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Effects of Evaporative Cooling on the Performance of a Gas Turbine Plant Operating in Bayelsa State, Nigeria

Robert Poku, Ezenwa A. Ogbonnaya
Department of Mechanical/Marine Engineering,
Niger Delta University, P.M.B. 071 Wilberforce Island,
Bayelsa State-Nigeria.

ABSTRACT

A location where the demand for electricity is very high with warm weather, Performance of gas turbine plants is greatly affected. Therefore, to enhance the performance, inlet air cooling becomes necessary. This paper takes an in-depth study of the influence of an evaporative cooler on turbine inlet air and how the turbine responds to the reduced ambient temperature. In this study, a set of actual operational conditions such as ambient temperature and relative humidity were employed to determine the compressor work, turbine net work, specific fuel consumption and the thermal efficiency of the plant. A GT plant located in Bayelsa State, Nigeria that generates electricity was used as a case study for this research. The results showed that as the ambient temperature was lowered, the compressor work reduces, turbine network increases, specific fuel consumption reduces while the efficiency of the plant increases. It was observed that the effect of the evaporative cooling is that a drop in ambient temperature of 2.4°C leads to an increase of about 0.14% and 2.02kJ/kg in efficiency and network of turbine respectively and also, 0.002kg/KWh drop in specific fuel consumption. The research shows that gas turbine plants perform better in temperate regions than tropical. Therefore, to increase the performance of an existing gas turbine plants in high temperature climates, retrofitting an air cooler that will always reduce the temperature back or close to the design temperature before compression is necessary.

Keywords: *Gas Turbine, Thermal Efficiency, Psychometric Chart, Wet Bulb Temperature, Evaporative Cooling, Combustion Turbine Inlet Air Cooling (CTIAC).*

Nomenclature

C_{pa} = *Specific Heat Capacity (KJ/KgK)*
$\eta_{thermal}$ =*Thermal Efficiency*
W_N = *Turbine NetWork (KJ/Kg)*
r_p = *Pressure ratio*
W_c = *Compressor Work (KJ/kg)*
k = *Isentropic index*
SFR = *Specific Fuel Ratio (kg/KWh)*
p = *Pressure (bar)*
η_{cs} =*Isentropic Efficiency of Compressor(KJ/Kg)*
T = *Temperature (^0C)*
C_{pg} = *Specific Capacity of the Gas(KJ/KgK)*
Q_{add} = *Heat Added (KJ/Kg)*
m_a = *Mass of the air (Kg/s)*
m_g = *Mass of the gas (Kg/s)*
GT = *GAS Turbine*
C_{pa} = *Specific Capacity of the air(KJ/KgK)*
Δp = *Pressure drop in the combustion chamber*
η_m = *Mechanical Transmission Efficiency*
LHV = *Lower Heat Value (kJ/kg)*
AFR = *Air Fuel Ratio*

1. INTRODUCTION

Bayelsa State is located in the very heart of Niger Delta in Nigeria. It experiences the equatorial type of climate in the southern coastal area and subequatorial in the Northern region. The temperature ranges between 25°C and 28°C with relative humidity of between 70 to 90% [1]. Its major source of electricity is a GT plant. The plant, as in any other GT is made up of compressor, combustion chamber and the turbine. The compressor takes in air, compresses it to a higher pressure which is then supplied to the combustion chamber. Increased air density results to improve output of GT and air density is a function of ambient temperature [2], [3], [4].

However, due to the location of Bayelsa State and other regions in similar locations, the environmental conditions needed for the plant operation differ substantially from the ISO conditions which are temperature of 15°C and relative humidity of 60% [4]. This difference largely affects the output of the plant such as the thermal efficiency, turbine net work, fuel consumption. This is because the power output is inversely proportional to ambient temperature. It is shown that drop in temperature leads to an augmentation in the air density and consequently increase air mass flow rate and this in turn results to increase in power output and efficiency of about 0.7% per degree Celsius for heavy duty gas turbine [5].

In other to achieve efficient and optimum performance, it became necessary to employ technologies which will make it possible for the plant to improve in its performance. This has led to several researches such as the use of chiller in the inlet system to reduce compressor inlet air and subsequently increase the plant output. [6] work showed that 1% of the output power is possible as the inlet temperature drops by 1.6% with the application of water chiller.

