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RESEARCH ARTICLE

PERFORMANCE ANALYSIS OF A REGENERATIVE GAS TURBINE PLANT MODELED WITH VAPOUR ABSORPTION REFRIGERATION SYSTEM

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ABSTRACT

One of the predominant factors that affect the performance of a gas turbine is the ambient temperature. This is basically because the power output of the gas turbine depends largely on the air mass flow through it. Therefore cooling the inlet air to the compressor increases the power output and overall efficiency of the plant by taking advantage of the GT higher mass flow rate when the temperature of the air entering the compressor decreases. In this work, the effects of vapour absorption refrigeration system on the performance of a GT plant were investigated. The results showed that a decrease of 2.97°C in ambient temperature from 25°C to 22.03°C at a peak turbine inlet temperature of 800K led to a gain in thermal efficiency of 0.7%, 2.443kJ/kg network output and 0.021kg/KWh drop in specific fuel consumption. The research showed that gas turbine plants perform better in temperate regions than in tropics. Therefore, retrofitting an air cooler that will restore the compressor inlet temperature to the design condition is imperative.

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INTRODUCTION

Gas turbines are widely used for electricity generation, operating air planes and have a plethora of industrial applications such as in the refineries and petrochemical plants (Mahmood and Mahdi, 2009). In quest for enhanced performance and efficiency, GT plants had gone through various modifications which have led to numerous types of GT plants. One of such is the regenerative type. Regenerative GT plant as seen in Figure 1 *typically used on low pressure ratio turbines* takes advantage of the heat energy of the exhaust gases by *recapturing some of the energy in the exhaust gas to pre-heat the air entering the combustor* in a regenerator to enhance the performance of the turbine. Regenerator is a type of heat exchanger whose function is to preheat the hot air stream coming from the compressor into the combustion chamber. The preheated air promotes better combustion (Agarwal *et al,* 2012). However, despite the regeneration, their performance could still be enhanced further. Reason being that they are usually made to work for long periods of time, under conditions that do not necessarily agree with their design specifications (Rahim *et al,* 2010). In GT plant, the combustion air is taken directly from the environment, therefore, their performance is largely affected by weather conditions.

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The power output of a GT plant depends largely on the flow of air mass through it. This is precisely why on hot days, when the air is less dense, power output falls. Therefore, the performance of GT plants is critically limited by the prevailing ambient temperature, mainly in hot and dry regions (Santos and Andrade 2012). One of the several methods of retrieving lost efficiency and power output of a GT plant through ambient temperature control is by cooling the air entering the compressor. This is achieved by adding an air cooler at the compressor inlet (Sadrameli and Goswami, 2007). When the mass flow rate of air into the compressor is increased, more power output of the gas turbine is achieved. The temperature drop provides an augment in the air density and consequently elevates air mass flow rate, this behavior increases Celsius for heavy duty turbines (Zuniga, 2005). Behdashti (2012) performed an experimental study on the effects of ambient temperature, pressure, humidity and turbine inlet temperature on power and thermal efficiency. He concluded that the ambient temperature has the greatest effect on the gas turbine performance and that GT performance improves with decrease inlet temperature and pressure ratio. Reductions of power and efficiency due to 1^{0}K temperature rise were found to be around 0.6% to 0.18% respectively. The work of Ibrahim *et al* (2011) showed that an increment of 1^{0}C in the compressor inlet air decreases the gas turbine power output by 1%. According to Kyoung *et al,* (2012), cooling the inlet air of the gas turbine decreases the inlet temperature which in turn increases the air density, and hence increasing the mass flow rate.

