IOSR Journal of Engineering (IOSRJEN) **www.iosrjen.org**
ISSN (e): 2250-3021, ISSN (p): 2278-8719
Vol. 05, Issue 03 (March. 2015), ||V2|| PP 59-69

Thermo-Economic Analysis of Evaporative Cooling in a Gas Turbine Plant in Niger Delta, Nigeria

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

Abstract: - Gas Turbine power plants are widely applied for power generation in Nigeria. Due to the geographical location of Nigeria, the international organization for standardization, ISO conditions of 15^oC and relative humidity of 60% for an ideal GT operation rarely occur. This variance of the environmental temperature results to drop in thermal efficiency and power output of the power plants. One of the most important ways of improving the performance of GT is the application of GT inlet cooling technology where the temperature of the air entering the compressor is brought lower than the prevailing ambient temperature. This paper, therefore, presents the results of the study of the thermal analysis and the economic benefits derived from the incorporation of an evaporative cooling technique in a Rolls Royce, Industrial Olympus-SK 30 GT plant located at Imiringi, Southern Nigeria. Data generated from the power station were used for both the thermal and economic analysis. The analysis shows that reducing the plant inlet temperature by 2^oC could lead to power gain of 2.02MW and increase of plant efficiency from 21.76% to 21.90%. The economic analysis also indicates that the total saving resulting from the application of the evaporative cooling depends on the power gained which is evident in the years 2004 when the power gain was 2.02MW with total saving resulting from cooling of US$21136.76 . However, in terms of the net profit and the fuel saving per annum, the plant seems to show a loss in profit owing to the low power being generated by it but the economic viability would be seen in the long run with the fact that the higher the power output, the greater the fuel saving per annum.

Keywords: *Power Gained, Thermal Efficiency, Dry Bulb Temperature, Evaporative Cooling, Net Profit, Total Saving, Gas Turbine, Psychometric Chart*

NOMENCLATURE: *C_{pa} = Specific Heat Capacity (kJ/kgK),□ □ □$_{thermal}$=Thermal Efficiency, W_N = Turbine NetWork (kJ/kg), r_p = Pressure ratio, Wc = 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), T_3= Temperature after Cooling, C_{pg} = Specific heat Capacity of the Gas (kJ/kgK), Q_{add} = Heat Added (kJ/kg), m_a = Mass of the Air (kg/s), GT= Gas Turbine, m_g = Mass of the Gas (kg/s), GT = GAS Turbine, C_{pa} = Specific heat Capacity of the air(kJ/kgK), ρ_w =Water Density (kg/m^3),ΔP = Power Gained, □$_m$ = Mechanical Transmission Efficiency, ε= Cooler Effectiveness, p_a =Air Density (kg/m^3)
LHV = Lower Calorific Value (kJ/kg), AFR = Air Fuel Ratio, □$_1$= Plant Efficiency before Cooling, w_1 and w_2 = Humidity Ratios, □$_2$= Plant Efficiency after Cooling, T_{b2}= Dry Bulb Temperature before Cooling, T_{w2}= Wet Bulb Temperature (^oC)

I. INTRODUCTION

The Niger Delta area of Nigeria consists of nine states of the federation that span the South and the fringes of the South-West and South-East of Nigeria. The states are Rivers, Bayelsa, Edo, Delta, Akwa Ibom, Cross Rivers, Ondo, Imo and Abia [1]. The states which experience the equatorial temperature of 25^oC to 28^oC and relative humidity of 70 to 90% are in the Southern coastal area of the country [2]. Among the different kinds of power plants, the most widely used source of electricity in this area is the GT plant. This is due to feature low capital cost to power ratio, high flexibility, high reliability without complexity, compactness, early commissioning and commercial operation and fast starting–accelerating and quick shut down. The GT is further recognized for its good environmental performance, manifested in the low environmental pollution [3]. GT plants depend on air to operate. Therefore, rise in air density leads to an increase GT output as air density is influenced by ambient temperature [4], [5].