A number of research works have compare different options for inlet air cooling and this are: Evaporative cooling, fogging, mechanical refrigeration (direct and indirect), mechanical refrigeration with ice storage, mechanical refrigeration with chilled water storage, single stage lithium bromide absorption chiller and two stage lithium bromide absorption chiller. In all these technologies, the aim is to reduce turbine inlet temperature and the result is the subsequent increase in turbine performance [2] . Beyond the turbine performance, other important benefits of inlet air cooling are reduction in the emission of COx, SOx and NOx. A sure way of reducing COx and SOx generation into the atmosphere is to reduce the quantity of fuel burnt [6] and one of the ways of achieving this is reduced compressor inlet temperature [7], [8] .This paper, therefore, seek to explore further the thermodynamic advantages and the inherent environmental benefits of a gas turbine plant fitted with an evaporative cooling which is operating in a tropical region like Nigeria and then make a comparison with the data generated from a gas turbine plant that is not fitted with an inlet temperature cooler and operating in the same region where all conditions applied to both plants. The GT plants being considered is shown as fig. 1. The technology employed in this paper to achieve the lower compressor inlet temperature is the evaporative cooling system.

A. Evaporative Cooling

This technology is the most prevalent of all known CTIACs owing to its numerous benefits such as lowest capital cost, lowest operation and maintenance cost, the water used for the operation could be applied raw, it serves as an air washer and cleans the compressor inlet air, and the delivery and installing time faster than all other techniques and its ability to reduce NOx emission by 0.8-1.5% per °C of cooling [9]. However, [10] showed that the amount of water required for evaporative cooling depends on the inlet air flow, temperature, pressure and humidity of the ambient air, the hardness of the water, degree of cooling required and turbine mass flow rate. Fig 1. Shows the working of the evaporative cooling techniques. Warm inlet filtered air passes through a saturated wetted media and then part of the water gains latent heat, evaporates and the air loses sensible heat and its dry bulb temperature decreases with consequent increase in mass density. Excess water that does not evaporate is channeled downward so as not to be carried along with the cooled air.

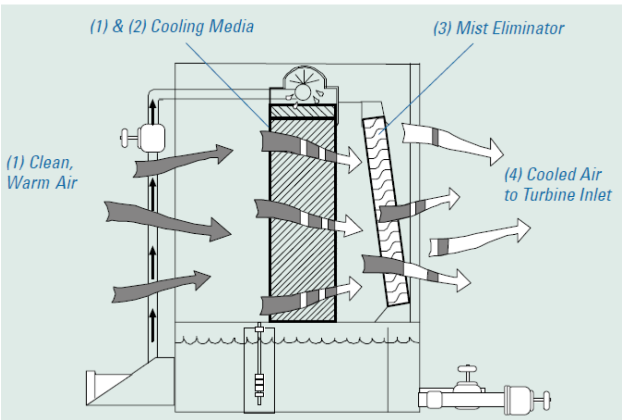

Fig. 1. Illustrating the working of evaporative cooling [11]

The cooled air then passes through the integral mist eliminator, where leftover water droplets are removed. This result to higher mass flow of air that goes into the compressor and gives the turbine higher output.

Fig. 2. gives a clearer illustration of the application of an evaporative cooling system and the subsequent supply of the cooled air to the turbine.

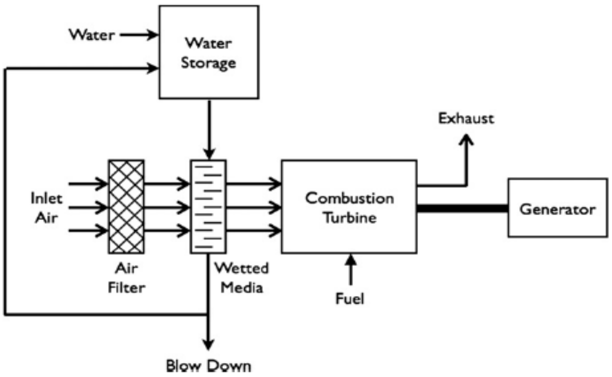

Fig. 2. CTIAC using evaporative cooling over wetted media [6]

1. MATERIALS AND METHODS

For a simple GT plant with configuration of a Brayton's cycle, the power output is the difference in the turbine work and the compressor work. It then imply therefore, that net work of the turbine depends to a large extent on the compressor output. More so, The power consumed in the compressor is directly proportional to the inlet temperature, that is, if the compressor inlet temperature is reduced, mass density of air taken into the turbine is increased and this in turn affects the plant performance [10]. For this very importance role turbine inlet air plays, it becomes so pertinent to study the plant operating environment.