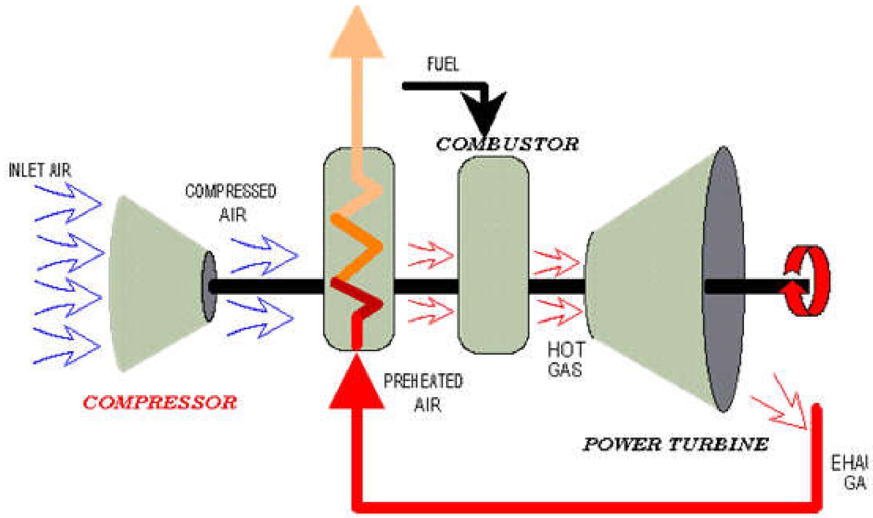

Figure 1. GT with Regeneration (Source: Nye Thermodynamic Corporation, 2018)

