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Performance Augmentation of a Gas Turbine Plant Using an Absorption Refrigeration System

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Authors' contributions

This work was carried out in collaboration between all authors. Author RP designed the study, performed the statistical analysis, wrote the protocol and wrote the first draft of the manuscript. Authors POO and TWO managed the literature searches. All authors read and approved the final manuscript.

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ABSTRACT

Performance of a gas turbine is mainly depended on the inlet air temperature. The power output of a gas turbine depends on the flow of mass through it. Inlet air cooling increases the power output by taking advantage of the gas turbine's feature of higher mass flow rate when the compressor inlet temperature decreases. In this paper the performance enhancement of gas turbine power plants by cooling the compressor intake air with an absorption refrigeration system was studied. This work investigated the effect of inlet air cooling system on the performance of an existing gas turbine power plant in Nigeria and it was made possible with MATLAB R2013a. The results showed that when the intake air temperature decreases by 10-15°K, the Efficiency and Net-work output increases by 7% and 8.4% respectively. Therefore, it becomes clear that there is increment in performance as the turbine inlet temperature approaches the ISO rating 15°C.

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Keywords: Ambient temperature; absorption refrigeration system; efficiency; net work output.

NOMENCLATURES

η_{th}=Thermal Efficiency, Wnet = Net Work of Turbine (kJ/kg), r=Pressure ratio, Wc = Compressor Work (kJ/kg), $\delta = Isentropic\ index$, SFC = Specific Fuel Consumption (kg/KWh), P = Pressure (bar), μ_c =Isentropic Efficiency of Compressor (kJ/kg), μ_T =Isentropic Efficiency of Turbine (kJ/kg), T_1 = Ambient Air Temperature (^{0}C), T_3 = t_3 = Temperature after Cooling (^{0}C), Cpg = Specific heat Capacity of the Gas (kJ/kgK), Q_{sup} = Heat Added (kJ/kg), m_a = Mass of the Air (kg/s), GT= Gas Turbine, m_g = Mass of the Gas (kg/s), ARS = Absorption Refrigeration System, Cpa = Specific heat Capacity of the air (kJ/kgK), HR=Heat Rate KJ/KWh, LHV_f = Lower Calorific Value of Fuel (kJ/kg), AFR = Air Fuel Ratio.

1. INTRODUCTION

The average efficiency of gas turbine (GT) plants in Nigeria energy utility sector over the past three decades was in the range of 20-28% after adjusting for site conditions [1]. The low efficiency of GT plants can be attributed to various reasons ranging from operation mode, poor maintenance as well as engine size and age. The rated capacities of all combustion turbines are based on standard ambient air temperature of 15°C, 60% relative humidity, 101.325 Kpa at sea level, zero inlet and exhaust pressure drop, as selected by the International Organization for Standardization (ISO) [2]. Throughout the year, the average temperature in Nigeria falls between 22°C and 32°C. However, the mercury can rise as high as 43.3°C during the hottest time of the dry season and drops to 18°C during the rainy season [1].

Virtually, all the GT power plants in Nigeria are sited at locations where the frequency of occurrence of ambient air temperature at 15°C or lower, in terms of the number of hours, rarely exists on yearly basis. The very high frequency of occurrence of ambient air temperature above 15°C implies that the gas turbines operate at off - ISO conditions almost all the time [3]. Whenever a gas turbine operates at site ambient conditions that differ from the ISO conditions, there is a deviation from the plant design performance rating. For all gas turbines, increased ambient air temperature or site elevation decreases power output and also reduces fuel efficiency [2]. In the area where the gas turbine considered for this study is located, in Kolo Greek, Imiringi, Bayelsa State, Nigeria, the yearly ambient air temperature varies between 25°C and 32°C, which has also been recorded in the past data acquired from the power station. According to study by Tolumoye [4] on the Kolo-Greek SK30 GT, owing to high ambient temperature above the ISO value, the full load capacity of the GT drops during hot daytime with about 1-2 MW out of the 20MW output achieved in morning and night hours

