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ANALYSIS OF CARBON DIOXIDE EMISSION OF GAS FUELED COGENERATION PLANT

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ABSTRACT

Gas turbines are widely used for power generation. In cogeneration system, the gas turbine generates electricity and the exhaust heat from the gas turbine is used to generate steam or chilled water. Besides enhancing the efficiency of the system, the process assists in reducing the emission of CO_2 to the environment. This study analyzes the amount of CO_2 emission by Universiti Teknologi Petronas gas fueled cogeneration system using energy balance equations. The results indicate that the cogeneration system reduces the CO_2 emission to the environment by 60%. This finding could encourage the power plant owners to install heat recovery systems to their respective plants.

Keywords: Carbon dioxide; cogeneration plant; exhaust heat; gas turbine.

INTRODUCTION

Gas turbines (GT) are widely installed at the gas fuelled power plant to generate electricity. It is integrated into the power generation systems either as an open cycle system, a combined cycle system or a cogeneration system. Besides generating electricity, the GT generates exhaust heat. The exhaust heat released to the environment consists of carbon dioxide (CO_2) and other air pollutant emission. Studies on CO_2 emission have been undertaken by a number of authors. (Graus and Worrell, 2011) reported the amount of CO_2 intensity released using power and heat method by gas-fuelled power generating system is 404 g/kWh. While (Ong, Mahlia and Masjuki, 2011) noted that the major contributor of GHG is CO_2 emissions. Changing the type of fuel percentage from fossil fuel to renewable fuel for electricity generation reduced the CO_2 emission (Masjuki, Mahlia, Choudhury and Saidur, 2002). (Mazandarani, Mahlia, Chong and Moghavvemi, 2011) have also studied the CO_2 emission with regards to fuel consumptions in Iran covering until 2025. The results show that the CO_2 emission will increase to 2.1 times by 2025. Other studies on CO_2 emission have also been undertaken by various authors. The studies are; emission assessment (Mancarella and Chicco, 2009), analysis of CO_2 emission factor (Nazari et al., 2010) and the capture and storage of CO_2 (Hendriks et al., 2009).

For the district cooling plant at Universiti Teknologi Petronas (UTP), the exhaust heat from GT is used to generate chilled water by absorption process. A study on the conversion process has been undertaken by (M Amin A Majid, 2012) . Findings from the study indicate only maximum of 66.6% of the exhaust heat can be used to generate steam while the remaining 33.4% was emitted to the environment. In another study by (Othman, Zakaria and Fernando, 2009), it is reported that the total amount of exhaust heat produced power generation is very large and to store all of the exhaust heat is not an option. This study is to focus on the evaluation of CO_2 emissions by GT for the cogeneration system.

METHODOLOGY

A typical GT for a cogeneration system is shown in Figure 1. The GT consists of a compressor, combustor and turbine section. Ambient air is first compressed in the compressor. Fuel is added and combusted in the combustor. Most GT applications rely on natural gas or fuel oil for fuel. The combustion products exit the combustor and expand in the turbine section. The expansion drives an electric generator to generate electricity. The exhaust heat from the turbine is normally used to produce process steam or hot water. The steam in turn could either be used to power steam turbine or to produce chilled water for absorption process.

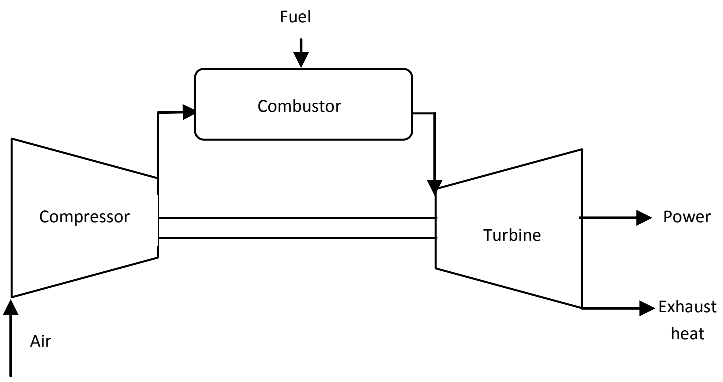

Figure 1. Typical gas turbine system.

