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Analysis of the turbine blade number towards the micro gas turbines performance

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Abstract. Micro gas turbines can be an alternative power plant that can be utilized in rural areas. The micro gas turbine performance utilizes the LPG combustion reaction to produce the high of velocity and pressure of fluid so that the condition is able to drive the alternator for producing electrical energy. This research was conducted to improve the micro gas turbine performance by changing the turbine blades number. The research method is carried out in two steps, among others calculating the turbine blade design and simulating the performance of micro gas turbines with changes in the turbine blades number by *Computational Fluid Dynamics* (CFD). Parameters for calculating turbine blade design among others are fuel characteristics, fluid pressure, fluid temperature, and fluid heat energy. These parameters affect the performance of micro gas turbines, which are shown by the turbine rotating speed and turbine power. The results showed that the turbine blades number as many as 18 pieces were able to produce a fluid velocity of 135.4 m/s and a fluid pressure of 3.21 bar. This condition can produce a turbine rotational speed of 331.26 rpm and turbine power of 139.15 kW. Based on the research results can be concluded that increasing the turbine blades number can influence the micro gas turbines performance.

1. Introduction

Electrical energy is a primary human requirement for socio-economic activities consisting of industrial activities, commercial activities, and daily routine activities. Electricity needs are increasing every year, where its demand reaches 2534 billion kWh in 1992 and it can increase to 9792 billion kWh by 2023 [1]. Generally, urban areas need the most electrical energy compared towards the rural areas, for example, the DKI Jakarta region has an electricity demand of 99.98% while the Papua region has only 47.78% of electricity demand. This condition can be seen in Figure 1.

Figure 1. Electrification rates in Indonesian provinces [2]

The development of power plant technology is very important in rural areas. The aim of developing this technology is to equalize the use of electrical energy in accordance with urban areas. The technology of micro-scale power generation is an alternative technology that utilizes renewable energy sources as the main material for the process of electricity production. Generally, renewable energy sources in rural areas have been converted into conventional energy sources, among others are LPG (60%), biogas (60%), charcoal (30%), and dung (12%) [3]. One of the micro-scale power plants is a micro gas turbine which has several advantages among others are easy installation and operation, high power ratio, low pollution, and easy to combine with cooling and heating systems [4], [5]. Micro gas turbines have a working principle based on the Brayton cycle by utilizing high pressure and temperature from the results of fuel combustion process [6]. Micro gas turbine have five main parts i.e. compressor, turbine, combustion chamber, recuperator, and generator / alternator, where it can be seen in Figure 2.
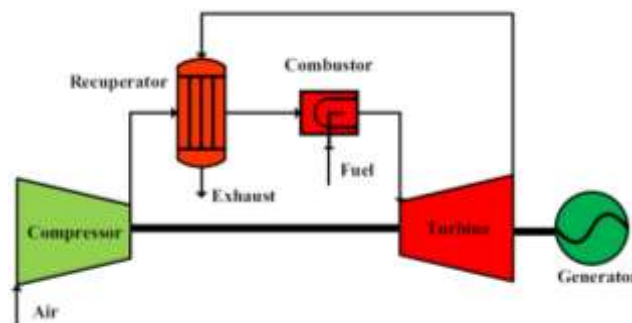

Figure 2. Micro gas turbine schematic [7]

Compressor, combustion chambers and turbines are an important part of the micro gas turbine system. The compressor can increase fluid pressure between 3 to 5 times [7]. The combustion chamber can determine the conditions of temperature, pressure and velocity from fluid flow and the turbine can determine the conditions of turbine rotation speed and turbine power which is useful for driving the generator in electricity production [8].

Some previous research regarding the development of micro gas turbine designs were able to improve the micro gas turbines performance to generate electricity on the alternator which is a micro gas turbine system part. Campanari [9] combines the heating and cooling systems with a micro gas turbine to increase fluid heat energy. Caresane [10] uses a cogeneration systems by utilizing the residual heat from the fuel combustion process in the combustion chamber to increase the temperature and fluid heat energy. Buffi [11] utilizes the bio-oil produced by the fast pyrolysis process as a fuel for micro gas turbine system to increase the turbine output power by 20 kW. Bruno [12] uses a desalination plant that is applied to a micro gas turbine system to produce heat energy. Previous research has a difficult treatment system when applied in rural areas. The micro gas turbine systems by changing the blade design is a solution to the difficulties of previous system maintenance management and it also improves the micro gas turbine performance. However, previous results have not been optimal to improve the micro gas turbines performance. Therefore, this research develops the micro gas turbine based on the rural areas conditions, by making changes in the working turbine system that focuses on turbine blades number demand. The micro gas turbine systems uses LPG fuel as the most energy source in rural areas. This research aims to increase the turbine outlet power and turbine rotation speed caused by high fluid velocity and pressure with the addition of blades number. The parameters that influence the turbine blade design are the thermodynamic conditions of the micro gas turbine system (based on the Brayton Cycle) which consists of the conditions of fluid temperature, fluid pressure, and fluid heat energy. The ideal fluid condition is obtained from the perfect fuel combustion process so that the fuel characteristics also affect that process. The research method uses two steps among others are turbine blade design calculation, and micro gas turbine system simulation based on changes in the blades number.