Turbine inlet chilling is defined as cooling the air before it gets into the compressor of the gas turbine. Jonsson and Yan [5] concluded that chillers can increase gas turbine power output by 15–20% and efficiency by 1–2% (i.e. if gas turbine exhaust gas energy is recovered). However, the specific investment cost of chillers is higher than for evaporative coolers. Kakaras et al. [6] presented a computer simulation of the integration of an innovative absorption chiller technology for reducing the intake-air temperature in gas turbine plants. They concluded that the effect of ambient air temperature variation has the large penalty in the plant's performance in high ambient temperatures and leads to drop in performance. The study of Elliot [7] showed that a 1% gain of output power was obtained for every 1.6°C drop in compressor inlet air temperature using water chillers. In addition, Mercer [8] reported in his study that chillers utilizing thermal storage systems would increase the gas turbine power output by over 25% during peak periods. Shukla and Singh [9] studied the effects of inlet evaporative cooling on the power augmentation of a steam injection GT. The study showed 3.2% increment in thermal efficiency with a drop in inlet temperature by 36°C (from 45°C-9°C).

The aim of this paper is to investigate the effect of an inlet air cooling system on the performance of an existing gas turbine plant in Nigeria. This work seeks to explore the inherent environmental benefit of the Kolo-Greek SK30 Gas turbine plant incorporating an absorption refrigeration system (ARS) in the prevailing climatic conditions. The purpose of the ARS is to increase the power output of the combined cycle by cooling the intake air entering the gas turbine compressor.

The performance of the proposed modified Brayton cycle is compared with base condition (without inlet air cooling).

1.1 Absorption Refrigeration System

The absorption refrigeration cycle has recently attracted much research attention because of the possibility of using waste thermal energy or renewable energy as the power source, thus reducing the demand for electricity supply [10]. The two great advantages of the absorption cycles compared to other cycles with similar production are that no large, rotating mechanical equipment is required and any source of heat can be used including low temperature waste heat [2].

Heat is rejected in the absorber and condenser while heat is injected into the system through the generator and the evaporator. Refrigerant rich solution from the outlet of the absorber is pumped by the solution pump to the generator. In the generator the refrigerant is then extracted from the refrigerant/absorbent solution by heat from an external heat source into the generator. The rest of the solution in liquid state is drained back to the absorber as absorbent, ready to absorb the refrigerant vapour from the evaporator. The extracted high pressure vapour refrigerant from the generator flows into condenser where heat is rejected resulting to high pressure liquid refrigerant. The liquid then passes through an expansion valve which lowers the pressure and produces a low temperature refrigerant liquid that flows into the evaporator where it evaporates, providing the cooling effect which is used to generate chilled water [11].

Lithium Bromide-Water and Water-Ammonia as conventional fluids still have desirable properties compared to other working fluid variants. The Lithium Bromide-Water combination is limited to temperatures above the freezing point of water (i.e. above 0°C) while the Water-Ammonia combination is favourable for sub-zero refrigerant temperatures (i.e. below 0°C). Hence, Lithium Bromide-Water was used in this study since the air to be cooled is above 0°C. The refrigerant should be more volatile than the absorbent so that the two can be separated easily. Water is usually used as the refrigerant for the solid absorbents [12]. It is very important to note that a compact heat exchanger/cooling coil should be designed for installation at the compressor inlet duct. The chilled water from the absorption chiller flows through the heat exchanger and cools the inlet air [13]. A schematic diagram of a typical gas turbine incorporated with an absorption refrigeration cooling system is shown in Fig. 1.

2. PARAMETER AND METHODOLOGY

SK30 GT unit using liquefied petroleum gas (LPG) as fuel was chosen for this study. LPG besides its availability was chosen because it is more pure and gives higher efficiency than light diesel oil [14]. Operating data were collected from the daily turbine control log sheet for a period of six years (2005 – 2010). The average daily operating variables were statistically analyzed and mean values were computed for the period of January to December, followed by an overall average. Summary of operating parameters for the SK30 gas turbine unit as obtained and computed from the log sheet is presented in Table 1. The parameters that could not be sourced were calculated