The data for this study were obtained from an operational Rolls Royce, Industrial Olympus-SK 30 gas turbine plant located at Imiringi, Bayelsa State, Nigeria. The plant is the major source of electricity in the State. The parameters used for this work were obtained from the logsheet over a period of seven years. However, where certain data could not be sourced, standard thermodynamic values were used. In other to make the data workable, statistic is used to arrive at the values used for the studies. This is done by computing the average of the daily, weekly, monthly and then yearly turbine readings. These procedures are repeated for the years covered by this work. And finally the average of the seven years values was computed.

Modeling and simulation of each of the plant component was done. Performance of the plant without an evaporative cooling was compared with one that incorporates evaporative cooling system. The result of the two systems were thereafter studied and compared.

A summary of the processes carried out in order to arrive at the values used for the computations is shown in table 1. This table shows the actual values from the plant's log sheets and the calculated values.

Fig. 3 shows the diagram of the plant. In other to arrive at a better understanding of this work, thermodynamic equations are derived which are then used for subsequent calculations. The reasons for deriving the thermodynamic equations are the thermal efficiency and its relationship with parameters such as the temperatures and pressures ratios. Fig. 3 is a schematic diagram of a simple gas turbine plant that consists of an inlet air filter, evaporative cooler, compressor, combustion chamber and a turbine.

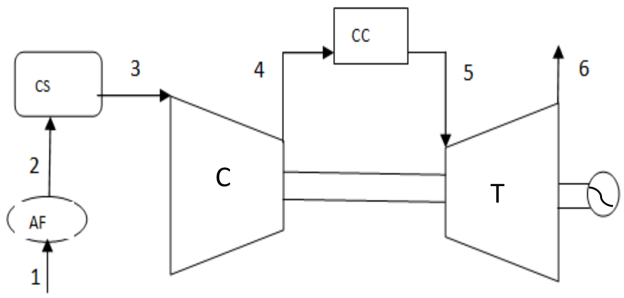

Fig. 3. Schematic of the Gas Turbine Plant

The working fluid passing through the compressor is air and is taken to be ideal gas while the working fluid through the turbine is the flue gas from the combustion chamber. The compressor inlet air after the cooling is

$$T_3 = Tb_2 - \varepsilon(Tb_2 - Tw_2) \qquad (1)$$

It is believed that the integral mist eliminator has removed any water that would have been taken along with the inlet air to the compressor. Therefore, water carry-over in the cooled air is neglected. Also, the relative humidity and the ambient temperature, Tb_2 for Niger Delta areas of Nigeria, which include Bayelsa State were studied to be 84.12% and 26.9°C respectively [12]. The wet bulb temperature is obtained from Psychometric chart.

The compression process is, 3-4

$$T_4/T_3 = P_4/P_3 = (r_p)^{\frac{k-1}{k}}$$

$$T_4 = T_3(r_p)^{\frac{k-1}{k}} \qquad (2)$$

The expansion process is, 5-6

$$T_5/T_6 = P_5/P_6 = (r_p)^{\frac{k-1}{k}}$$

For isentropic expansion process,

$$T_{6s}/T_5 = (P_6/P_5)^{\frac{k-1}{k}}$$

$$W_t = m_g c_{pg} T_5 \eta_t \left[1 - \frac{1}{\left(\frac{P_5}{P_6}\right)^{\frac{k-1}{k}}}\right] \qquad (7)$$

$$T_5 = T_6(r_p)^{\frac{k-1}{k}} \qquad (3)$$

The compressor work of the gas turbine is:

$$W_c = m_a c_p(T_4 - T_3)$$

Substituting equation 1 for T_4 above gives,

$$W_c = m_a c_{pa} T_3\left(r_p^{\frac{k-1}{k}} - 1\right) \qquad (4)$$

The turbine work of the gas turbine is:

$$W_t = m_g c_{pg}(T_5 - T_6)$$

Substituting equation 3 for T_5 gives,

$$W_t = m_g c_{pg} T_6\left(r_p^{\frac{k-1}{k}} - 1\right) \qquad (5)$$

The isentropic efficiency of the compressor is:

$$\eta_{cs} = \frac{Isentropic\ work}{Actual\ work}$$

$$\eta_{cs} = \frac{(T_4' - T_3)}{(T_4 - T_3)}$$

For an isentropic compression process,

$$T_{4s}/T_3 = (P_4/P_3)^{\frac{k-1}{k}}$$

The isentropic efficiency of the turbine is

$$\eta_{ts} = \frac{Actual\ work}{Isentropic\ work}$$

$$\eta_{ts} = \frac{T_5 - T_6}{T_5 - T_{6s}}$$

Therefore, the actual work required to drive the compressor becomes:

$$W_c = \frac{m_a c_{pa} T_3}{\eta_m \eta_{cs}}\left[\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}} - 1\right] \qquad (6)$$

The actual turbine work is therefore,

The net power from the GT plant is

$$W_N = W_t - W_c \qquad (8)$$

Heat supplied by the fuel in the combustion chamber is:

$$Q_{add} = m_a c_{pg}(T_5 - T_4) \qquad (9)$$

The specific fuel consumption as determined by equation 10 compares the ratio of the fuel used by an engine to the amount of power the engine produces.

$$SFC = \frac{3600}{AFR \times W_N} \quad (10)$$

The air-fuel ratio is obtained from equation 11 as

$$AFR = \frac{LHV_f}{Q_{add}} \quad (11)$$

Where the thermal efficiency of the plant can be determined as:

$$\eta_{therm} = \frac{W_N}{Q_{add}} \quad (12)$$

The heat rate (HR) of the GT power plant can be determined as as the inverse of the thermal efficiency [13]:

$$HR = \frac{1}{\eta_{therm}} \quad (13)$$

Data used for the computation are presented in table 1. The parameters that could not be sourced were calculated from thermodynamic formulae.

Table 1. Summary of overall average of the working parameters (from 2002 to 2008).

Components	Parameters	Units	Values From Logsheet
Compressor	*Inlet Temperature, T_3*	*K*	*300.05*
	Outlet Temperature, T_4	*K*	*509.08*
	Inlet Pressure, P_3	*Bar*	*1.013*
	Outlet Pressure, P_4	*Bar*	*6.43*
	Mass flow rate, m_a	*Kg/s*	*82.14*
	Isentropic Efficiency compressor, η_{cs}	*%*	*0.85*
	Cooler Efficiency, ε	*%*	*0.90*
Combustion Chamber	*Inlet Temperature, T_4*	*K*	*509.08*
	Maximum Temperature, T_5	*K*	*1055.40*
	Inlet Pressure, P_4	*Bar*	*6.43*
	Outlet Pressure, P_5	*Bar*	*6.30*
	Mass flow rate of fuel, m_g	*Kg/s*	*3.05*
Turbine	*Inlet Temperature, T_5*	*K*	*1055.40*
	Outlet Temperature, T_6	*K*	*668*
	Inlet Pressure, P_5	*Bar*	*6.30*
	Outlet Pressure, P_6	*Bar*	*1.013*
	Mass flow rate, m_g	*Kg/s*	*85.19*
	Isentropic Efficieny of Turbine, η_{ts}	*%*	*0.87*
Exhaust	*Exhaust gases temperature, T_6*	*K*	*688*
	Exhaust gases pressure, P_6	*Bar*	*1.013*
	Mass flow rate, m_g	*Kg/s*	*85.19*

2. RESULTS AND DISCUSSION

The parameters relevant for this study are actually the turbine compressor inlet temperature from the evaporative cooler, temperature at the end of the compression process, specific fuel consumption, net turbine work produced, and the thermal efficiencies of the plant calculated in the periods under study. Table 1 shows the summarized data with respect to the plant components used for the compilation of tables 2 and 3 respectively. Table 2 represents the performance of the plant without the evaporative cooler whereas, table 3 parameters were generated after the ambient air has been made to pass through the compressor inlet cooling process.

