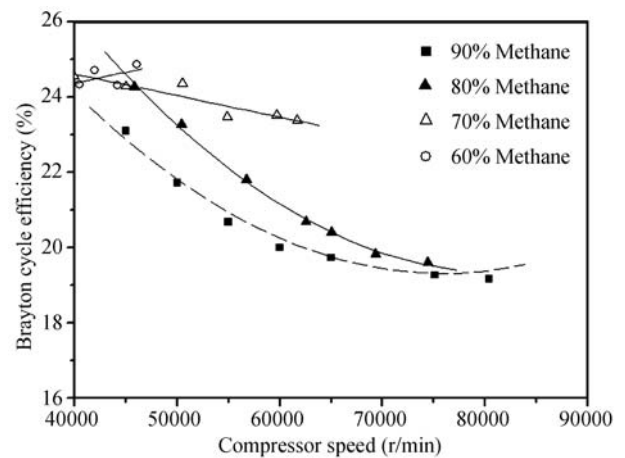


Figure 13 Brayton cycle efficiency of different compressor speeds with different concentrations of CH₄.

4.2.2 Testing the MGT with a generator. In this section, the MGT with a generator was tested; all the test parameters are the same as those used in Section 4.2. Temperature was measured and the amount of electricity generated by the generator was discussed. All the temperatures measured in the MGT with a generator are similar to those of the unloaded MGT. Notably, the MGT does not directly drive the generator; the generator is driven by a gearbox, which has a power turbine driven by high-temperature and high-pressure gas from the MGT. The generator weakly influences MGT performance. Choosing a generator that fits the MGT is extremely important. If the generator power is too high, loading will be excessive, and the MGT cannot then be operated in a suitable range of rotational speed. If the power generated is too low, the load will be too low and optimal performance cannot be obtained. The worst outcome is that the gearbox of the MGT system breaks when the rotational speed limit is exceeded. The power output obtained using 90% CH₄ fuel is roughly 170 W, which is the maximum output of the generator, and 70 W at 60000 r/min as 70% CH₄ with 30% CO₂ is used (Figure 14). When a critical limit of 60% CH₄ is used, the power output is extremely low. Additionally, the electric efficiency, calculated from the power output divided by the fuel release heat due to combustion shown in eq. (4.4), is around 10% by assuming a reversible, adiabatic, steady-state process.

$$\eta_{\text{electric}} = \frac{W}{\sum \Delta h_i m_i}, \quad (4.4)$$

where η_{electric} is the electric efficiency, W is the actual work of system, h_i is enthalpy to different species and m_i is mass flow rate to different species. This considerable

difference is due to the losses from the connected transmission gearbox and the bridge rectifier.

5 Conclusions

In this study, the effects of fuels with LHV on the performance of an annular MGT power system were assessed numerically and experimentally. The proposed system in this study has the potential to reduce the size of a distributed power supply system for biogas and enhance its popularization. The numerical analyses show that the cool air flows via dilution holes on liners were fully functioned and chamber design did not cause hot spots on the chamber liner or make the temperature of exhaust gas exceed the maximum allowable that temperature of the turbine wheels can tolerate. In the experiments, the proposed MGT system was operated successfully under each test condition; minimum composition to the fuel with the LHV was roughly 50% CH₄ with 50% CO₂. The power output was around 170 W at 85000 r/min as 90% CH₄ with 10% CO₂ was used and 70 W at 60000 r/min as 70% CH₄ with 30% CO₂ was used. When a critical limit of 60% CH₄ was reached, the power output was extremely low. When the experiment result was compared to the numerical results, the trade of maximum temperature at turbine inlet showed great coincidence. Furthermore, the theoretical Brayton cycle efficiency and electric efficiency of the MGT were calculated as 23% and 10%, respectively. The electric efficiency might be improved by replacing the generator as suggested in the next section. The completed micro bio-gas power supply system is small, low cost, easy to maintain and suited to household use.

6 Future Work

With a reduction in the fuel pipe length in the combustion chamber, fuel can be ejected into the combustor earlier than what the current design does. Thus, the flame might keep a distance from the front wall of the combustion chamber to reduce its thermal impact on the liner wall. Also to reduce fuel supply pressure and enhance safety, fuel pipe diameter of the current annular MGT system can be enlarged to deliver additional volume of fuel. Moreover, the data-acquisition facilities of the MGT system can be enhanced to determine gas exhaust components. The CO concentration profile in exhaust gas can help researchers determine whether complete combustion is achieved. Furthermore, in some fields, a brushless DC motor can be utilized as a genera-

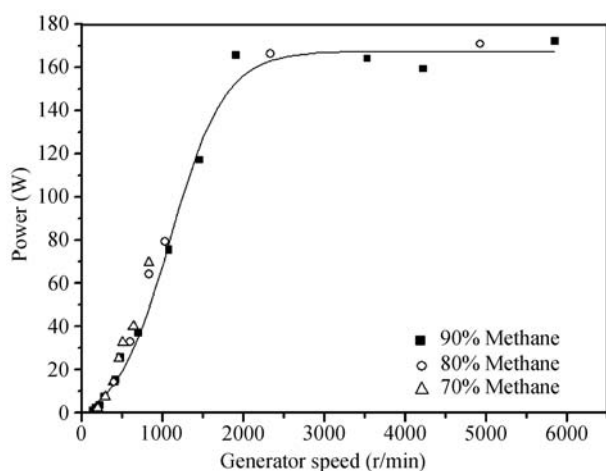


Figure 14 Output power for different generator rotational speeds with different LHV fuels.

tor and actuator simultaneously. Thus, if such a generator is applied to the proposed MGT, power loss is avoidable

compared to that when transmissions are used to connect the power turbine and generator.

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