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Abstract: This article reports on a research conducted at the Gas District Cooling (GDC) facility at Universiti Teknologi PETRONAS (UTP) on the use of Turbine Inlet Air Cooling (TIAC) technology to improve turbine performance. The size of the heat exchanger was calculated and the Gas Turbine (GT) performance was examined using an analytical model. The analysis's findings show that the plant's power generation has increased by 19% as a result of the installation of TIAC technology. A heat exchanger with a capacity of 3208 m^2 may lower the GT inlet's temperature to 15°C. In order to meet the cooling demand, the size of the Thermal Energy Storage was calculated to be 1156.3m^3.

Keywords: chilled water; heat exchanger; turbine inlet air cooling

1. Introduction

There is a number of gas turbine (GT) used in Malaysia to generate electricity. The three major components that make up a GT are an air compressor, a gas turbine and a combustion chamber. The thermodynamic cycle known as the Brayton cycle is used to operate the GT[1)]. De Sa and Al Zubaidi2 conducted a study on how ambient temperatures affected a GT's performance. The findings showed that power output efficiency decreases with increasing ambient temperature for both researchers.

Most of the time, the GT relies on natural gas as fuel for its operation. In recent times, the technology of the GT has changed rapidly for the purpose of improving its efficiency towards using less fuel and generating more power. As a result, the innovation of component cooling techniques has been developed.

The GT is considered a constant volume flow rate rotating equipment [3, 4)]; therefore, its performance or output is influenced by the ambient temperature, humidity and pressure. In their analysis, several researchers found that when humidity and ambient temperatures increased, there would be a decrease in thermal efficiency and specific output[2, 5)]. It seems that the power output and efficiency of a GT is closely related to its inlet air temperature [6)]. The inlet temperature of the GT in Malaysia is the ambient temperature of the Malaysian climate. Taking the average temperature of the town of Sitiawan, in Perak in the year 2016 as 37°C [7, 8)], there was a large difference compared to the design set by the ISO gas turbine[10)] which considers the inlet air temperature of a GT to be 15°C.

According to many researchers who conducted studies on the gas turbine inlet air cooling (GTIAC) which is also the most popular method used in industry, the mechanical chiller because of its high efficiency and reliability is much preferred [5, 9, 10, 11, 12)]. Although the technology of the gas turbine inlet cooling has not been implemented in Malaysia, research showed that GT power output increased when the GTIAC technology was implemented[12, 13, 14, 15, 16)]

In UTP, the electricity is generated by its own GT to meet the needs of the campus. It is known that the GT performance is dependent on its input air inlet temperature[17, 18)]. Every GT operated is influenced by the mass flow rate of air entering the compressor[19)]. The increase of the volumetric efficiency resulted in the augmentation of power in the GT. At the same time, a decrease in GT inlet temperature led to the improvement of its volumetric efficiency. The GT inlet temperature is the same as the ambient temperature of the location where the GT is operated. Several studies have proven that by implementing the TIAC the power generated by a GT could be enhanced[20, 21, 22, 23)]. Thus, a study on the efficiency improvement of the GT at UTP GDC plant was carried out. The objective of this study was to determine a suitable size for the heat exchanger in order to decrease the inlet air temperature of the GT.

2. Methodology

This study used the acquired hourly temperature data for the town of Sitiawan, Perak, Malaysia for the year of 2016. The GT understudied were Solar Taurus 60-T7300 installed at the UTP District Cooling (DC) plant. The chiller and coil sized were based on ASHRAE's 0.4% evaporation design point. The configuration of the proposed model is shown in Fig. 1, consisting of a heat exchanger, a GT, an electric chiller, and a chilled water storage. The methodology adopted for this research is shown schematically in Fig. 2.

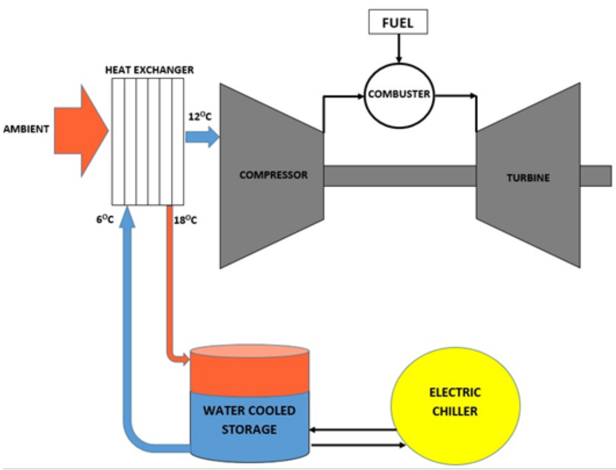

Fig. 1: Configuration of proposed model.