Nonetheless, as a consequence of the geographical location of the Niger Delta area, the air conditions required for the plant operation vary considerably from the ISO conditions which are temperature of 15°C and relative humidity of 60% [5]. This difference largely affects the output of the plant such as the thermal efficiency, turbine net work, fuel consumption. The reason is that the power output is inversely proportional to ambient temperature [6]. Kakaras [7] reported that the GT power output and efficiency are strong functions of the ambient air temperature. [8] in a study on micro GT plants showed that when the ambient air temperature went up, the electrical efficiency decreased. Depending on the GT type, power output is reduced by a percentage between 5 to 10 percent of the ISO-rated power out for every 10K increase in ambient air temperature. At the same time the specific heat consumption increases by a percentage between 1.5 and 4 percent [9]. It is shown that a temperature drop leads to an increase in the density of air and accordingly improved air mass flow rate. This further leads to rise in GT power output and efficiency of about 0.7% per ^{o}C for heavy duty GT. Ameri [10], reported that in a 16.6 MW GT when the ambient temperature decreases from 34.2^{o}C to ISO-rated condition, the average output power can be increased by as much as 11.3 per cent. He also indicated that for each 1^{o}C increase in ambient air temperature, the power output will decrease by 0.74 percent and [11] stated that for every ^{o}C rise in ambient temperature above ISO conditions, there is a loss of 0.55 per cent of the GT rated power. Alhazmy [12], in a separate work also affirmed an average power output increment of 0.57 per cent for each 1^{o}C drop in inlet temperature.

In the search for efficient and optimum performance of GT plants, a number of research works have compared different types of inlet air cooling. Some of these 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 of these techniques, the objective is to reduce inlet temperatures and the result is the subsequent increase in the plant performance [4], [6].

Performance evaluation and economic analysis of a GT plant in terms of power outage cost due to system downtime in Nigeria was studied by [3] for a period of 2001-2010 and it was illustrated that retrofitting an inlet cooler can lead to GT improve performance and measures for enhancing the performance indices were also suggested as training of operation and maintenance (O & M) personnel regularly, improvement in O & M practices, proper spear parts inventory and improvement in general housekeeping of the plant.

Among the benefits of inlet air cooling other than enhancing the performance of GT plants are reduction of poisonous/dangerous exhaust effluents such as COx, SOx, NOx and the economic benefit of it as less fuel is burnt which ultimately reflects on electricity tariffs [13]. 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 [14], [15]. Numerous researches have been carried out on the effects of inlet cooling on the performance of GT plants. However, the economic advantage of incorporating the evaporative technique is one area that deserves a deeper understanding. This paper, therefore, seeks to take an in-depth study of the thermodynamic advantages and the economic benefits of a GT 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 GT plant that is not fitted with an inlet temperature cooler and operating in the same region where all conditions applied to both plants. The type of GT plants being considered is shown as fig. 1. And it is made up of the inlet air filter (AF), evaporative cooler (CS), compressor (C), Combustor (CC) and a turbine (T).