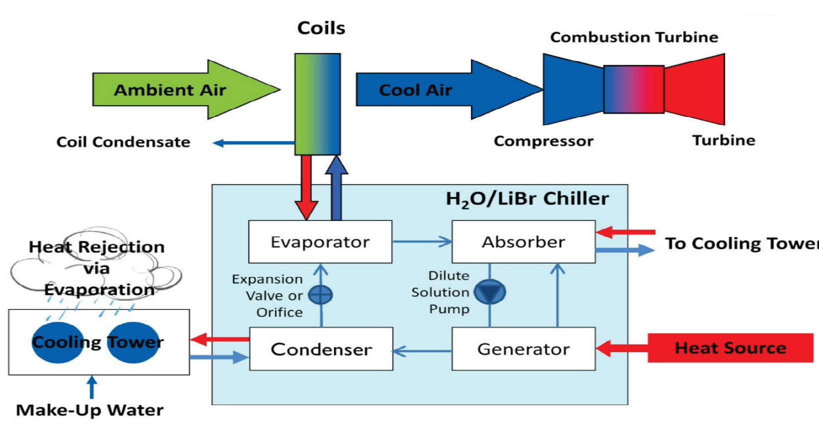

Fig. 1. Typical turbine inlet chilling process [11]

thermodynamically using standard formulae. Mass and energy conservation laws were applied to each component and the performance of the plant was determined for the base-case, without an ARS and for the cooled system with an ARS. MATLAB R2013a was also employed to generate the values for the analysis.

2.1 Thermodynamic Analysis of GT Unit (Simple type)

The inlet pressure is given as:

$$P_0 = P_1 \tag{1}$$

The pressure of the air leaving the compressor (P_2) is calculated as:

$$P_2 = rP_1 \tag{2}$$

Where, r represents the pressure ratio

It is assumed that the change in kinetic energy between the various points in the cycle is negligible. Then applying the flow equation to each part of the cycle, we have the following for unit mass.

2.2 For the Compressor

According to Ogbonnaya [17] and Cengel and Boles [18];

Compressor Work input,

$$w_c = C_{pa}(T_2 - T_1) \tag{3}$$

From Eastop and McConkey [19], for an isentropic process,

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{(\delta-1)}{\delta}} \tag{4}$$

Where:

$$\delta = Isentropic\ index$$

The Isentropic efficiency of the compressor is defined as the ratio of the work input required in isentropic compression between P_1 and P_2 to the actual work required. Neglecting change in kinetic energy we have,

Compressor isentropic efficiency, μ_c is given as:

$$\mu_c = \frac{C_p(T_{2s}-T_1)}{C_p(T_2-T_1)} = \frac{(T_{2s}-T_1)}{(T_2-T_1)} \tag{5}$$

2.3 For the Combustion Chamber

Heat supplied,

$$Q_{sup} = C_{pg}(T_3 - T_2) \tag{6}$$

2.4 For the Turbine

From Cengel and Boles [18], Turbine work Output is:

$$w_t = C_{pg}(T_4 - T_3) \tag{7}$$

For isentropic expansion process,

$$\frac{T_3}{T_{4s}} = \left(\frac{P_3}{P_4}\right)^{\frac{(\delta-1)}{\delta}} \tag{8}$$

Similarly, the isentropic efficiency of the turbine μ_T is defined as the ratio of the actual work output to the isentropic work output between same pressures. Neglecting change in kinetic energy we have,

Table 1. Summary of an overall average operating data for the SK30 GT power plant

S/N	Operating Parameters	Log sheet Data	Units
1	Inlet air temperature to compressor	29.35	°C
2	Outlet air temperature from compressor	278.31	°C
3	Inlet air pressure to compressor	0.1013	MPa
4	Outlet air pressure from compressor	0.65	MPa
5	Inlet temperature of turbine	792.02	°C
6	Turbine outlet temperature	457.35	°C
7	Exhaust gas pressure	0.1013	MPa
8	Mass flow rate of air through compressor, $\dot{m}_a$	82.1	Kg/s
9	Mass flow rate of fuel-gas	85.19	Kg/s
10	Isentropic efficiency of compressor	0.85	%
11	Isentropic efficiency of turbine	0.87	%
12	Low Heating Value of Fuel (LHV_f)	47321.5	KJ/kg

$$\mu_T = \frac{T_3 - T_4}{T_3 - T_{4s}} \tag{9}$$

The network output from the GT plant is given as:

$$w_{net} = w_t - w_c \tag{10}$$

The Specific fuel consumption (SFC) which compares the ratio of the fuel used by an engine to a certain force such as the amount of power the engine produces is given as:

$$SFC = {}^{3600}/_{AFR \cdot w_{net}} \tag{11}$$

Where the Air-Fuel Ratio (AFR) is given as:

