International Journal of Environmental Research & Clean Energy
30th July 2018. Vol.11 No.1

ISSN: 2502-3888 http://isomase.org/IJERCE1.php

Energy Analysis of Gas Engine Biogas Power Plant 835 kW in Kampar - Indonesia

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Paper History

Received: 10-October-2017
Received in revised form: 13-January-2018
Accepted: 30-July-2018

ABSTRACT

Riau Province is one of the largest palm oil producers in Indonesia. Pome of biogas is one form of renewable energy that can be utilized through biogas power plants. One of the biogas power plant in Riau is a Rama-Rama biogas power plant in Kampar. As a first step to improving power plant efficiency, this plant has been identified as the location of the largest loss in power plant by energy analysis. Energy analysis were performed for a gas engine with multiple cylinder on biogas power plant 835 kW otto cycle. The data from the plant's record books using for this analysis. The result, each gas engine cylinder has different efficiency values. The highest and lowest energy efficiency of gas engine found in cylinder 11 and cylinder 7, respectively 56,12% and 56,02%. That different efficiency value occurred due to the fact that there are large temperature differences between the combustion process every cylinder and the working fluid.

KEY WORDS: *Biogas; Energy; Gas Engine.*

NOMENCLATURE

CH_4	Methane
CO_2	Carbon Dioxide
C_p	Spesific Heat at Constant Pressure
C_v	Spesific Heat at Constant Volume
k	Specific Heat Ratio
H_2	Hydrogen
H_2O	Water
H_2S	Hydrogen Sulfide
O_2	Oxygen
$\dot{m}$	Mass Flow Rate
N_2	Nitrogen
NH_3	Ammonia
η_{th}	Thermal Efficiency
Q_{in}	Heat input
Q_{out}	Heat output
r_c	Compression Ratio
T_0	Ambient Temperature
$T_{1,2,3,4}$	Temperature of each process
P_0	Atmospheric Pressure
$P_{1,2,3,4}$	Pressure Temperature
$\dot{V}_{bb}$	Fuel Flow Rate
W_c	Compression Work
W_{net}	Net Work

1.0 INTRODUCTION

Electricity energy demand growth will cause a significant electrical energy crisis all over the world, including in Indonesia. To cope with this electrical energy crisis, the new power plant construction program should be started and the efficiency of existing power plants should be increased [1]. Riau Province's oil palm plantations nationally occupy the top position in Indonesia of 2.2 million hectares or 25 percent of the total area of oil palm plantations Indonesia [2].

Biogas from palm oil waste is a fuel used by gas engines at power plants, the development of biogas energy in Riau is very suitable and one of them is a biogas power plant. One of the biogas power plant in Kampar – Riau is a Rama-Rama biogas power plant. As a first step to improving power plant efficiency, this plant has been identified as the location of the largest loss in

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International Society of Ocean, Mechanical and Aerospace Scientists and Engineers

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power plant by energy analysis. [3]

One of the methods of thermal system analysis is based on the first law of thermodynamics. This method uses the energy balance in the system to calculate the heat transfer that occurs between the system. The first law of thermodynamics provides a concept of energy conservation, which states that the energy entering the thermal system with fuel, electricity, material flow, etc.

The purpose of this study was to evaluate the performance gas engine of biogas power plants through thermodynamic analysis.

2.0 METHODOLOGY

2.1 Data Collection

In this study, the research sample used is owned by PT. Ramajaya Pramukti. The research data used is daily report or daily operation data from the gas engine. Parameters that could not be directly measured were derived using appropriate existing equations such us compression temperature, combustion temperature, pressure during all step in gas engine. From the record of log books, the ambient temperature is 301.65 K and the ambient pressure was assumed to be 101.325 kPa.

2.2 System Description

The biogas power plant of 835 kW used in this study is an closed system with gas engine as engine drive and is located at Kampar, Riau. The gas engine works on the principle of Otto cycle four stroke, biogas fuel, water-cooled, has 16 cylinder with V configuration (70^o) and equipped with turbocharger and intercooler [4]. All Gas engine specification of biogas power plant PT. Ramajaya Pramukti is as follows on Table 1.