The vital difference in the two tables as can be seen from a thorough observation is that Tb2 in table 2 is the ambient air temperature that goes into the turbine. While in table 3, though Tb2 represent the ambient air temperature, T3 is the temperature that actual goes in to the turbine after the cooling process.

Table 2:*T*urbine parameters before compressor inlet cooling

YEAR	*Tb2*	*T4*	*Wc*	*Wt*	*AFR*	*SFC*	*Qadd*	*WN*	*ηtherm*
2002	*300.05*	*509.39*	*250.03*	*386.30*	*75.27*	*0.351*	*626.26*	*136.27*	*21.76*
2003	*300.55*	*510.24*	*250.45*	*386.30*	*75.39*	*0.351*	*625.29*	*135.85*	*21.73*
2004	*301.05*	*511.09*	*250.86*	*386.30*	*75.51*	*0.352*	*624.32*	*135.44*	*21.69*

2005	301.55	511.94	251.28	386.30	75.63	0.356	623.34	135.02	21.66
2006	302.05	512.79	251.70	386.30	75.74	0.353	622.37	134.60	21.63
2007	302.55	513.64	252.11	386.30	75.86	0.354	621.40	134.19	21.59
2008	303.05	514.49	252.53	386.30	75.98	0.354	620.42	133.77	21.56

Table 3:*T*urbine parameters after compressor inlet cooling

YEAR	Tb2	T3	T4	Wc	Wt	AFR	SFC	Qadd	WN	ηtherm
2002	300.05	297.62	504.97	248.01	386.30	74.67	0.349	631.34	138.29	21.90
2003	300.55	298.39	506.28	248.65	386.30	74.85	0.349	629.84	137.65	21.86
2004	301.05	298.62	506.67	248.84	386.30	74.90	0.350	629.39	137.46	21.84
2005	301.55	299.44	508.06	249.52	386.30	75.09	0.351	627.80	136.78	21.79
2006	302.05	299.67	508.45	249.72	386.30	75.14	0.351	627.35	136.58	21.77
2007	302.55	300.44	509.76	250.36	386.30	75.32	0.352	625.85	135.94	21.72
2008	303.05	301.03	510.76	250.85	386.30	75.46	0.352	624.70	135.45	21.68

A. Discussion

Figure 4 clearly shows the variations of the ambient temperature with the efficiency of the plant. It shows that the efficiency of a gas turbine plant and the environmental temperatures are inversely proportional. Fig. 5 also, illustrates the Influence of compressor inlet temperatures on the efficiencies of a GT plant. But as can be seen, the effect of turbine inlet temperature is such that a decrease of ambient temperature of 2.4°C does results to efficiency increase of about 0.14%. This shows that the lower the turbine inlet temperature the higher the efficiency of the plant obtained [14]