$$AFR = LHV_f / Q_{sup} \tag{12}$$

Heat rate, HR which is the heat consumed to generate unit energy of electricity [20] can be determined by:

$$HR = SFC \cdot LHV_f \tag{13}$$

The thermal efficiency of the GT can be determined by the equation below:

$$\eta_{th} = \frac{w_{net}}{Q_{sup}} \tag{14}$$

2.5 Thermodynamic Analysis of GT Unit with an Inlet Air Cooler

Fig. 2 illustrates a schematic diagram of a GT power plant with an intake air cooler. In this study, the inlet air cooling technique proposed for analysis is absorption refrigeration cooling. The performance of the gas turbine will be evaluated with the absorption refrigeration cooling and compared to that of the simple GT setup.

The main advantage of the absorption system is that, it is independent of ambient air conditions. A simple sketch of the cooling coil is shown in Fig. 3. Ambient air enters the cooling coil at state 2 and leaves at state 3. Air passing over the outer surface of the coil experiences a drop in temperature and possibly a decrease in specific humidity, ω. The coil temperature can be adjusted to allow air to reach a certain desired temperature. In this case, the cooling load to be removed using cooling coil can be estimated.

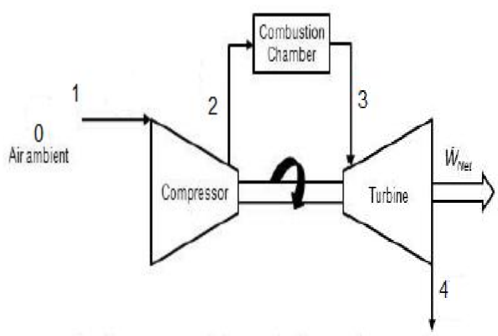

Fig. 2a. Simple GT Cycle [15]

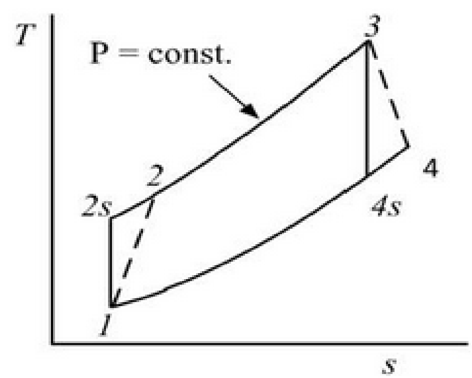

Fig. 2b. T-S Diagram of the GT Cycle [16]

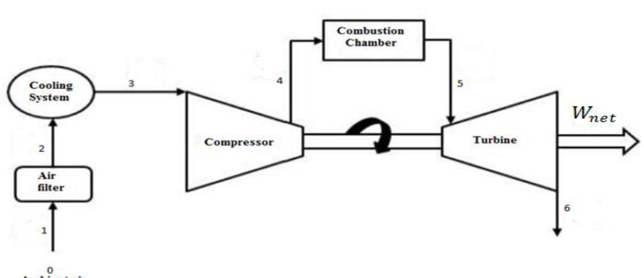

Fig. 2. Schematic diagram of the GT cycle with an inlet air cooler [15]

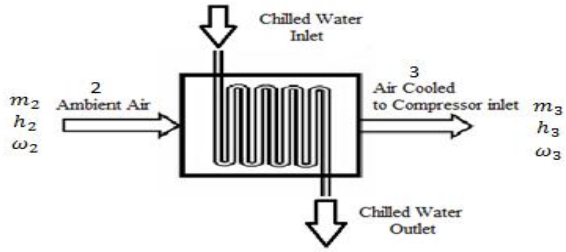

Fig. 3. Typical design of the absorption chiller cooling system

2.6 Dehumidification of Moist Air by Cooling

Whenever air is made to pass over a surface or through a spray of water that is at a temperature less than the dew point temperature of the air, condensation of some of the water vapour in air will occur simultaneously with the sensible cooling process. Fig. 4 shows a psychometric chart showing the cooling of moist air below its initial dew point temperature.