The energy balance equation was used to evaluate the energy input, energy output and the difference between them. Eq. (1) (de Souza, 2012) is used for analysis:

$$\dot{m}_i\left(h_i+\frac{V_i^2}{2}+Z_i\right)+\dot{Q}=\dot{m}_o\left(h_o+\frac{V_o^2}{2}+Z_o\right)+\dot{W} \tag{1}$$

where $\dot{m}_i$ is mass flow rate (kg/s), h is enthalpy (kg/kJ), V is velocity (m/s), Z is elevation (m), $\dot{Q}$ is the rate of heat and $\dot{W}$ is power generated by the system. Mass and energy balance for any control volume with negligible potential and kinetic energy is expressed by Eq. (2);

$$\dot{m}_i(h_i)+\dot{Q}=\dot{m}_o(h_o)+\dot{W} \tag{2}$$

For the cycle outline in Figure 1, the thermodynamic models and related balance equations for the compressor, combustor and the turbine section are covered in the following sections:

Compressor

The model for the compressor is shown in Figure 2. By applying the energy equations for the compressor, the power output from the compressor according to (Cengel and Boles, 2006) is given as;

$$\dot{W}_c = \dot{m}_{a1}(Cp_{a2'}T_{2'} - Cp_{a1}T_1) \tag{3}$$

where $\dot{W}_c$ is power of the compressor (kW), $\dot{m}_{a1}$ is mass flow rate of air entering the compressor (kg/s), T_1 is air temperature entering the compressor (K), $T_{2'}$ is actual temperature air leaving the compressor (K) and Cp_a is the specific heat of air (kJ/kg.K).

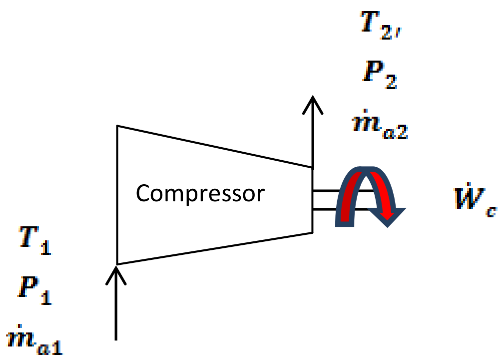

Figure 2. Thermodynamic model for compressor.

Figure 2 shows air enters the compressor at T_1, P_1 in atmospheric condition. Using the ideal gasses relation for air, the ideal temperature of T_2 is obtained by Eq. (4) (Elsaied, 2008).

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \tag{4}$$

where P_1 and P_2 are pressure entering and leaving the compressor respectively and k is the specific heat ratio of air. With an isentropic efficiency, the actual air temperature leaving the compressor is calculated using Eq. (5) (Karim and Yongo, 2011);

$$T_{2'} = \frac{(T_2 - T_1)}{\eta_c} + T_1 \tag{5}$$

where the η_c is the compressor isentropic efficiency.

Combustor

The compressed air from the compressor enters the combustion section. In the combustor, the heat is supplied by fuel as shown in Figure 3.

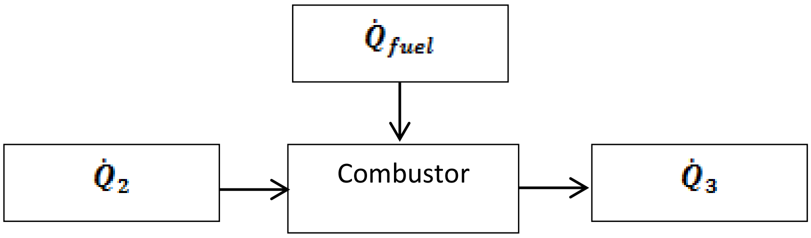

Figure 3. Thermodynamic model for combustor.