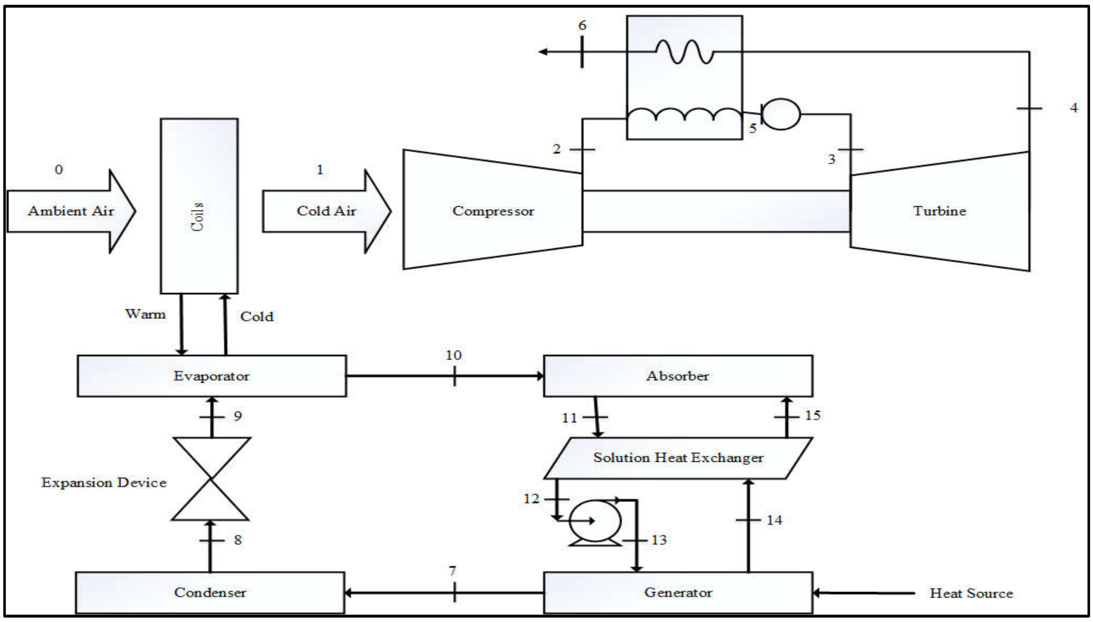

Figure 2. Schematic Diagram of a Regenerative GT with VARS Inlet Cooling

Different innovative techniques have been employed to achieve the GT compressor inlet air cooling. Amongst such methods are: evaporative inlet cooling and inlet air cooling with refrigeration systems. Marzouk and Hanafi (2013) reported that evaporative cooling could enhance the power produced by 2-4% per year depending on the weather. Power augmentation of a typical gas turbine cycle using a desiccant based evaporative cooling system was studied by (Zadpoor and Golshan, 2006). The technique requires a desiccant based dehumidifying process to direct the air through an evaporative cooler, which could be either media based or spray type. According to Johnke and Mast (2002) a media cooler can increase the relative humidity of the inlet air to about 90%, thus increasing power output by 5-10% and the efficiency by 1.5-2.5%. The study of Elliot (2001) showed that a 1% gain of the output power was obtained for every 1.6^{0}C drop in compressor inlet air temperature using water chillers. The feasibility of improving the simple GT cycle efficiency and power by cooling the inlet air using an absorption system was reported by Najjar (1996) and Poku (2017). Bartolini and Salvi (1997) also reported 8% increase in power and 4% increase in thermal efficiency of the steam injected gas turbines when cooling the inlet air using absorption chiller. Poku *et al.* (2017) studied the performance augmentation of a Brayton's cycle GT plant employing VARS as compressor inlet air cooling technique. The study was able to achieve efficiencies of 7% and 8.4% with temperatures drop by 10^{0}K and 15^{0}K respectively. Bassily (2001), presented the effects of the turbine inlet temperature cooling and relative humidity on the performance of the recuperated gas turbine cycle with indirect evaporative inlet cooling, and the results showed that evaporative cooling of the inlet air could boost the efficiency by up to 3.2%. This research therefore attempts to study the performance of a regenerative GT plant employing a VARS as the compressor inlet air cooler at different turbine peak temperatures.

MATERIALS AND METHODS

Operating data were obtained from a working gas turbine plant and the average of the daily readings was calculated for a period of one year.

Table 1. Basic Parameters

S/N	Operating Parameters	Values	Units
1	Mass flow rate of air through compressor (m_a)	125.2	kg/s
2	Temperature of inlet air to compressor (T_1)	298	^{0}K
3	Pressure of inlet air to compressor (P_1)	1.013	Bar
4	Compressor Exit Pressure (P_2)	6.4	Bar
5	Fuel (natural) mass flow rate (m_f)	4.5	kg/s
6	Turbine Inlet Pressure (P_3)	6.43	Bar
7	Inlet Temperature to gas Turbine (T_3)	800 - 1100	^{0}K
8	Outlet Pressure from Turbine	1.013	Bar
9	Lower Heating Value (LHV)	47,541.60	kJ/kg
10	Isentropic Efficiency of Compressor	85	%
12	Isentropic Efficiency of Turbine	87	%
13	Generator Effectiveness	87	%
14	c_{pa} for compressor process	1.005	kJ/kg^{0}K
15	c_{pg} for expansion	1.15	kJ/kg^{0}K
16	Pressure loss across heat exchanger	3%	
17	Υ_a, Υ_g	1.4, 1.333	
18	Pressure Drop across regenerator	3%	

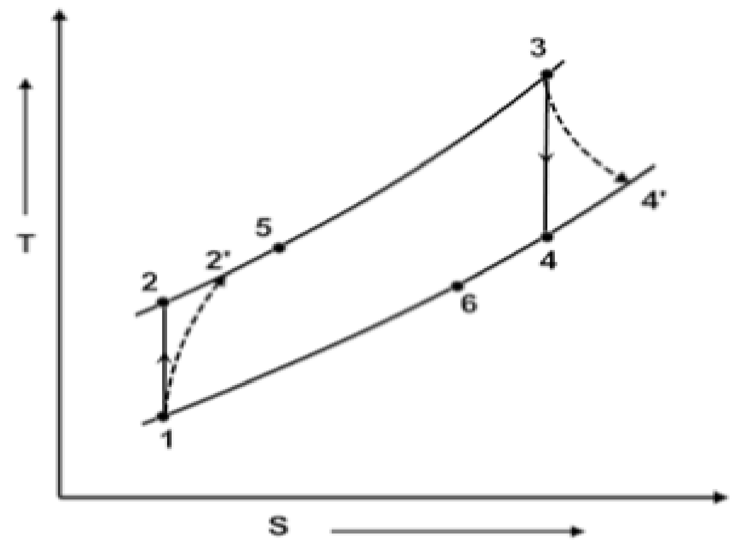

Figure 3. T-S Diagram of a Regenerative Cycle

These results were then used to calculate the work done by the compressor, turbine network output, heat supplied, air/fuel ratio, specific fuel consumption and thermal efficiency for a regenerative GT cycle incorporating a VARS as the inlet air cooler. Mass and energy conservation laws were applied to each component and the performance of the plant was determined. A schematic of the regenerative GT plant with the VARS as the inlet air cooler is shown in Figure 2. Also, the parameters used for the determination of the plant parameters are as seen in tables 1.