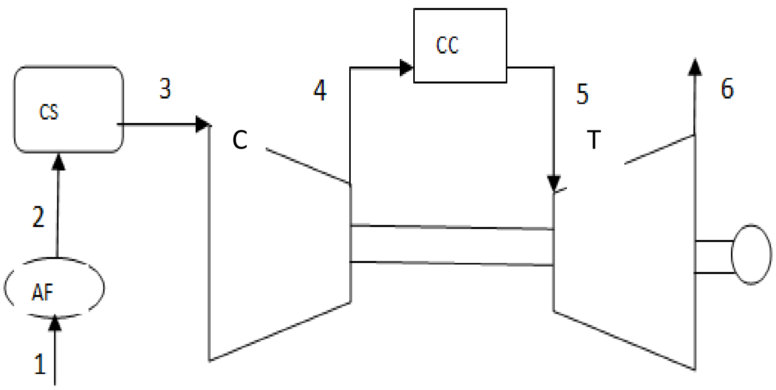

Fig. 1. Schematic of the Gas Turbine Plant with Evaporative Cooler

The evaporative cooling is the most common of all known combustion turbine inlet air cooling systems (CTIACs) owing to its numerous merits 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. The other advantages are that the installation time is also faster than all other techniques and its ability to reduce NOx emission by 0.8-1.5% per °C of cooling [16]. However, [17] shows 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 2. shows the working of the evaporative cooling techniques. A tank located at the bottom of the GT unit is used to store water. The water is pumped in through a header at the top of the media, sprays on an inverted half-pipe and as warm inlet filtered air passes through the saturated wetted media, part of the water gains latent heat, evaporates and the air loses sensible heat and its dry bulb temperature decreases with consequent increase in the air mass density. Excess water that does not evaporate is channeled downward to the tank situated below, so as not to be carried along with the cooled air. The water level is maintained by a valve that allows in water to make up for the losses. The cooled air then passes through the integral mist eliminator, where leftover water droplets are eliminated. This results to higher mass flow of air that goes into the compressor and gives the turbine higher output [6].

The water needed for the cooling is always available since there is abundance of water in the Niger Delta area where the plant is operating. More so, the quality of water is good as it is neither seawater nor brackish water with calcium hardness of 70 ppm which again is within the 50–150 ppm range of Calcium hardness ($CaCO_3$) recommended for evaporative cooling [18]. Service water pumps supply water from the flowing River which is very close to the plant into the tank at the bottom of the GT module.

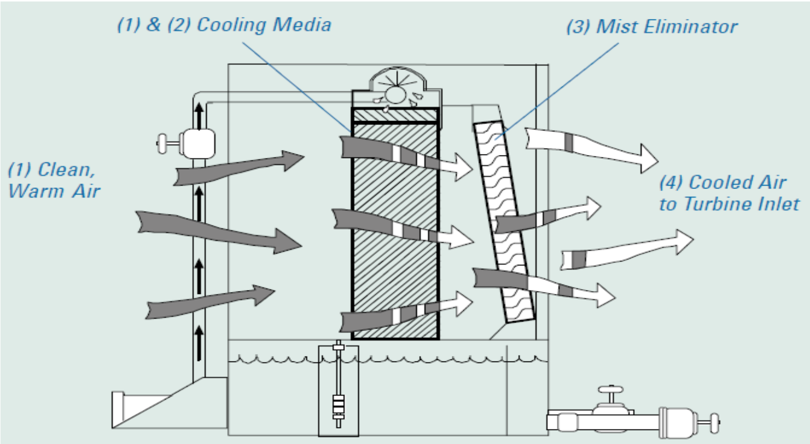

Fig. 2. Schematic drawing of the evaporative cooling process [6]

II. MATERIALS AND METHODS

For a GT plant operating in a Brayton's cycle, the output power is the difference in the turbine work and the compressor work. Therefore, the 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. This means that, if the compressor inlet temperature is made lower and the mass density of air taken into the turbine is increased, it will in turn affect the plant performance [19]. For this very importance role that turbine inlet air plays, it becomes pertinent to study the plant operating environment.

The data for this study were obtained from an operational Rolls Royce, Industrial Olympus-SK 30 GT 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 generated from the logsheet over a period of seven years. However, where certain data could not be sourced, standard thermodynamic values were used. In order to make the data workable, statistic was used to arrive at the values used for the studies. This was done by calculating the average of the daily, weekly, monthly and then yearly readings of the GT. These procedures were repeated for the years covered by this work. Ultimately, the averages of the seven years values were computed.

Modeling and simulation of each of the plant component were done. Performance of the plant without an evaporative cooling and the one that incorporates evaporative cooling system were investigated and tabulated. The result of the two systems were thereafter studied, compared and used in the computation of the values used for the economic analysis..

In order to arrive at a better understanding of this work, thermodynamic equations were derived and 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. 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.