The energy input, fuel energy input and energy output from the combustor were calculated using Eq. (6), (7) and (8) respectively (Karim and Yongo, 2011);

$$\dot{Q}_2 = \dot{m}_{a2} Cp_{a2'} T_{2'} \tag{6}$$

$$\dot{Q}_{fuel} = \dot{m}_f \times LHV \tag{7}$$

$$\dot{Q}_3 = \dot{m}_{g3} Cp_{g3} T_3 \tag{8}$$

where $\dot{m}_{a2}$ is mass flow rate of air leaving the compressor (kg/s), $\dot{m}_f$ is mass flow rate of fuel (kg/s), *LHV* is lower heating value, $Cp_{a2'}$ is specific heat of air at $T_{2'}$, $\dot{m}_{g3}$ is mass flow rate of flue gas leaving the combustor, T_3 is temperature outlet of the combustor section and Cp_{g3} is specific heat of flue gas at constant pressure (kJ/kg.K). The energy balance equations model with reference to Figure 3;

$$\dot{Q}_3 = \dot{Q}_2 + \dot{Q}_{fuel} \tag{9}$$

where the mass flow rates of flue gas is denoted by Eq. (10);

$$\dot{m}_{g3} = \frac{\dot{Q}_3}{Cp_{g3} T_3} \tag{10}$$

Turbine

Figure 4 shows the thermodynamic model of the turbine. The energy output from combustor entered the turbine to generate power. During the process, exhaust heat is emitted to the environment.

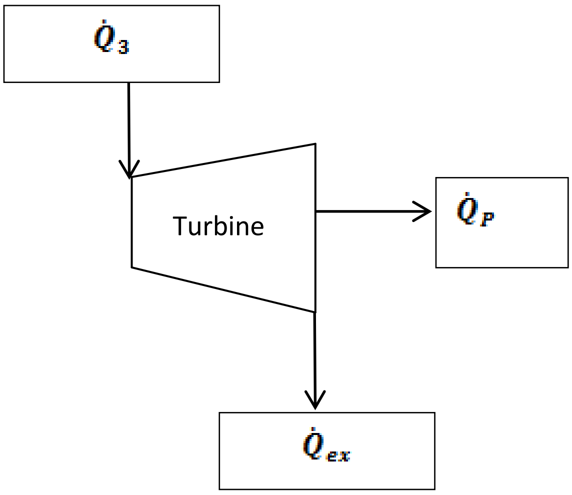

Figure 4. Thermodynamic model for turbine.

The power of the gas turbine was calculated by Eq. (11) (Karim and Yongo, 2011);

$$\dot{Q}_P = \dot{m}_{g3}(Cp_{g3}T_3 - Cp_{gP}T_P) \tag{11}$$

The exhaust heat released by the turbine was calculated as (Karim and Yongo, 2011);

$$\dot{Q}_{ex} = \dot{m}_{g3}Cp_{gex}T_{ex} \tag{12}$$

where T_{ex} is exhaust heat temperature (K), T_P is temperature for power generating (K), Cp_{gex} is specific heat of gas for the exhaust temperature (kJ/kg.K) and Cp_{gP} is specific heat of gas for power generating temperature (kJ/kg.K). The temperature of exhaust heat can be calculated as (Karim and Yongo, 2011);

$$T_{ex} = T_3 + \eta_t T_3\left(\left(\frac{1}{r_c}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{13}$$

where η_t is the isentropic efficiency of the turbine, γ is the specific heat ratio of gas and r_c is the pressure ratio of the compressor.

Gas District Cooling (GDC) plant at UTP was selected for the case study. The gas turbine is examined for operating conditions shown in Table 1 with the fluid properties shown in Table 2. The gas turbine performance was specified for continuous duty rating in standard ISO condition of 15ºC and 60% relative humidity and at sea level ambient pressure.

Table 1. Operating conditions of the UTP GDC gas turbine

Components	Operating condition	Value
Compressor	Mass flow rate of air inlet and outlet	19.0 kg/s
	Inlet temperature	300K
	Pressure Ratio	11.7
	Isentropic efficiency	0.89
	Specific heat ratio of air	1.4
Combustor	LHV	41 000 kJ/kg.K
Turbine	Specific heat ratio of gas	1.33
	Isentropic efficiency	0.91

Table 2. Physical properties of fluids (Ahmadi and Dincer, 2011)

Substance	Property	
Air	Specific heat	$Cp_a = 1.04841 - (3.8371 \times 10^{-4} \times T) + (9.4537 \times 10^{-7} \times T^2) - (5.49031 \times 10^{-10} \times T^3 + (7.9298 \times 10^{-14} \times T^4)$
Flue Gas	Specific heat	$Cp_g = 0.991615 + (6.99703 \times 10^{-5} \times T) + (2.7129 \times 10^{-7} \times T^2) - (1.22442 \times 10^{-10} \times T^3)$