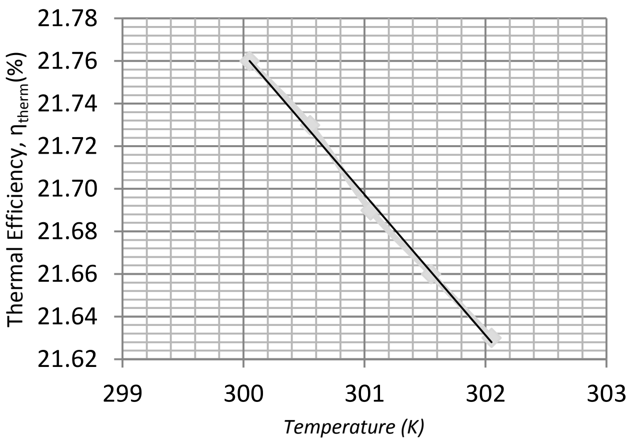

Fig. 4: Thermal Efficiency Vs Ambient Temperature (K) before Cooling

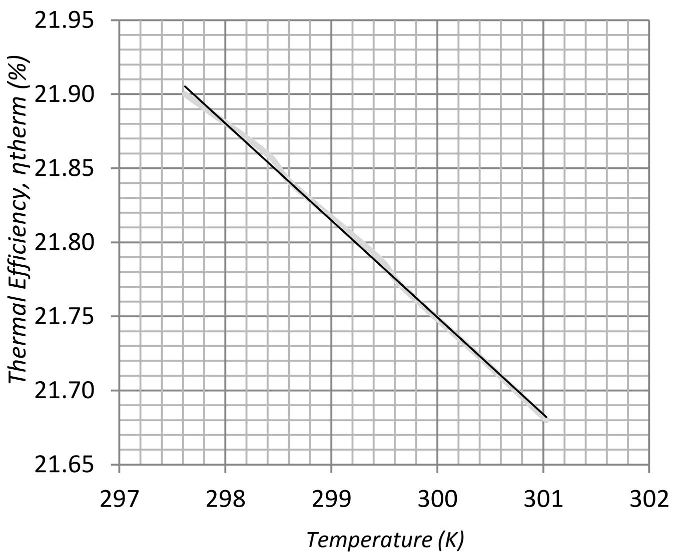

Fig.5: Thermal Effiency Vs Cooled Temperature (K) to Compressor after Eveporative Cooling

Fig. 6 and fig. 7 demonstrate that the environmental temperature does have a great influence on the specific fuel consumption of a plant. In the two graphs, it is seen that although the specific fuel consumption is in direct proportion with increase in ambient temperature, it could be made less with decrease turbine inlet temperature. Comparison of both tables proves that a temperature reduction of 2.4°C produces a drop in the specific fuel consumption by 0.002kg/KWh. The implication of this is that since the specific fuel consumption compares the ratio of fuel used by an engine to the power produced [15]. The less this value, more power (net work) is produced with less fuel consumption as demonstrated in fig 10 and fig 11.

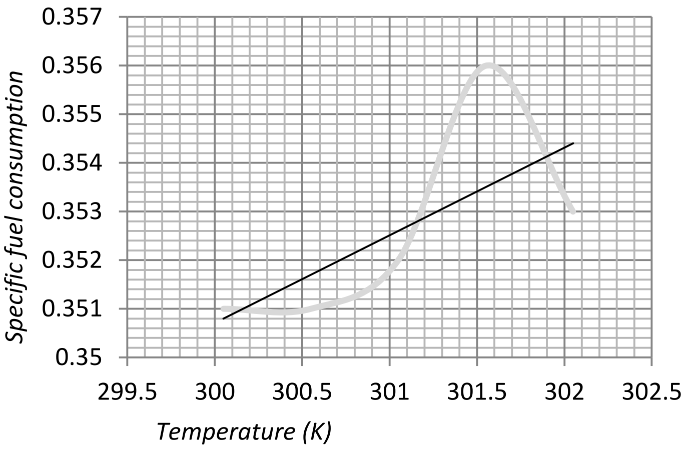

Fig 6. Specific Fuel Consumption Vs Compressor Inlet Temperature (K) before cooled

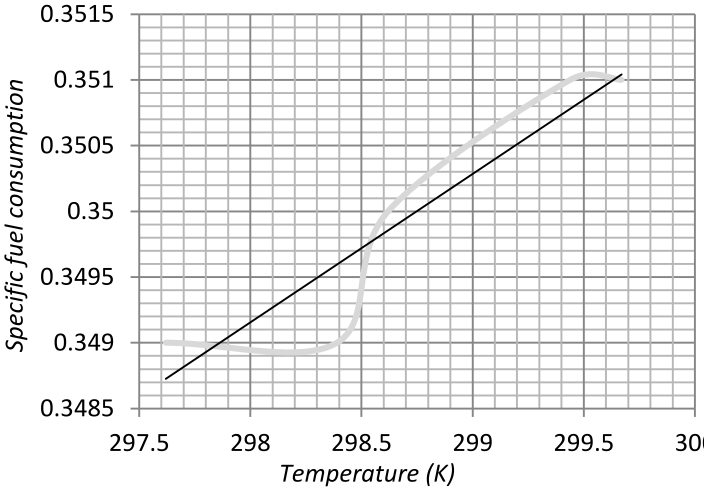

Fig. 7. Specific Fuel Consumption Vs Cooled Temperature (K) to Compressor after Evaporative Cooler.