Thermodynamic analysis of the regenerative gt plant

Figure 3 is the T-S diagram of a regenerative GT cycle without a compressor inlet air cooler. Figure 3 shows the actual and ideal processes with dashed and full lines respectively. In order to arrive at a better understanding of this work, thermodynamic equations were derived which were used for subsequent calculations. This is aim at obtaining thermodynamic equations for the net work, heat input into the combustor, specific fuel consumption, power output, thermal efficiency and their relationship with parameters such as temperature and pressure ratios.

Isentropic compression process 1-2:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma a-1}{\gamma a}} = \left(r_p\right)^{\frac{\gamma a-1}{\gamma a}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots..1$$

Where γ_p = pressure ratio

Isentropic Efficiency of the compressor, η_c:

$$\eta_c = \frac{T_2 - T_1}{T_{2'} - T_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 2$$

Isentropic expansion process 3-4:

$$\frac{T_4}{T_3} = \left(\frac{P_{4'}}{P_3}\right)^{\frac{\gamma g-1}{\gamma g}} = \left(r_p\right)^{\frac{\gamma g-1}{\gamma g}}$$

$$T_4 = T_3\left(r_p\right)^{\frac{\gamma g-1}{\gamma g}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 3$$

For isentropic efficiency of the turbine, η_t:

$$\eta_t = \frac{T_3 - T_4{}'}{T_3 - T_4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots..4$$

Regenerative effectiveness ε_r is given as:

$$\varepsilon_r = \frac{T_5 - T_{2'}}{T_4{}' - T_2{}'}$$

$$T_5 = T_{2'} + \varepsilon_r\left(T_4{}' - T_2{}'\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 5$$

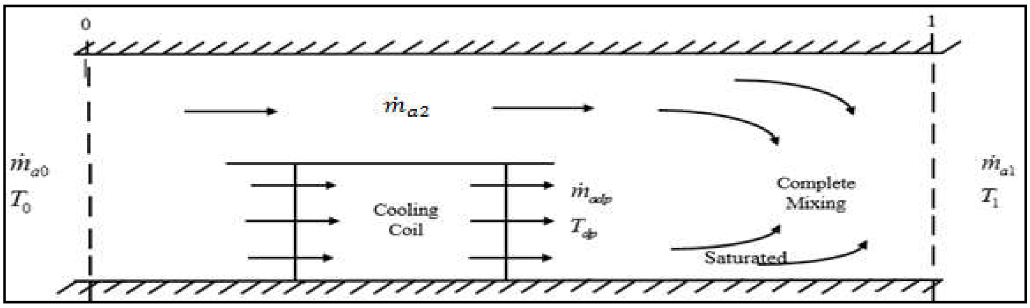

Figure 4. Schematic diagram of the bypass – factor apparatus for a cooling/dehumidifying