Firstly, the cooling follows a line of constant specific humidity to the saturation point; the water in the air begins to condense as dehumidification follows between point m-p. Point p represents the apparatus dew point temperature of the cooling coil. In a real process, the entire air passing through the surface of the cooling coil is not cooled to the surface temperature. When air mass, $\dot{m}_{a_2}$ passes over a coil, as seen in Fig. 5 some of it, $\dot{m}_{a,b}$ just by-passes unaffected while the remaining, $\dot{m}_{a_{dp}}$ 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 by-pass factor depends upon the pitch of the cooling coil fins, the number of rows in a coil in the direction of flow and the velocity of flow air. The final state (state 3), is then thought of as the adiabatic mixture of the bypassed air, which is at state 2, and the saturated air at state p.

The mass flow rate, $\dot{m}_{a,b}$, is thought of as bypassing the coil while $\dot{m}_{a,3}$ passes through the coil. The continuity Equation for dry air becomes:

$$\dot{m}_{a,2} = \dot{m}_{a,3} = \dot{m}_a$$
$$\dot{m}_{a,b} + \dot{m}_{adp} = \dot{m}_a$$

The *bypass factor* b_f, is defined as:

$$b_f = \frac{\dot{m}_{a,b}}{\dot{m}_a} \tag{15}$$

For the mixing process:

$$t_3 = \frac{\dot{m}_{a,dp}}{\dot{m}_a} t_{d,p} + \frac{\dot{m}_{a,b}}{\dot{m}_a} t_2 \tag{16}$$

$$\frac{\dot{m}_{a,b}}{\dot{m}_a} = \frac{t_3 - t_{d,p}}{t_2 - t_{d,p}} \tag{17}$$

$$b_f = \frac{t_3 - t_{d,p}}{t_2 - t_{d,p}} \tag{18}$$

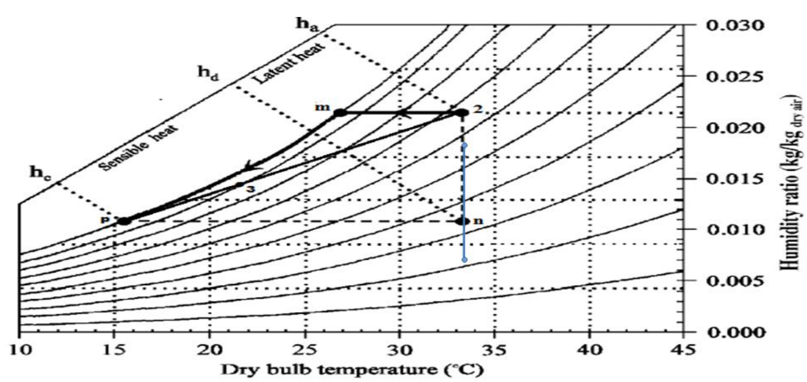

Fig. 4. Air cooling process on a psychometric chart [21]

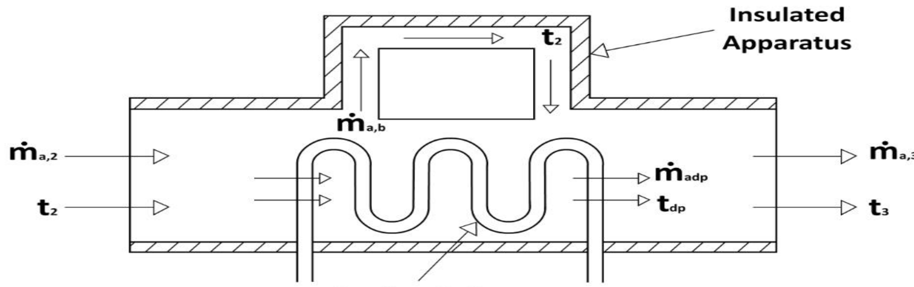

Fig. 5. Schematic representation of the bypass – factor Apparatus for a cooling/dehumidifying

Thus, the compressor inlet air temperature after cooling is given as:

$$t_3 = b_f(t_2 - t_{d,p}) + t_{d,p} \tag{19}$$

Where,

b_f= bypass Factor
t_1= dry bulb temperature of ambient air
$t_{d,p}$= Apparatus Dew Point Temperature

The achievable temperature is restricted only by the capacity of the chilling device to cause cooling and the ability of the coils to transfer heat. It is important to note that this intake air cooling method must be designed to avoid the formation of ice fragments on the compressor inlet or anywhere in the air intake structure. As a means of preventing icing, some authors have advised that the temperature drop should be greater than 5°C [22]. It is known that some water vapor condenses on the cooling coils; hence to avoid damage to the system, this condensate has to be eliminated by adding a separation system at the entrance of the air compressor.