The effect of ambient temperature on the power output of the plant is illustrated with fig. 8 and fig. 9. The two figures showed that the power output (net work) gradually increase with lower turbine inlet temperature. However, the benefit of the application of the evaporative cooler is such that a decrease in the ambient temperature could lead to a gain of 2.02KW power output.

The simple explanation to this is, when the air temperature is lowered, its density increases which leads to lower volume of air required for same mass handled. The result of this is a decline in the compressor specific work. Therefore, if the peak temperature of the turbine and its pressure ratio remain constant, then the only variable parameter due to the cooling becomes the compressor [16]. This mains that lower compressor work produces higher net work

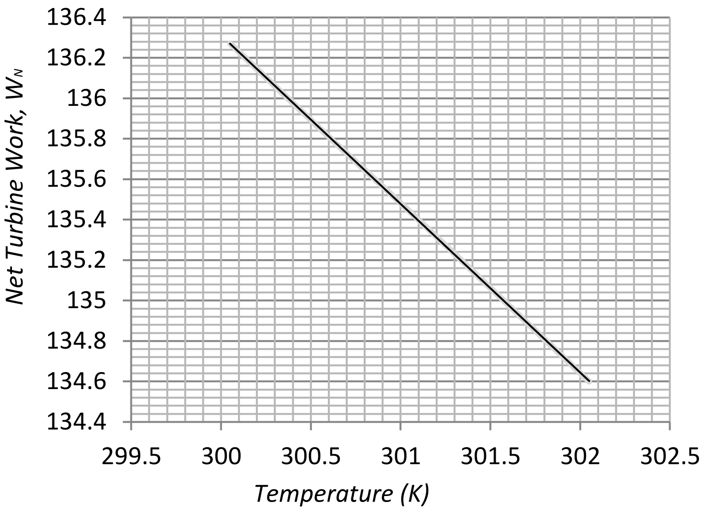

Fig. 8. Net Turbine Work Vs Compressor Inlet Temperature (K) before cooled

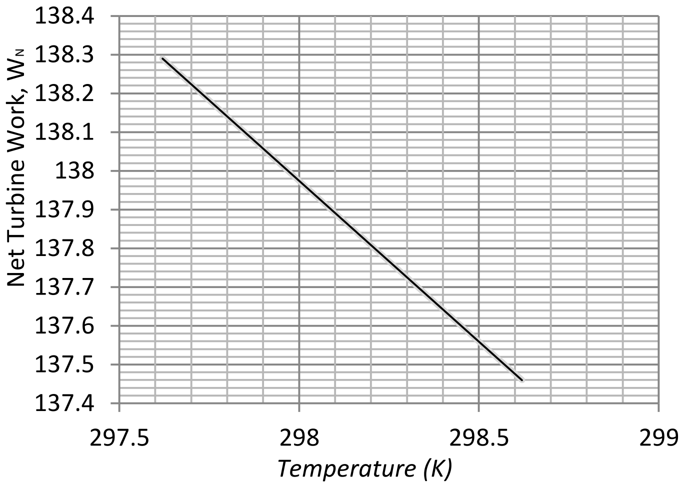

Fig. 9. Net Turbine Work Vs Cooled Temperature (K) to Compressor after Evaporative Cooler

B. Conclusion

A gas turbine power plant was studied so as to have a clearer understanding on how its performance responds to environmental variations. This study is very vital as to efficiently understand

the effects of locations on the performance of the turbine. The parameters considered in this paper are the ambient temperatures, work done by compressor, combustion chamber inlet temperature, turbine net work and the thermal efficiency of the plant. The study of the performance parameters showed that:

i. As the environmental temperature decreases, the density of the air taken into the compressor increases and this causes an increase in the air mass flow rate. Also, less work is required from the turbine to drive the compressor.

These all contributes to the increase in efficiency and the power output of the turbine.

ii. The gas turbine plant performance improves with small but continual rise in specific fuel consumption and air-fuel ratio. This suggests that as the compressor inlet air temperature decreases, fuel consumed by the turbine reduces

iii. Since it has been proved that decrease in compressor inlet temperature reduces CO_2 and NOx emission, evaporative cooling can be an alternative to mitigating air pollutants from gas turbine emission.