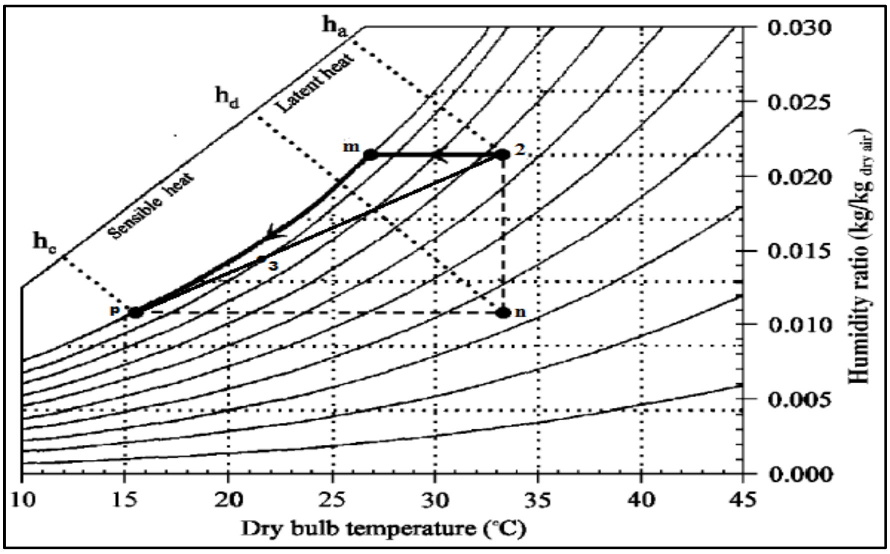

Figure 5. Air cooling process on a psychometric chart (Ameri and Hejazi, 2004)

Table 2. GT study results with inlet cooling with turbine inlet temperatures of 800K and 1000K

GT with inlet cooling having turbine inlet temp of 800K								
T_0 (^{0}C)	T_1 (^{0}C)	Compressor Work (kJ/kg)	Turbine Work (kJ/kg)	Heat Supply(kJ/kg)	Efficiency	SFC	AFR	Net Work(kJ/kg)
25	22.03	242.63	292.11	293.75	0.17	0.45	161.84	49.48
27	22.77	243.24	292.11	295.07	0.17	0.46	161.12	48.87
29	23.42	243.78	292.11	296.23	0.16	0.46	160.49	48.33
31	24.07	244.31	292.11	297.38	0.16	0.47	159.87	47.80
GT with inlet cooling having turbine inlet temp of 1000K								
25	22.03	242.63	365.14	387.42	0.32	0.24	122.71	122.50
27	22.77	243.24	365.14	388.74	0.31	0.24	122.30	121.90
29	23.42	243.78	365.14	389.89	0.31	0.24	121.94	121.36
31	24.07	244.31	365.14	391.05	0.31	0.25	121.58	120.83

Compressor work is:

$$W_c = \dot{m}_a C_{Pa}(T_2 - T_1)$$

The actual work of the compressor is:

$$W_c = \frac{\dot{m}_a C_{pa} T_1}{\eta_c}\left[\left(\frac{P_2}{P_1}\right)^{\gamma_a - 1/\gamma_a} - 1\right] \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 6$$

Turbine work is:

$$W_t = \dot{m}_g C_{pg}(T_3 - T_4{}')$$

The work developed by the Turbine is:

$$W_t = \dot{m}_g C_{pg} T_3 \eta_t \left(1 - \frac{1}{\left(\frac{P_4{}'}{P_3}\right)^{\frac{\gamma_g - 1}{\gamma_g}}}\right) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 7$$

Network is expressed as:

$$W_{net} = W_t - W_c \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 8$$

The heat input in combustion chamber is:

$$Q_{add} = \dot{m}_g C_{pg}(T_3 - T_5) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 9$$

The Power output is given by:

$$P = \dot{m}_a \times W_{net} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 10$$

Air-fuel ratio is given by

$$AFR = \frac{LCV_f}{Q_{add}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 11$$

Table 3. GT study results without inlet cooling with turbine inlet temperatures of 800K and 1000K

GT without inlet cooling having turbine inlet temp of 800K							
T_0 (^{0}C)	Compressor Work (kJ/kg)	Turbine Work (kJ/kg)	Heat Supply(kJ/kg)	Efficiency	SFC	AFR	Net Work(kJ/kg)
25	245.08	292.11	292.82	0.16	0.47	162.36	47.03
27	246.72	292.11	296.38	0.15	0.49	160.41	45.39
29	248.37	292.11	299.93	0.15	0.52	158.51	43.74
31	250.01	292.11	303.49	0.14	0.55	156.65	42.10
GT without inlet cooling having turbine inlet temp of 1000K							
25	245.08	365.14	381.03	0.32	0.24	124.77	120.06
27	246.72	365.14	384.59	0.31	0.25	123.62	118.42
29	248.37	365.14	388.14	0.30	0.25	122.49	116.77
31	250.01	365.14	391.70	0.29	0.26	121.37	115.13