The following assumptions were considered in the cooling coil model.

- The condition of the air leaving a chilled-water coil is nearly saturated, therefore, the relative humidity of the outlet air, $\emptyset_{out}$ from the coil can be assumed as 95 %.
- The pressure drop of air in the coil is 1% of the ambient air pressure.
- Cooling coil Bypass Factor (b_f) is 0.15

3. RESULTS AND DISCUSSION

The effect of the turbine compressor inlet temperature on the specific fuel consumption, Net work output produced and the thermal efficiency of the SK30 GT plant was obtained by the energy balance equations by utilizing MATLAB R2013a in the periods under study. In this study a single shaft gas turbine operating with a natural gas was numerically simulated. Table 1 which shows the summarized data with respect to the plant components was used for the compilation of Tables 2 and 3 respectively. Table 2 represents the performance of the plant without the ARS whereas values in Table 3 were generated after the ambient air has been cooled by the ARS before inhalation by the compressor. Figs. 6 and 7 clearly show the variations of the ambient air temperature with the efficiency of the plant. It is shown that the efficiency of a gas turbine plant is dependent on the compressor inlet air temperature. This implies that lower compressor work produces higher net-work output. The effect of inlet cooling as shown in Figs. 6 and 7 is that the ARS inlet cooling techniques was able to achieve a temperature

Table 2. Turbine parameter before compressor inlet cooling

Year	T_1	T_2	W_C	W_t	W_{net}	AFR	Q_{Sup}	SFC	HR	η_{th}
2005	301.48	550.05	249.81	395.82	146.01	79.91	592.22	0.3086	14603.41	24.65
2006	301.79	550.61	250.07	395.82	145.75	79.99	591.57	0.3088	14612.88	24.64
2007	302.10	551.22	250.34	395.82	145.48	80.09	590.87	0.309	14622.34	24.62
2008	302.53	551.96	250.68	395.82	145.14	80.2	590.01	0.3093	14636.54	24.59
2009	303.02	552.86	251.09	395.82	144.73	80.34	588.99	0.3096	14650.74	24.57
2010	303.17	553.13	251.21	395.82	144.61	80.39	588.67	0.3097	14655.47	24.56

Table 3. Turbine parameter after compressor inlet cooling

Year	T_1	T_3	T_4	W_C	W_t	W_{net}	AFR	Q_{Sup}	SFC	HR	η_{th}
2005	301.48	287.47	524.49	238.2	395.82	157.6	76.13	621.61	0.3000	14196.45	25.63
2006	301.79	287.52	524.58	238.24	395.82	157.6	76.14	621.51	0.3001	14201.18	25.35
2007	302.1	287.57	524.67	238.28	395.82	157.5	76.15	621.41	0.3001	14201.18	25.35
2008	302.53	287.63	524.78	238.33	395.82	157.5	76.17	621.28	0.3001	14201.18	25.35
2009	303.02	287.7	524.91	238.4	395.82	157.4	76.19	621.12	0.3002	14205.91	25.35
2010	303.17	287.73	524.95	238.41	395.82	157.4	76.19	621.08	0.3002	14205.91	25.34