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REFERENCES

[1] Odjugo, P. A. O. (2011). Perception of Climate Change in the Niger Delta Area of Nigeria. In A. G. Onorkerhoraye (Ed.), *Centre for population and Environmental Development Policy Paper Series* (pp. 1-14). Benin: Centre for Population and Environmental Development.

[2] Omidvar, B. (2001). Gas Turbine inlet Cooling System. Retrieved 7 January, 2014 from the World Wide Web: *http://www.albadronline.com/oldsite/books/49_GasTurbineInlet.pdf*

[3] Ogbonnaya, E. A. (2004). Thermodynamics of Steam and Gas Turbine, Oru's Press Ltd, Port Harcourt, pp. 68-71.

[4] Cengel, Y.A. and Boles, A. B. (2006). Thermodynamics: An Engineering Approach; Fifth Edition. McGraw-Hill Companies, Inc., 1221 Advance of the Americans, New York NY 10020.p.584.

[5] Ogbonnaya, E.A., Poku, R. and Adigio, E. (2013). Effects of Greenshipping to the Maritime Industry. Engineering Technology and Applied Science Research (ETASR). Vol. 3, No. 2, pp. 402-407.

[6] Poku, R. (2013). Thermodynamics Performance Evaluation of a Two-Shaft Gas Turbine Power Plant. *International Journal of Engineering Research and Technology (IJERT)*, Vol.3, No. 1: 842-848.

[7] Santos, A. P. and Andrade, C. R (2012). Analysis of Gas Turbine Performance with Inlet Air Cooling Techniques Applied to Brazilian Sites. Retrieved 17 December, 2013 from the World Wide Web: *http://www.jatm.com.br/ojs/index.php/jatm/article/viewFile/175/248*

[8] Ramsey, S. H. (2010). Climate Change- Driven Market Opportunities for Combined-Cycle Gas Turbines. Retrieved 3 January, 2014 from The World Wide Web:

[9] Gupter, A., Prakash, O. and Shukla, S. K. (2011). Thermodynamic Analysis of Gas-Steam Combined Cycle with Carbon Dioxide Emissions Saving. Retrieved 20 December, 2013 from the World Wide Web: *http://www.ijee.ieefoundation.org/vol2/issue2/IJEE_03_v2n2.pdf*.

[10] Hart, H. I. (1998). *Thermodynamic Appraisal of Gas Turbine Performance in the Niger Delta*. Unpublished Ph.D's Dissertation, Mechanical Engineering, University of Nigeria, Nsuka, Nigeria.

[11] Al-Ibrahim, A. M. and Varnham, A. (2010). A Review of Inlet- Cooling Technologies for Enhancing the Performance of Combustion Turbine in Saudi Arabia. Retrieved 2 January, 2014 from the World Wide Web: *http://ipac.kacst.edu.sa/eDoc/2010/190977_1.pdf*.

[12] Donaldson (2005). Filtration Solutions for Gas Turbines, Generator and Compressors. Retrieved 7 January, 2014 from the World Wide Web: *http://www.donaldson.com/en/gasturbine/support/datalibrary/000586.pdf*.

[13] Akpan, I. E. (2000). *Analysis of Psychometric Data for Niger Delta Region of Nigeria*. Unpublished Master's Thesis, University of Port Harcourt, Port Harcourt, Nigeria.

[14] Kurz, R. (2005). Gas Turbine Performance. Retrieved 2 January, 2014 from The World Wide Web: *http://turbolab.tamu.edu/proc/turboproc/T34/t34-14.pdf*.

[15] Ibrahim, T. K., Rahman, M. M. and Abdalla, A. N. (2010). Improvement of gas turbine performance based on inlet air cooling system: A technical review. *International Journal of Physical Sciences*, Vol. 6, No. 4: pp. 620-627.

[16] Abam, F. I., Ugot, I. U. and Igbong, D. I. (2012). Performance Analysis and Component Irreversibilities of a (25MW) Gas Turbine Power Plant Modeled with a Spray Cooler. *American Journal of Engineering and Applied Science*, Vol. 5, No. 1: pp. 35-34