2. Methods

The research method focused on developing the number of blade turbines is divided into two steps, among others are turbine blade design calculation and blade turbine design simulation. Turbine blade design parameters use fluid temperature, fluid pressure, fluid heat energy, and fuel characteristics. The research results are the velocity and pressure of the fluid, which both of them influence the turbine rotational speed and turbine outlet power. The results would be compared with the results of micro gas turbines performance in ideal conditions.

2.1. Blade turbine design calculation

The micro gas turbine blade design that focuses on increasing the turbine blades number using the actual Brayton cycle calculation. The actual Brayton cycle (see Figure 3) has some cycle parts with different conditions among others, temperature, pressure and heat energy.
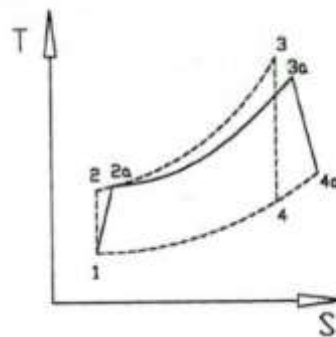

Figure 3. Actual and ideal Brayton cycles [13]

The actual calculation of the Brayton cycle requires the specification of the turbocharger as part of the micro gas turbine system to determine the fluid outlet pressure magnitude that affects the condition of adding a blades number. The turbocharger used has the CT16 type (see Figure 4). Turbocharger specifications can be seen in Table 1.

Figure 4. CT 16 Turbocharger type

Table 1. CT16 Turbocharger Specifications

No.	Parameters	Specifications
1	Temperature resistance	950 ^{0}C
2	Fluid mass flow rate	0.0233 – 0.18 m^3/s
3	Turbocharger rotational speed	3600 rpm
4	Pressure	16 psi *boost power*

Some parameters in the actual Brayton cycle (see Figure 3) are also calculated among others, temperature, pressure, heat energy, work, efficiency and *back work ratio* that occur in micro gas turbine systems. Calculation of some of these parameters can be shown in Equation 1 through Equation 15 [14].

$$T_1, P_{1compressor} = valid \tag{1}$$

$$\eta_T = \frac{\ln\left[1-\eta_k + \eta_k(\frac{P_2}{P_1})^{\frac{k-1}{k}}\right]}{\left(\frac{k-1}{k}\right)\ln\left(\frac{P_1}{P_2}\right)} \tag{9}$$

$$T_{2compressor} = valid \tag{2}$$

$$W_{k-actual} = \frac{W_k}{\eta_k} \tag{10}$$

$$P_2 = (r_p)P_1 \tag{3}$$

$$W_{t-actual} = \eta_T \cdot W_{Tideal} \tag{11}$$

$$T_{3turbine} = valid \tag{4}$$

$$W_{net} = W_{t-actual} - W_{k-actual} \tag{12}$$

$$P_{3a(combustionchamber)} = P_2(1-\Delta P_{rb}) \tag{5}$$

$$bwr = \frac{W_{k-actual}}{W_{t-actual}} \tag{13}$$

$$T_{4turbine} = T_3\left(\frac{1}{r_p}\right)^{\frac{k-2}{2}} \tag{6}$$

$$q_{in} = (h_{3a} - h_{2a}) \tag{14}$$

$$T_{4a} = valid \tag{7}$$

$$\eta_{th} = \frac{W_{net}}{q_{in}} \tag{15}$$

$$P_{4turbine} = P_{1compressor} \tag{8}$$

The micro gas turbine utilizes combustion gases as the main material for the combustion reaction process in the combustion chamber. The combustion reaction between combustion gases and airflow has the aim to produce high temperatures and pressures so that it can be used as an alternator drive that has a function to generate electricity. This research utilizes LPG (Liquid Petroleum Gases) as the main material of the combustion reaction, due to the LPG has the advantage on the Enhanced Oil Recovery (EOR) process. LPG has a composition of propane and butane gas with the percentage of 40% and 60%, respectively [15]. Comparison between LPG and airflow is needed to get a perfect combustion reaction, where the calculation can be seen in Equation 16 until Equation 20 [14].