The acquired hourly temperature data were used to calculate the cooling load to size of the chilled water storage. The calculation of the cooling load was based on the ASHRAE Handbook 2009[24]. Based on the ISO rated inlet temperature of a GT, the sizing of heat exchanger was determined by using the number of transfer units (NTU) method[25].

As given in the ASHRAE Handbook 2009[10, 26], the cooling load was calculated as per equation (1).

$$CCL = AF_m(H_a - H_c) \tag{1}$$

where CCL is the chiller cooling load, AF_m is the mass flow rate of cooled air, H_a is the enthalpy of ambient air and H_c is the enthalpy of cooled air.

As for the chiller water storage, the size can be determined as per equation (2):

$$\text{Chilled water storage size} = \frac{\text{Cooling capacity} \times \text{Storage hours} \times 3.024\frac{Mcal}{RTh}}{\text{Chilled water } \Delta T \times 1.0\frac{Mcal}{m^{3} {}^{\circ}C} \times \text{Tank Storage efficiency}} \tag{2}$$

where, the cooling capacity was extracted from the heat balance done in equation (1), storages hours was set to 10 hours and tank storage efficiency was 85%.

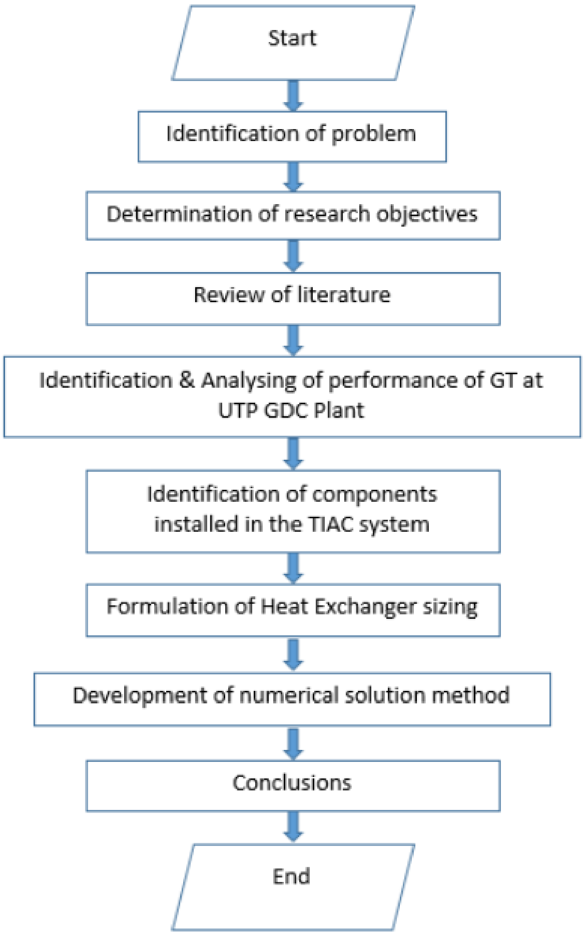

Fig. 2: Research flowchart.

2.1 Gas turbine efficiency

A schematic diagram of a standard GT cycle is shown in Fig 3. Each part has been labelled accordingly as listed in Table 1.

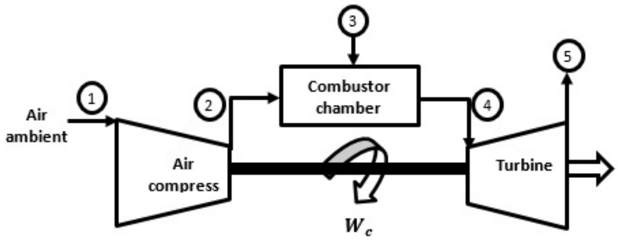

Fig. 3: Gas turbine cycle.

Table 1. Operating parameters of gas turbine.

Substance	Mas flow rate (kg/s)	Temperature (K)	Pressure (kPa)
Atmospheric condition	-	-	-
Air enters the compressor atmospheric condition	$\dot{m}_{01}$	T_{01}	P_{01}
Air at ideal outlet temperature	$\dot{m}_{02}$	T_{02}	P_{02}
Fuel enters combustor chamber	$\dot{m}_{03}$	T_{03}	P_{03}
Combustion gas enters the turbine	$\dot{m}_{04}$	T_{04}	P_{04}
Exhaust gas leaves the turbine	$\dot{m}_{05}$	T_{05}	P_{05}

The performance of the GT was determined by calculating its energy efficiency as per equation (3).