Table 1: Gas engine spesification and parameters [5]

Maximum power	: 835 kW
Compression ratio	: 12.5
Congfugiration	: V 70^o
Bore	: 135 mm
Stroke	: 170 mm
No. of cylinder	: 16 cylinder
Nominal speed	: 1500 rpm
Mean piston speed	: 8,5 m/s
Mean effective pressure	: 17,70 bar
Air pressure	: 1000 mbar or 100 m above sea level
Air temperature	: 25 ^{o}C or 298 K
Relative humidity	: 30%
Gas flow pressure	: 80-200 mbar

The scematic of the gas engine bigas plant is shown in Figure 1:

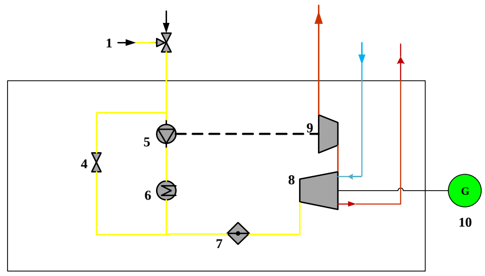

Figure 1: Schematic of methodology

Description :

1. Gas supply
2. Air inlet
3. Gas mixer
4. Turbo bypass valve
5. Compressor
6. Fuel mixture cooler
7. Throttle valve
8. Engine
9. Exhaust gas turbine
10. Generator

In addition Figure 2 showing the gas engine uses LEANOX method that supplies excess air in fuel and air mixing to minimize combustion gas emissions [5]. The excess air factor can be determined from the LEANOX method for more combustion efficiency. Then the LEANOX method show on Figure 2.

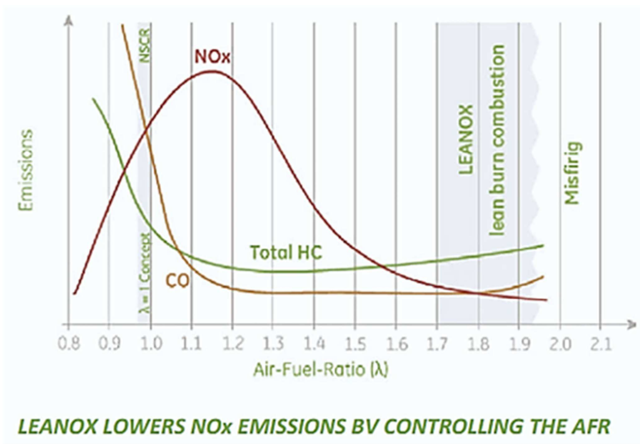

Figure 2: LEANOX method

2.3 Energy Analysis of Biogas Power Plant

The thermodinamic analysis of the gas engine by treating each component of the system as a control volume at steady state. A general energy-balance equation, applicable to any component of a thermal system may be formulated by utilizing the first laws of thermodynamics. The otto 4 stroke cycle was described in this analysys. The working fluid in the actual Otto Cycle is a mixture of fuel and air. There is a combustion process as a source of heat and there are combustion products. At the suction step, the pressure is lower than the exhaust step. The combustion process starts from ignition of the spark plug or ignition until the end of

IJERCE | Received: 30 October 2017 | Accepted: 30 July 2018 | May 2018-July 2018 [(11)1: 7-11]
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International Journal of Environmental Research & Clean Energy
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burning. The process of compression and expansion is not adiabatis, because there is a loss of heat outside the combustion chamber. The ideal Otto Cycle schematic is shown in Figure 3.

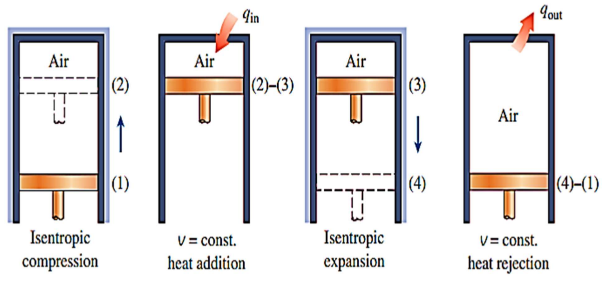

Figure 3: Ideal otto cycle schematic

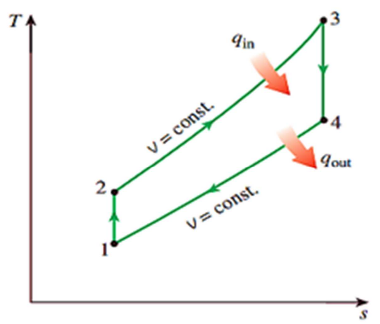

Figure 4: A typical T-s diagram of ideal otto cycle

Base on Figure 4, The thermodynamic analysis of the ideal Otto Cycle is as follows:

Isentropic compression stoke (1-2)

Compression temperature (T_2) can be determain with equation 1 [6];

$$T_2 = T_1(r_c)^{k-1} \quad (1)$$

Compression ratio (rc) is given by equation 2 [6];

$$r_c = \frac{V_1}{V_2} \quad (2)$$

Heat specific (Cp) is obtained by polynomial form as a function of temperature as given by equation 4 [7];

$$\bar{C}_p = a + bT + cT2 + dT3 \quad (3)$$

Heat specific ratio (k) is given by equation 4 [6];

$$k = \frac{C_p}{C_v} \quad (4)$$

Pressure during compression stroke is given by equation 5 [6];

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^k = P_1(r_c)^k \quad (5)$$

Work during compression process (W_c) is given by equation 6 [6]:

$$W_c = C_v(T_1 - T_2) \quad (6)$$

In this process no heat input or rejected.

$q_{1\text{-}2} = 0$

Constant-volume heat input or combustion process (2-3)

Combustion temperature (T_3) can be determain with equation 7 [7];

$$T_3 = T_4(r_c)^{k-1} \quad (7)$$

Pressure during combustion process is given by equation 8 [7];

$$P_3 = P_2\left(\frac{T_3}{T_2}\right)\left(\frac{V_2}{V_3}\right) \quad (8)$$

Heat input during combustion process is given by equation 9 [6];

$$q_{2\text{-}3} = q_{in} = c_v(T_3 - T_2) \quad (9)$$

Isentropic power or expansion stroke (3-4)

Expansion temperature (T_4) can be determined with equation 10 [7];

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k\text{-}1} = T_3\left(\frac{1}{r_c}\right)^{k\text{-}1} \quad (10)$$

Expansion pressure (T_4) can be determined with equation 11 [6];

$$P_4 = P_3\left(\frac{1}{r_c}\right)^k \quad (11)$$

In this process no heat input or rejected.

$q_{3\text{-}4} = 0$

Constant-volume heat rejection (exaust blowdown) (4-1)

Heat rejected from the engine is given by equation 12 [6]

$$q_{4\text{-}1} = q_{out} = c_v(T_4 - T_1) \quad (12)$$

While the efficiency thermal of gas engine is given as;

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \quad (13)$$

3.0 RESULTS

Physical properties flows at various state points in the gas engine at rated conditions at various state points in the cycle are shown in Table 2 and the composition of biogas that enter the gas mixture before combustion chamber shown in Table 3. These flow rates were calculated based on actual data obtained from log data record PT. Ramajaya Pramukti.

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ISSN: 2502-3888 http://isomase.org/IJERCE1.php

Table 2: Physical properties of gas engine [4]

Parameters		Value	Unit
Biogas flow rate ($\dot{V}_{bb}$)		0.118	m^3/s
Air Inlet Temperature (T_0)		301.65	K
Charge Temperature (T_1)		322.85	K
Boost Pressur (P_1)		2.76	kPa
Exhaust Gas Temperature (T_4)	$T_{4,1}$	851	K
	$T_{4,2}$	847	K
	$T_{4,3}$	855	K
	$T_{4,4}$	854	K
	$T_{4,5}$	852	K
	$T_{4,6}$	859	K
	$T_{4,7}$	845	K
	$T_{4,8}$	855	K
	$T_{4,9}$	856	K
	$T_{4,10}$	856	K
	$T_{4,11}$	861	K
	$T_{4,12}$	847	K
	$T_{4,13}$	850	K
	$T_{4,14}$	857	K
	$T_{4,15}$	845	K
	$T_{4,16}$	850	K

Table 3: Biogas composition of gas engine [4]

Ccompound	Volume (%)
CH_4	44.7
CO_2	39.1
O_2	1.7
H_2O	9.18
N_2	2.62
NH_3	1.31
H_2	1.31
H_2S	0.07
$\sum$	100

Using the value given by Table 2 and Table 3, the temperature and pressure every state are calculated by using equation 1, 5 and the result shown in the Table 4.