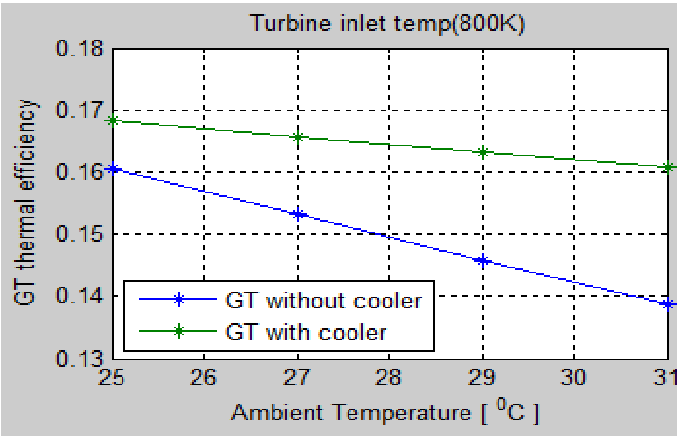

Figure 6. Effects of Ambient Temperature on Thermal Efficiency at T_3 = 800 K.

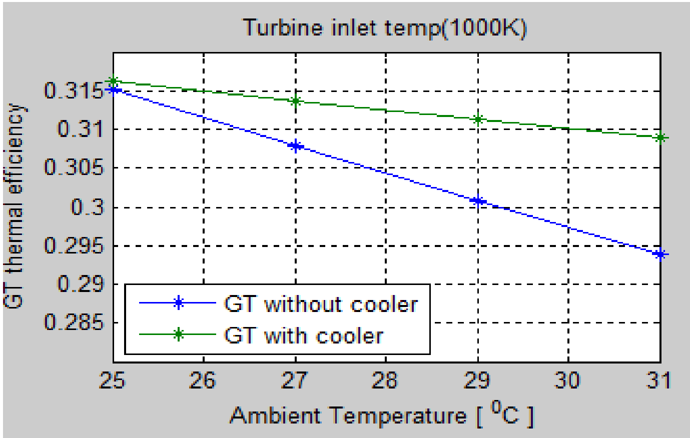

Figure 7. Effects of Ambient Temperature on Thermal Efficiency at T_3 = 1000 K.

The specific fuel consumption can be express as:

$$SFC = \frac{3600}{AFR \times W_{net}} kg/kWh \qquad \text{.........................12}$$

Thermal efficiency is given by:

$$\eta_{th} = \frac{W_{net}}{Q_{add}} = \frac{W_{net}}{\dot{m}_f \times LHV} \qquad \text{... ... ... 13}$$

Effect of Cooling Coil on the Compressor Inlet air

Air could be made cooled by placing a cooling coil across the air flow stream. When air of mass, $\dot{m}_{a0}$ passes over a cooling coil as shown in Figure 4, part of it, $\dot{m}_{a2}$ passes unaffected while the remaining, $\dot{m}_{adp}$ comes in direct contact with the coil. The fraction of air that misses the coil is measured in terms of a by-pass factor, b*f*. The b*f* depends upon the pitch of the cooling coil fins, numbers of rows in a coil in the direction and velocity of air flow.

The continuity equation for dry air is:

$$\dot{m}_{a0} = \dot{m}_{a1} = \dot{m}_a \qquad \text{.......................14}$$

$$\dot{m}_{a2} + \dot{m}_{adp} = \dot{m}_{a1} \qquad \text{......................15}$$

The by-pass factor, b*f* is given as:

$$b_f = \frac{\dot{m}_{a2}}{\dot{m}_a} \qquad \text{... ... ... .16}$$

The compressor inlet temperature, T_1 could be calculated from the mixing process as:

$$T_1 = \frac{\dot{m}_{adp}}{\dot{m}_a} T_{dp} + \frac{\dot{m}_{a2}}{\dot{m}_a} T_0 \qquad \text{.................17}$$

From equations 15, 16 and 17, the b_f is expressed as:

$$b_f = \frac{T_1 - T_{dp}}{T_0 - T_{dp}} \qquad \text{... ... ... 16}$$

Therefore, T_1 is given as:

$$T_1 = b_f(T_1 - T_{dp}) + T_{dp} \qquad \text{......................18}$$

The pressure drop of air in the coil is 1% of the ambient pressure and the cooling coil by-pass factor is 0.15. The cooling in the coil could be explained with the psychometric chart in Figure 5. The process takes the path of dehumidification from 2 to n and then cooled from n to P.