$$(FAR)_{actual} = \frac{q_{in}}{LHV} \tag{16}$$

$$(AFR)_{actual} = \frac{1}{FAR} \tag{17}$$

$$m_{air} = valid \tag{18}$$

$$m_{fuel} = (FAR)_{actual} \cdot m_{air} \tag{19}$$

$$W_{turbine} = (m_{air} + m_{fuel}) \cdot C_p(T_3 - T_4) \tag{20}$$

The design of micro gas turbine blades is focused on the blades number. Temperature, pressure, heat energy conditions in micro gas turbine systems and LPG characteristics are the basic parameters in designing the blades number. Calculation of the blades number design can be seen in Equation 21 to Equation 24 [8].

$$L = \frac{144 \cdot q \cdot \left(\frac{862}{D}\right) \cdot h_3}{862 \cdot C \cdot k \cdot (2 \cdot g)^{1/2} \cdot h_3} \tag{21}$$

$$n = \frac{862}{D} H^{1/2} \tag{22}$$

$$t = \frac{k \cdot D}{\sin \beta_1} \tag{23}$$

$$n_s = \frac{\pi \cdot D}{t} \tag{24}$$

The calculations in Equation 21 until Equation 24 explain the turbine housing design, turbine rotational speed, and the turbine blade characteristics, i.e. diameter, the pitch of blades and blades number. The results of turbine blade design would be simulated with the aim of measuring the micro gas turbine performance with the addition of blades number.

2.2. Blade turbine design simulation

The research simulation method uses *Computational Fluid Dynamics* (CFD) by ANSYS Fluent solver. ANSYS Fluent has advantages among others, the simulation process results are more accurate based on the actual data parameters, and have a small deviation level. The ANSYS Fluent simulation basic is derived from the *Reynolds Average Navier-Stokes* (RANS) equation, where its equation is divided into two equations, namely the continuity equation and the momentum equation which can be shown in Equation 25 and Equation 26 [16].

$$\frac{\partial}{\partial x_i}(\overline{u}_i + u_i) = 0 \tag{25}$$

$$\frac{\partial \overline{u}_i}{\partial t} + \overline{u}_j \frac{\partial \overline{u}_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \overline{p}}{\partial x_i} + \frac{\mu}{\rho} \frac{\partial^2 \overline{u}_i}{\partial x_i \partial x_j} - \frac{\partial}{\partial x_j} \overline{u}_i \overline{u}_j \tag{26}$$

The simulation method uses turbulent fluid flow conditions in the *k-ε equation* which is shown in Equation 27 [17], [18]

$$\rho \frac{Dk}{Dt} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right) \frac{\partial k}{\partial x_i}\right] + G_k + G_b - \rho\varepsilon - Y_m \tag{27}$$

The rate of dissipation of fluid flow as a form of transportation equation can be shown in Equation 28 [17], [18].

$$\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right) \frac{\partial \varepsilon}{\partial x_i}\right] + C_{1\varepsilon} \frac{\varepsilon}{k}(G_k + C_{3e}G_b) - C_{2\varepsilon}\rho \frac{\varepsilon^2}{k} \tag{28}$$

Micro gas turbine performance simulations also use some parameter conditions to increase its performance results, where these parameters can be presented in Table 2.

Table 2. Performance simulation parameters of micro gas turbines

No.	Parameters	Quantity
1	Solver Equation	*Dencity Based*
2	Material	*Air* (*compressible condition*)

No.	Parameters	Quantity
3	Turbocharger rotational speed	3600 rpm
4	Fluid mass flow rate	0.032 kg/s
5	Fluid temperature	1223 K
6	Fluid pressure inlet	2.7 bar

The simulation results of micro gas turbine performance are the velocity and pressure of fluid flow after passing the micro gas turbine that compared toward the calculated data results. Simulation results affect the turbine rotational speed and turbine outlet power. The results of these two parameters are expected to have a high amount because it can help the performance of the alternator as a power plant.

3. Results and Discussion

The micro gas turbine system utilizes the results of LPG combustion reaction as the most energy source in the rural areas and a turbocharger system i.e. fluid flow with the high velocity and pressure to control the alternator performance as a power plant. Alternator electric products can be an alternative electrical energy that can be applied in daily activities, especially in rural areas. The turbine blade design is the basic parameter for determining the micro gas turbine performance.