$$\eta_e = \frac{W_{nett}}{\dot{m}_{fuel} \times LHV} \quad (3)$$

Equation (4) is used to calculate W_{nett} which is the power output (kWatt). The mass flowrate of fuel (kg/s) is represented by $\dot{m}_{fuel}$ while the lower heating value is represented by LHV referred to the LHV of natural gas.

$$W_{nett} = W_{turbine} - W_{comp} \quad (4)$$

The turbine work was calculated using equation (5).

$$W_{turbine} = \dot{m}_{exhaust\ gas} \times C_{pg}(T_{inlet\ turbine} - T_{outlet\ turbine}) \quad (5)$$

Work needed by the compressor was determined using equation (6).

$$W_{comp} = \dot{m}_{air} \times C_{pg}(T_{outlet\ compressor} - T_{inlet\ compressor}) \quad (6)$$

2.2 Heat exchanger

The three main parts of a heat exchanger are the air side, water side and the fin [27]. The effectiveness of the air flow arrangement, being the counter flow, was calculated using equation (7), with the number of transfer units method known as NTU. Based on the NTU, an assumption was made that the effectiveness of the air flow and the cross flow would be similar, ε[11]:

$$NTU = \frac{U_o A_o}{C_{min}} \quad (7)$$

where the overall heat transfer co-efficient is U_o, the area of heat exchanger is A_o, whereas the minimum value of unit thermal conductance is C_{min}.

The overall heat transfers co-efficient U_o, is contributed by the water capacity, fluid capacity and fin efficiency, equation (8) is used,

$$\frac{1}{U_o} = \frac{1}{h_o \eta_{so}} + \frac{1}{h_i \left(\frac{A_i}{A_o}\right)} \quad (8)$$

where h_o is the exterior heat transfer co-efficient while, h_i is the internal heat co-efficient and, η_{so} is the surface effectiveness. The ratio of the water side and air side areas is represented by Ai / Ao.

To calculate the heat-transfer co-efficient based on the diameter, for ReD > 10,000 and using the Prandtl number in the range of 0.7<Pr<100, the Dittus-Boelter's equation (9) was used.

$$\frac{hD}{k} = 0.023 \times (Re_D)^{0.8}(Pr)^n \quad (9)$$

where n is 0.4, as the t_{wall}>t_{bulk}.

To determine the NTU value, equation (7) was used. The effectiveness of cross-flow exchanger, ε was taken into account as shown in equation (10):

$$\varepsilon = \frac{t_{co} - t_{ci}}{t_{hi} - t_{ci}} \quad (10)$$

2.3 Fluid capacity

The Reynolds number was calculated using equation (11):

$$Re_D = \frac{G_c D}{\mu} \quad (11)$$

While the J-factors were extracting from heat transfer correlation graph from ASHRAE with the correlating parameter, JP is given in Eq. (12)[28]:

$$JP = Re_D^{-0.4}\left[\frac{A}{A_t}\right]^{-0.15} \quad (12)$$

Hence, the heat transfer coefficient was determined as:

$$St\ Pr^{\frac{2}{3}} = \left(\frac{h_o}{G_c C_p}\right)\left(\frac{\mu C_p}{k}\right)^{\frac{2}{3}} \quad (13)$$

For the fin efficiency, η, it is obtained from,

$$\eta = \frac{tanh(mr\phi)}{mr\phi} \quad (14)$$

where m is given as per equation (15).

$$m = \left(\frac{2h}{ky}\right)^{\frac{1}{2}} \quad (15)$$

The term ϕ appearing in equation (14) was obtained using;

$$\phi = \left(\frac{R}{r} - 1\right)\left[1 + 0.35ln\left(\frac{R}{r}\right)\right] \quad (16)$$

To bring about a reduction in a high heat loss, an assumption was made of water velocity at 4ft/sec[29], and ρ and μ was evaluated at 15°C. The Prandtl number was calculated using equation 17[28]:

$$Pr = \frac{\mu c_p}{k} \quad (17)$$

3. Results and Discussions

The analytical model's findings indicate that thermal energy storage (TES) has a volume of 1156.3 m^3. The cooling capacity of the TES was 1143 kW extracted from the heat balance done in equation (1). For the heat exchanger analysis, using equation (1) to equation (14), the results of the heat exchanger sizing are tabulated in Table 2.

Table 2. Results of heat exchanger sizing.

Element	Size
Cooling load	1143 kW
Chiller size	198 kW
Water cooled storage	1156 m^3
Heat exchanger size (surface area)	3208 m^2
No. of rows	17
No. of tubes per row	74
Total tubes	1258

Based on the results, the cooling load capacity is estimated around 1143 kW with the chiller size needed to be 198 kW, water cooled storage, 1156 m^3, heat exchanger size (surface area), 3208 m^2, number and rows and number tubes per row are 17 and 74, respectively. The total tubes for the heat exchanger are 1258 tubes. The illustrated graphic of the heat exchanger design is shown in Fig. 4.