Table 4: Temperature and pressure every state and cylinder

Cyl	T_2 (K)	T_3 (K)	P_2 (K)	P_3 (kPa)	P_4 (kPa)
1	779,380	1775,021	7765,786	17686,406	621,886
2	779,380	1766,677	7765,786	17603,274	618,963
3	779,380	1783,364	7765,786	17769,539	624,810
4	779,380	1781,278	7765,786	17748,755	624,079
5	779,380	1777,106	7765,786	17707,189	622,617
6	779,380	1791,707	7765,786	17852,671	627,733
7	779,380	1762,506	7765,786	17561,708	617,502
8	779,380	1783,364	7765,786	17769,539	624,810
9	779,380	1785,450	7765,786	17790,322	625,540
10	779,380	1785,450	7765,786	17790,322	625,540
11	779,380	1795,879	7765,786	17894,237	629,194
12	779,380	1766,677	7765,786	17603,274	618,963
13	779,380	1772,935	7765,786	17665,623	621,156
14	779,380	1787,535	7765,786	17811,105	626,271
15	779,380	1762,506	7765,786	17561,708	617,502
16	779,380	1772,935	7765,786	17665,623	621,156

After the temperature and pressure calculated, the input dan output energy can be determined by using equation 9 and equation 12. The result shown in Table 5.

Table 5: Input and output energy in gas engine

Cylinder	Qin (kW)	Qout (kW)	Wnet (kW)
1	1449,18	636,83	812,35
2	1438,39	632,45	805,94
3	1459,97	641,22	818,76
4	1457,27	640,12	817,15
5	1451,88	637,93	813,95
6	1470,76	645,60	825,17
7	1433,00	630,26	802,73
8	1459,97	641,22	818,76
9	1462,67	642,31	820,36
10	1462,67	642,31	820,36
11	1476,16	647,79	828,37
12	1438,39	632,45	805,94
13	1446,49	635,74	810,75
14	1465,37	643,41	821,96
15	1433,00	630,26	802,73
16	1446,49	635,74	810,75

Using the value from Table 5, the efficiency of gas engine every cylinder are calculated by using equation 13 and the result shown in the Figure 5.

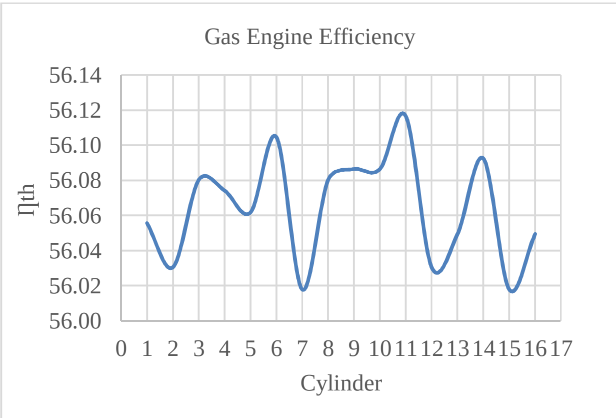

Figure 5: Efficiency gas engine of biogas power plant

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4.0 DISCUSSION

Table 2 shows all physical properties of gas engine and Table 3 shows the biogas composition as fuel in the gas engine. Each cylinder on a gas engine has a different work value even with the same fuel input, as shown in Table 5. These input, output and network from the system were calculated based on the values of measured properties such as biogas composition, pressure, temperature at various points The largest energy input in gas engine equal to 1476.16 kW (cylinder 11) and lowest energy input in gas engine equal to 1433 kW (cylinder 7). Energy efficiency is also often referred to as the efficiency of the first law of thermodynamics and the energy efficiency of gas engine multiple cylinder shown in Figure 5. The largest energy efficiency in gas engine is 56.12 % (cylinder 11) and lowest energy efficiency in gas engine is 56.02 % (cylinder 7).

5.0 CONCLUSION

The performance of a gas engine with multiple cylinder was calculated by using energy analysis. The result, each gas engine cylinder has different efficiency values. Through the energy analysis we can know the location of the performance degradation of the gas engine. It was found the highest and lowest energy efficiency of gas engine in cylinder 11 and cylinder 7, respectively. It can be concluded that different efficiency value occurred due to the fact that there are large temperature differences between the combustion process every cylinder and the working fluid.

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