RESULT AND DISCUSSION

The parameters used in determining the performance of the regenerative GT cycle are as shown in Tables 2 and 3. Parameters in Tables 2 were generated at different ambient air temperature, T_0 and cooled to their corresponding compressor inlet temperature of T_1 at turbine inlet temperatures, T_3 of 800K and 1000K respectively.

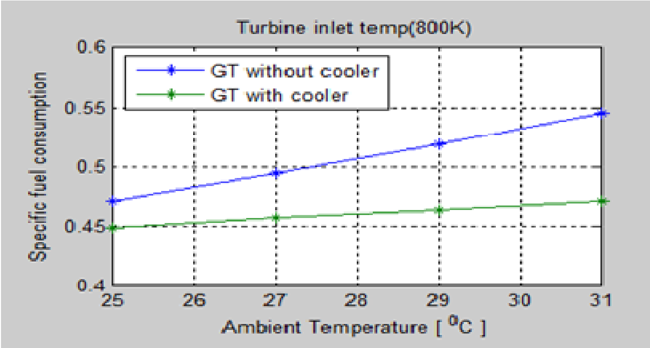

Figure 8. Effects of Ambient Temperature on the Specific Fuel Consumption at T_3 = 800 K.

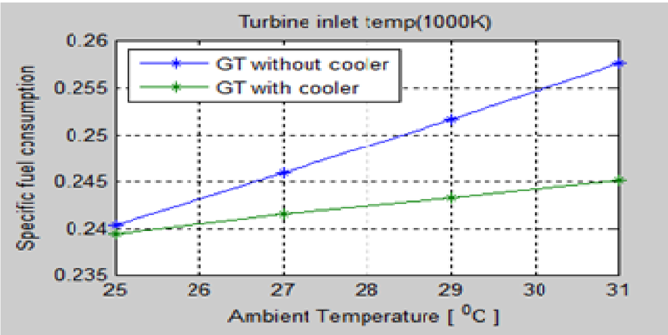

Figure 9. Effects of Ambient Temperature on Specific Fuel Consumption at T_3=1000 K

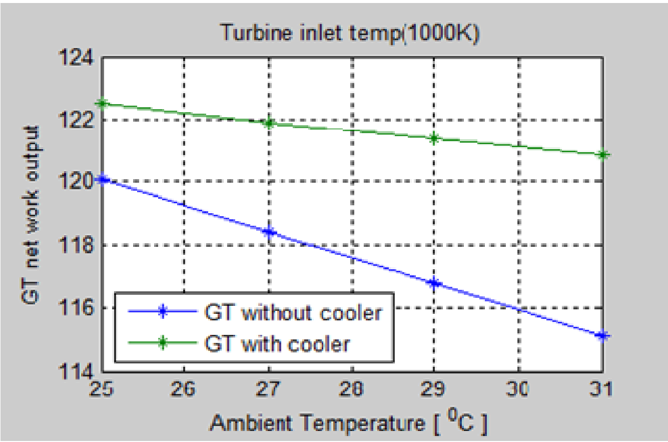

Figure 10. Effects of Ambient Temperature on GT Net Work Output at T_3 = 800 K.