For the analysis, the following assumptions were used (Karim and Yongo, 2011):

i. The process at compressor section was an adiabatic process.
ii. The total energy output from the compressor was to be taken as energy input for combustor section and remained constant.
iii. The total energy output from the combustor was to be taken as energy input for the turbine.
iv. The outlet temperature of the compressor was the temperature inlet of the turbine.
v. A turbine engine was at steady state.
vi. The pressure ratio was constant both in compressor and turbine
vii. Heat lost due to lubrication was eliminated.

For this study, historical data from 12 September 2011 to 16 September 2011 was used. In order to evaluate the amount of CO_2 emission released to the environment, the calculation was based on the earlier researches as well as on published literature.

RESULTS AND DISCUSSION

Using historical data for 12/9/2011 to 16/9/2011 and Eq. (12), the amount of exhaust heat generated is as shown in Figure 5.

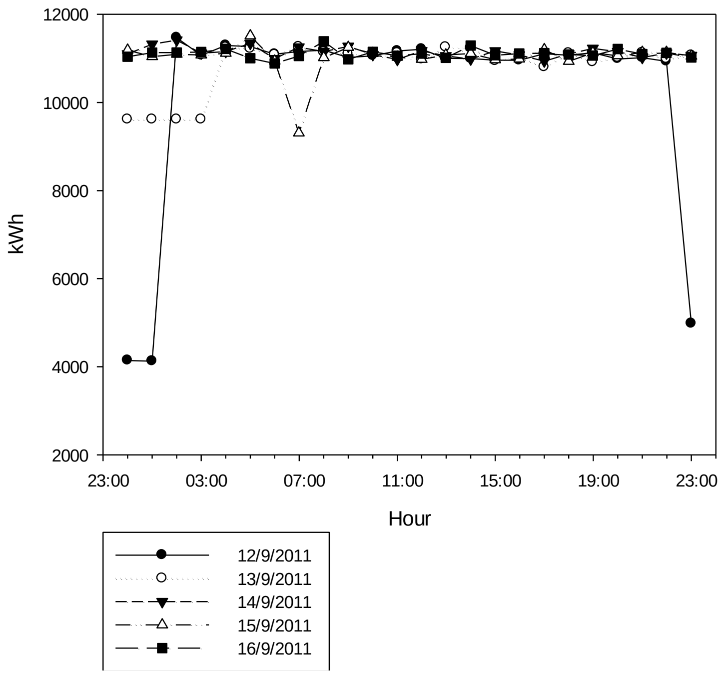

Figure 5. Estimated Exhaust heat generated by GT for 12/9/2011 to 16/9/2011.

Figure 5 shows the total hourly exhaust heat from GT from 12 a.m. to 11 p.m. for five working days in September 2011. The average amount of estimated exhaust heat released to the environment during this period is equivalent to 10 875 kWh daily. The results are summarized in Table 3.

Table 3. Summarized results of estimated exhaust heat in terms of equivalent kWh from energy analysis of GT.

	12/9/2011	13/9/2011	14/9/2011	15/9/2011	16/9/2011
Min (kWh)	4127	9615	10 940	9307	10 887
Max (kWh)	11 467	11 262	11 418	11 513	11 390
Mean (kWh)	10 267	10 843	11 138	11 020	11 105

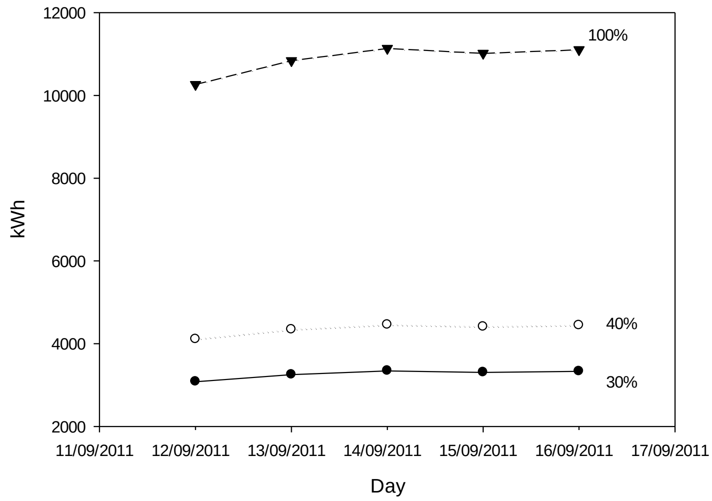

Figure 6. Amount of estimated exhaust heat based on variation percentage of exhaust heat released to the environment for 12/9/2011 to 16/9/2011.