3.1. Result of blade turbine calculation

This research analyzes the blades number as one of the factors determining the optimal turbine blade design. The addition of blades number makes the blade surface area increase so as to optimum the reception of fluid flow resulting from the LPG combustion process in the combustion chamber. Calculation of blades number must also know the condition of turbine house, so that the turbine blade can be applied. The results of geometry blade design calculations focused on the blades number can be seen in Table 3.

Table 3. Results of blade turbine design

No.	Geometry	Quantity
1.	Turbine housing diameter	74 mm
2.	Turbine runner height	29.41 mm
3.	Turbine runner diameter	69.62 mm
4.	Turbine rotational speed	331.26 rpm
5.	Pitch of blades	12.1 mm
6.	Blades number	18 pieces

Based on Table 3 can be explained that the addition of blades numbers not affect the turbine house condition and the blade turbine geometry (diameter and height). The close pitch position in the around of blade turbine can be able to increase the fluid flow pressure that moves out of the turbine. Its parameters is the basic for calculating the actual Brayton cycle in the micro gas turbine performance system, where the calculation results can be seen in Table 4.

Table 4. Results of micro gas turbine system

Parameters	Ideal Cycle	Previous Research Results	Research Results
		9 Blades	18 Blades
Compressor Inlet			
Temperature Inlet (T_1)	309 K	309 K	309 K
Enthalpy Inlet (h_1)	309.23 kJ/kg	309.23 kJ/kg	309.23 kJ/kg
Pressure Inlet (P_1)	1.0132 bar	1.0132 bar	1.0132 bar

Parameters	Ideal Cycle	Previous Research Results	Research Results
		9 Blades	18 Blades
Compressor Outlet			
Temperature Outlet (T_2)	322 K	321 K	322 K
Enthalpy Outlet (h_1)	318.28 kJ/kg	321.29 kJ/kg	318,28 kJ/kg
Pressure Outlet (P_1)	2.8351 bar	2.43 bar	2.73 bar
Turbine Inlet			
Temperature Inlet (T_3)	1223 K	1223 K	1223 K
Enthalpy Inlet (h_3)	1304,85 kJ/kg	1304,85 kJ/kg	1304,85 kJ/kg
Heat Energy Inlet	392,5 kJ/kg	392,5 kJ/kg	392,5 kJ/kg
Turbine Outlet			
Temperature Outlet (T_4)	920.83 K	703 K	703 K
Enthalpy Outlet (h_4)	948.45 kJ/kg	1304.85 kJ/kg	1304.85 kJ/kg
Pressure Outlet (P_4)	1.0132 bar	1.0111 bar	1.0132 bar
Compressor Efficiency	98%	85.80%	99%
Turbine Efficiency	98%	85.80%	99%
Actual Work of Compressor	9.2 kJ/kg	14.1 kJ/kg	9.2 kJ/kg
Actual Work of Turbine	352.83 kJ/kg	141.8 kJ/kg	352.83 kJ/kg
Actual Pressure of Combustion Chamber	2.68 bar	2.06 bar	2.68 bar
Actual Temperature of Turbine Inlet	933.121 K	-	-
Actual Enthalpy of Turbine Inlet	952.02 kJ/kg	-	-
Actual Heat of Turbine Inlet	986.57 kJ/kg	392.5 kJ/kg	986.57 kJ/kg
Actual Heat of Turbine Outlet	925.83 kJ/kg	359.1 kJ/kg	953.17 kJ/kg
Work Net (W_{net})	343.63 kJ/kg	127.7 kJ/kg	343.63 kJ/kg
Back Work Ratio	2.6 %	9.9 %	2.6 %
Ideal Cycle Efficiency	34.8 %	32.3 %	34.8 %
Mass flow rate of fluid		0.1478 kg/s	
Mass flow rate of fuel LPG Fuel		0.0313 kg/s	
FAR_{actual}		0.02117	
AFR_{actual}		47.24	
Power outlet of turbine		139.15 W	

Based on Table 4 can be explained that the blade numbers parameter can increase fluid flow pressure in the turbine outlet. The rise of fluid flow pressure influences the alternator performance to produce electrical energy. In addition, the fluid flow pressure produces high heat energy, where the half portion of heat energy can be used to support the fuel combustion process in the combustion chamber. This condition is also supported by a significant decrease in the turbine outlet temperature. This results of the micro gas turbine system after the addition of blade number approach the micro gas turbine system in the ideal Brayton cycle conditions. The results of turbine blade design calculation would be simulated by Computational Fluid Dynamics (CFD).