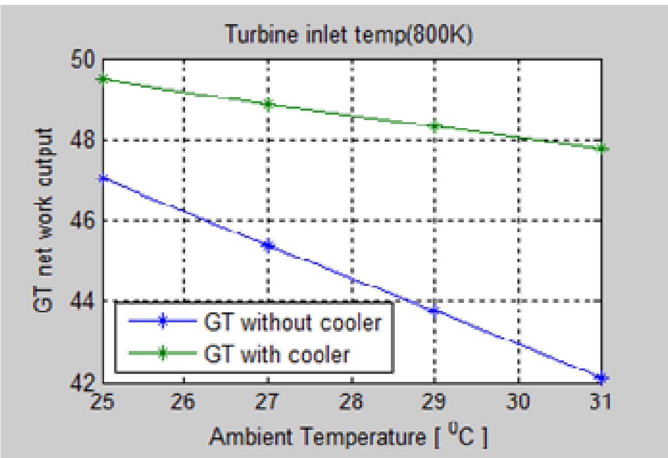

Figure 11. Effects of Ambient Temperature on GT Net Work Output at T_3 = 1000 K.

DISCUSSION

The influences of parameters such as variations in ambient temperatures and turbine inlet temperatures on the performance of a regenerative gas turbine were studied in this section. The parameters used for the plant analysis were thermal efficiency, net work output and the specific fuel consumption. Figures 6 and 7 showed that the gas turbine thermal efficiency is obviously affected by ambient temperature due to the change in air densities that affect the compressor work. According to Farouk *et al.* (2013) GT power decreases due to reduction in air mass flow rate (the density of the air declines as temperature increases) and the efficiency decreases because the compressor requires more power to compress air of higher temperature. Since drop in ambient temperature leads to a higher air density and a lower compressor work, it can be seen in Figures 6 and 7 that when the ambient temperature increases the thermal efficiency decreases. The reason was that as the ambient temperature decreases, the air mass flow rate into compressor increases. It was therefore observed that when the compressor inlet cooler is installed in a prevailing environmental temperature of 25^{o}C and was cooled to 22.3^{o}C, there is an approximately 0.7% increase in thermal efficiency. This result is observed for all the different ambient conditions but with varying values in thermal efficiency. Figures 6 and 7 also showed the relation between the thermal efficiency and turbine inlet temperatures of GT power plants. Thermal efficiency increases with rise in turbine inlet temperature.

Figures 8 and 9 showed the variations of the ambient temperature with the specific fuel consumption. It is clear that environmental conditions have a tremendous influence on the specific fuel consumption of the plant. With a close observation of table 1, it is observed that there is approximately 0.025kg/KWh increase in specific fuel consumption with a 2^{o}C rise in ambient temperature. With prevailing environmental temperature of 25^{o}C, a drop of 2.7^{o}C in the ambient temperature after cooling was able to achieve a decrease in specific fuel consumption of 0.021 kg/KWh. It was also observed that there is a significant drop in specific fuel consumption as the turbine inlet temperature increases. This confirms the assertion according to Ibrahim *et al.* (2011) that the effect of variation of sfc is more significance at higher ambient temperature than lower temperature. The turbine inlet temperature increased means that more fuel would be burned which leads to decrease in specific fuel consumption. Therefore, the fuel mass flow rate will increase, since air to fuel ratio is kept constant. The specific fuel consumption therefore decreases with rise in turbine inlet temperature. This shows that gas turbine power plant performance can be improved by increasing the turbine inlet temperature from 800K to 1000K. Figures 10 and 11 illustrate the effect of the ambient temperature on the net work output of the gas turbine. The result showed that as the ambient temperature increases by 2^{o}C as in table 1, there is approximately 1.644 kJ/kg decrease in the net work output of the plant but with the cooler installed with the prevailing ambient temperature of 25^{o}C, there is 2.443kJ/kg increase in the net work of the gas turbine. It is also observed that as the turbine inlet temperature increases from 800K to 1000K, the work output also increases. This result is observed for the different ambient temperatures but with varying values in the network output.

Conclusion

The effects and influence of ambient temperatures, turbine peak temperatures were carried out on a GT plant with VARS as the compressor inlet cooler. The results are therefore summarized as follows:

- The thermal efficiency and network increase by 0.7% and 2.443kJ/kg respectively when there is slightest drop in ambient temperature and the specific fuel consumption increases by 0.025kg/KWh with 2^0C rise in the ambient temperature.
- The network output increases by 2.443kJ/kg with drop in ambient temperature and increase in turbine peak temperature from 800K to 1000K.

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