Figure 6 and Table 4 shows the calculated amount of estimated exhaust heat released to the environment. The case of 30 percent to 40 percent refers to the maximum amount of exhaust heat released taking into account with the amount captured which is in between 60 percent to 70 percent for heat recovery process. While the case for 100 percent implies all of the exhaust heat is emitted to the environment which means there is no heat recovery process.

Table 4. Summarized results of estimated exhaust heat from GT released to the environment (kWh) based on 30%, 40% and 100%.

	12/9/2011	13/9/2011	14/9/2011	15/9/2011	16/9/2011
30%	3080	3253	3341	3306	3332
40%	4107	4337	4455	4408	4442
100%	10267	10843	11138	11020	11 105

Using conversion factor from (Kannan, Leong, Osman, Ho, and Tso, 2005), that is 474g of CO_2/kWh will be released to the environment. Figure 7 shows the plotted results for the amount of CO_2 emitted to the environment by the GT during the study period.

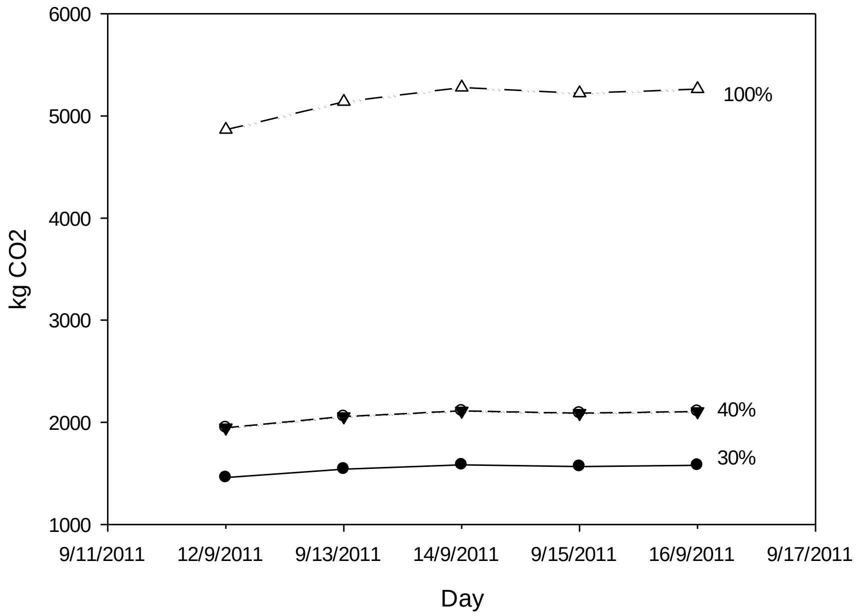

Figure 7. Estimated amount of CO_2 equivalent to exhaust heat based on 30%, 40% and 100% released to the environment for 12/9/2011 to 16/9/2011.

For the case of 100 percent of exhaust heat released to the environment, the total estimated amount of CO_2 emitted is 5155 kg. While if only 30 percent of the exhaust heat is released to the environment, the estimated amount of CO_2 emitted is 1546 kg. Thus, the steam conversion process helps to save about 60 percent of CO_2 that is 3600 kg from being released to the environment.

CONCLUSION

The energy balance analysis shows that the estimated exhaust heat in terms of equivalent per kWh from the GT is 10 875 kWh daily for district cooling plant at UTP. It contributed 5155 kg of CO_2 emission released to the environment for the GT with 8.4 MW. Increasing the amount of exhaust heat will increase the amount of CO_2 released to the environment. Thus, the heat recovery process avoided about 60 percent of CO_2 from being released to the environment. These findings could be useful for power plant owners as well as the researchers.

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