3.2. Result of Computational Fluid Dynamics (CFD) simulation

The results of blade numbers design calculation can be explained that an increase in the blade numbers can influence the micro gas turbine performance. This condition is simulated by Computational Fluid Dynamics (CFD) simulation to measure and analyze the fluid velocity and pressure condition. The simulation conditions of fluid velocity and pressure from micro gas turbine systems can be seen in Figure 5.
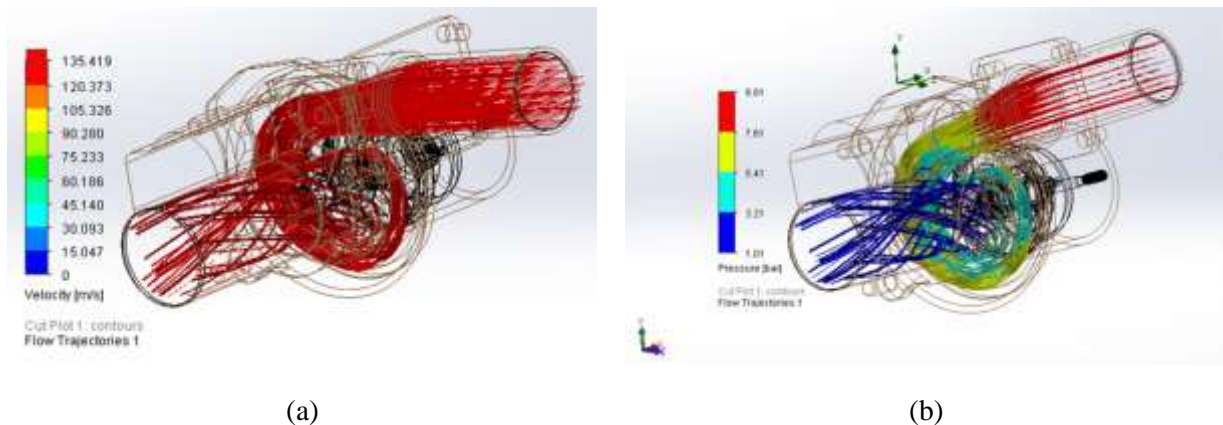

(a)   (b)

Figure 5. The results of fluid velocity a) and fluid pressure b) from micro gas turbine system

Based on Figure 5 can be explained that the fluid velocity condition at the inlet and outlet position of micro gas turbine has the same conditions, it due to the turbocharger function support to accelerate the fluid flow velocity resulting from the combustion process of LPG in the combustion chamber. The fluid pressure that occurs in the outlet position of micro gas turbine increases due to the blades number and blade sufficient curvature can increase the pressure at the blade surface. The high fluid pressure can adjust the alternator rotation. Comparison of simulation and calculation result of fluid velocity and pressure can be shown in Table 5.

Table 5. Comparative analysis between calculation and simulation results

No.	Comparative analysis		
	Parameters	Calculation	Simulation
1.	Fluid pressure outlet	1.01 – 2.73 bar	1.01 – 3.21 bar
2.	Fluid velocity outlet	-	135.4 m/s

Based on Table 5 can be described that the conditions of fluid pressure outlet from simulation result are higher than the calculations result. This condition occurs due to LPG material between calculations and simulations has different conditions. The calculation step uses the LPG characteristic in accordance with the actual conditions while the simulation section uses the LPG characteristic in ideal conditions that focused on LPG density, LPG combustion temperature, LPG heat rate, and LPG chemical compounds. The fluid velocity outlet from simulation result gets the highest value of 135.4 m/s, which is explained that the results of LPG combustion process with the ideal chemical compound was burned perfectly and the addition of turbocharger function can increase the fluid velocity that allows the alternator to rotate and be able to produce electrical energy. The condition of fluid velocity and pressure can produce a turbine rotation speed of 331.26 rpm, and the turbine outlet power reaches 139.15 kW, which results in being able to rotate the alternator faster than previous. The results of calculating and simulating additional blade blades can be concluded that its results improve the performance of micro gas turbines.

4. Conclusion

The redesign of the turbocharger turbine micro blade which is focused on the turbine blades number can be the right step in developing the turbocharger turbine micro design. This research can be concluded that an increase in the turbine blades number can influence the velocity and pressure of the fluid entering the turbine blade so as to increase the turbine power. 18 turbine blades as a new turbine blades number can increase fluid velocity by 135.4 m/s and fluid pressure by 3.21 bar, where this condition can produce a turbine rotating speed of 331.26 rpm and turbine power of 139.15 kW.

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