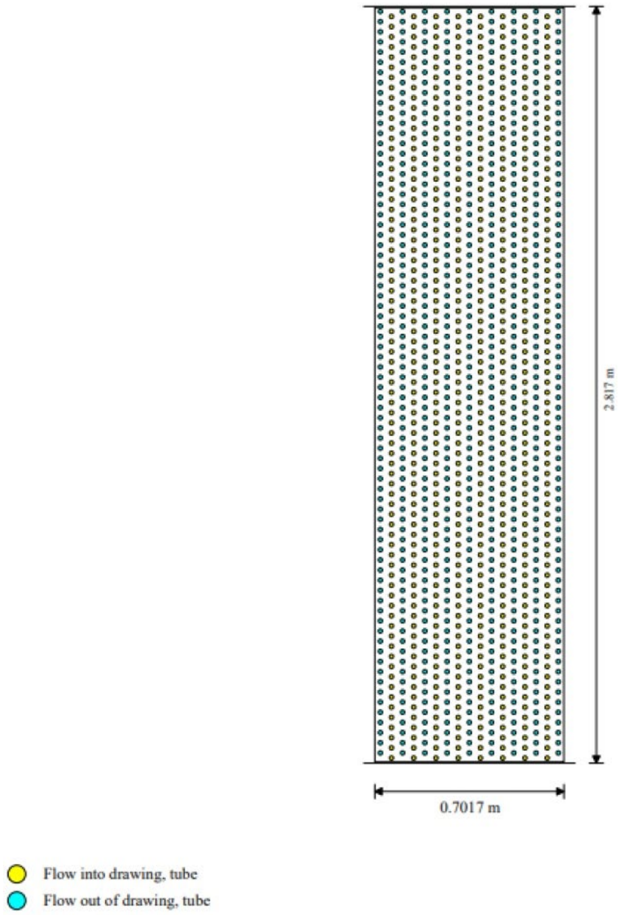

Fig. 4: Proposed heat exchanger sizing.

Based on historical data for year 2016, using equation (3) to equation (6), the efficiency of the GT is shown as Fig. 5.

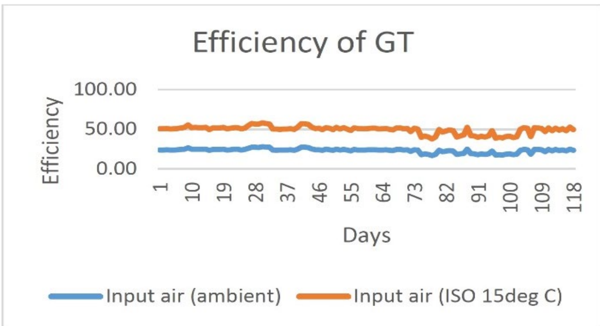

Fig. 5: Gas turbine efficiency.

Fig. 5 shows when the input air to the GT is taken from the ambient temperature, the efficiency of the GT is lower compared to when the input air is at 15°C. It was proved that in order to increase the efficiency of the GT, the input air, which is key factor, must be given significant consideration[30].

4. Conclusion

It has been demonstrated that using a heat exchanger to lower the temperature of the GT's intake air increases the plant's efficiency. This study shows that one of the crucial elements that needs to be considered while attempting to improve the GT power production and raise the plant's efficiency is the heat exchanger's size.

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Nomenclature

TIAC	Turbine inlet air cooling
UTP	Universiti Teknologi PETRONAS
GDC	Gas district cooling
GT	Gas turbine
TES	Thermal energy storage
ISO	International Standard Organization
NTU	Number of transfer unit
CCL	Cooling load
AF_m	Mass flow rate cooled air (kg/s)
$\dot{m}$	Mass flow rate (kg/s)
H_a	Enthalpy (kg/kJ)
W_c	Work for compressor (kW)
$W_{Turbine}$	Work for Turbine (kW)
W_{nett}	Work net (kW)
T	Temperature (°C)
U_o	Overall heat transfers co-efficient (W/m^2k)
A_o	Area of Air (m^2)
Re_D	Reynold number (–)
Pr	Prandtl number (–)

Greek symbols

ε	Effectiveness of cross-flow exchanger (–)
St	Stanton number (–)
η	Fin efficiency (–)
η_e	Energy efficiency (%)
ρ	Density of air (kg/m^3)
ε	Effectiveness of cross-flow exchanger (–)

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