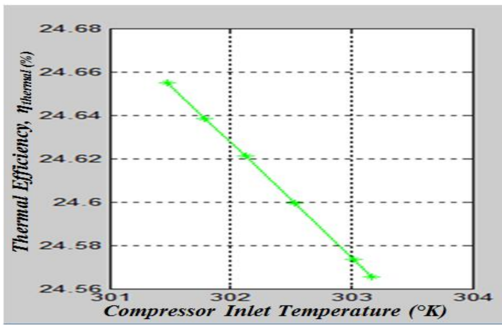

Fig. 6. Before chilling

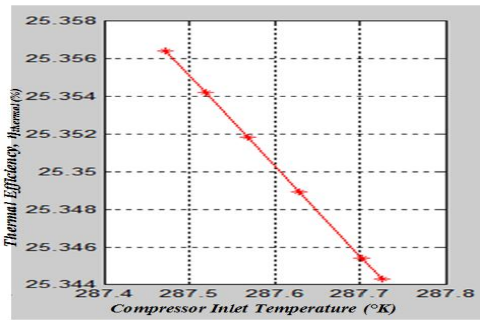

Fig. 7. After chilling

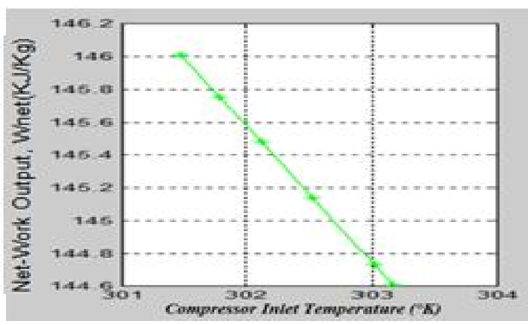

Fig. 8. Before cooling

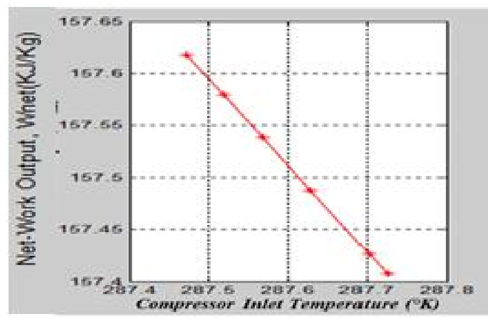

Fig. 9. After cooling

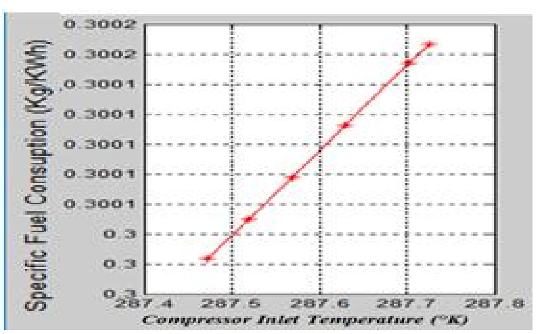

Fig. 10. Before cooling

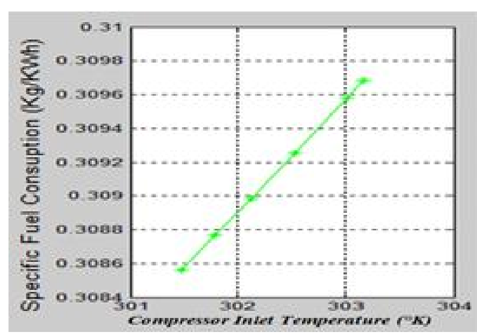

Fig. 11. After cooling

reduction of 25.32^{0}K and 15.44^{0}K in the years 2009 and 2010 respectively. This in turn corresponds to efficiency gain of 25.34% which is an increase of the plant efficiency by about 7%.

Figs. 8 and 9 depict plots of Net-Work Output against the compressor inlet temperature. Mathematically, the plots show that the Net-Work Output is inversely proportional to the compressor inlet temperature. As a comparison of Figs. 7 and 8, we can say that an increment of approximately 1% in the compressor air inlet temperature decreases the gas turbine power output by 7%. Also, as illustrated in Fig. 9, a maximum gain in the Net-Work Output from 144.61 KJ/Kg to 157.40 KJ/Kg; 144.73 KJ/Kg to 157.40 KJ/Kg with the application of ARS was observed in the years 2009 and 2010 where drops in temperature are 15.32 ^{0}K and 15.44 ^{0}K respectively. These drops in temperatures in the years 2009 and 2010 lead to about 8.4% gain in Net-Work Output.

Figs. 10 and Fig. 11 show plots of specific fuel consumption against the compressor inlet air temperature. It also shows that the specific fuel consumption increases with increase in ambient temperature. This according to Rahman et al. [20] occurs because of increased losses due to the increased amount of flue gases.

4. CONCLUSION

Influence of compressor inlet air cooling on the performance of a gas turbine power plant was investigated with vapour absorption refrigeration. The study was done on two similar gas turbine plants, one without Inlet cooling and one with inlet cooling with the aid of MATLAB R2013a. The results showed that:

1. The thermal efficiency and net work output increase while the specific fuel consumption decreases with decrease in ambient temperature.
2. Absorption refrigeration system improved the performance of gas turbine power plant as the thermal efficiency and network output are higher than that obtained from simple gas turbine plant.

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COMPETING INTERESTS

Authors have declared that no competing interests exist.

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