2.1 THERMODYNAMIC MODELLING OF THE GAS TURBINE PLANT

Industrial Olympus-SK 30 GT plant is the major source of electricity in Bayelsa State, Nigeria. The plant, as in any other GT is made up of compressor, combustion chamber and the turbine. The compressor takes in air from the atmosphere, compresses it to a higher temperature and pressure which is then sent to the combustor and the products of combustion are expanded in the turbine [14].

The compression, 3-4 process in the compressor is:

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

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

The expansion process, 5-6:

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

For isentropic expansion process,

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

$$T_5 = T_6(r_p)^{\frac{k-1}{k}} \quad 2$$

The compressor work of the system is:

$$W_{c} = m_a c_p (T_4 - T_3)$$

Substituting equation 1 for T_4 above gives,

$$W_c = m_a c_{pa} T_3 \left(r_p^{\frac{k-1}{k}} - 1\right) \quad 3$$

Also, the turbine work of the plant is:

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

Substituting equation 2 for T_5 gives,

$$W_t = m_g c_{pg} T_6 \left(r_p^{\frac{k-1}{k}} - 1\right) \quad 4$$

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,

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

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:

$$W_c = \frac{m_a c_{pa} T_3}{\eta_m \eta_{cs}} \left[\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}} - 1\right] \quad 5$$

Also, the actual turbine work is therefore,

$$W_t = m_g c_{pg} T_5 \eta_t \left[1 - \frac{1}{\left(\frac{P_5}{P_6}\right)^{\frac{k-1}{k}}}\right] \quad 6$$

The net power from the GT plant is

$$W_N = W_t - W_c \quad 7$$

Heat supplied by the fuel in the combustion chamber is:

$$Q_{add} = m_a c_{pg} (T_5 - T_4) \quad 8$$

The thermal efficiency of the plant is determined as:

$$\eta_{therm} = \frac{W_N}{Q_{add}} \quad 9$$

Fig. 3 is a psychometric chart explaining the actual cooling process. The average relative humidity and the ambient temperature, Tb_2 for Niger Delta areas of Nigeria were taken to be 74.12% and 26.9°C respectively. So, while the air passes through the cooling media which is already wet with water, the air relative humidity increases and that results to decrease in the dry bulb temperature. The line B-C on the psychometric chart represents the cooling process. The wet bulb temperature, Tw_2 (indicated A) in fig. 3, is obtained as the horizontal line to the right of the Psychometric chart.

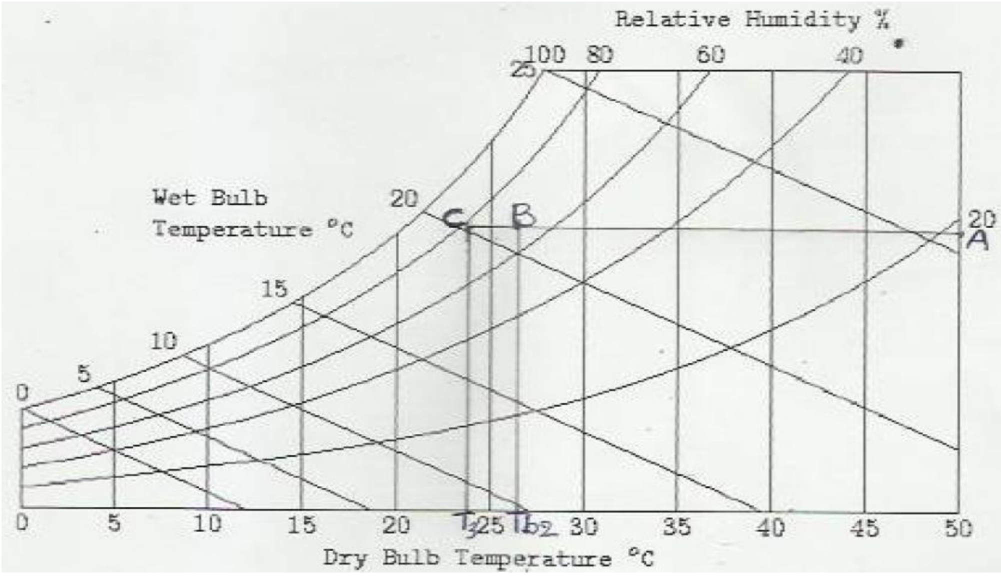

Fig. 3: Psychometric chart showing cooling from T_{b2} to T_3

The compressor inlet air temperature after the cooling is given by:

$$\boldsymbol{T_3 = Tb_2 - \varepsilon(Tb_2 - Tw_2)} \qquad 10$$

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 [6].
The rate of water evaporation is expressed by the relation:

$$E = \frac{V(w_2 - w_1)p_a}{\rho_w} m^3/_s \qquad 11$$

The blow down ratio is as shown in the ordinate of the water hardness curve in fig. 4. Therefore, the blow down rate can be obtained from the expression in equation 12.

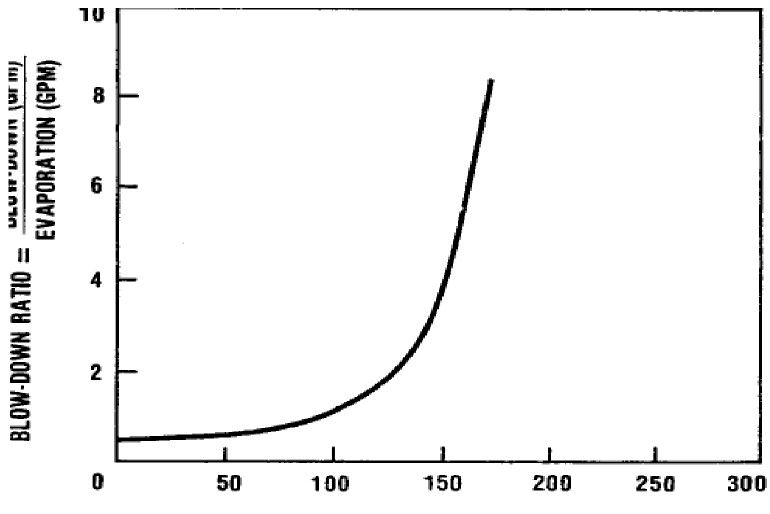

Fig.4: Hardness as $CaCO_3$ in feed water against blow-down ratio [18]

$$Blow\ down\ ratio = \frac{the\ blow\ down\ rate}{evaporation\ rate} = \frac{B}{E} \qquad 12$$

Fig. 4 is a graph depicting the variations in the feed water hardness with the blow down ratio. Having considered feed water hardness of 70 ppm. Trace from the 70 ppm vertically upward to the curve and then locate the corresponding value of blow down ratio. The blow down rate is obtained as:

$$B = 0.6E \qquad 13$$

The total water consumed in the evaporating cooling (Q_T) is equal to the sum of the water evaporation rate and the blow down rate and this is expressed as:

$$Q_T = E + B \quad 14$$

Since the water carried-over with the cooled air was neglected, Q_T is also equal to the water needed to make up for the losses. The quantity of calcium salt, 70 ppm is small and therefore the water is considered soft and good to be used for the evaporation and as a result, the cost for the water treatment is not considered.

Power gained (saved) as a result of the inlet cooling is expressed as:

$$\Delta P = P_2 - P_1 \quad 15$$

2.2 ECONOMIC ANALYSIS FOR THE EVAPORATIVE COOLER IN GT

One essence of engineering is economic. The design and choice of plants must be economical and should show an adequate return on investment [20]. To carry out the economic analysis, it is believed that the evaporative cooler reduces the ambient and dry bulb temperatures from *Tb2* to *T3*.

Therefore, according to [21], [22], the Fuel Savings Per Annum (FSPA) is given by the relation:

$$FSPA = \left(\frac{1}{\eta_1} - \frac{1}{\eta_2}\right) \times P_2 \times O_h \times \frac{Load\ factor}{LHV} \times \frac{Cost\ of\ fuel}{mass\ per\ therm} \quad 16$$

Power cost savings (Cp) for a year according to [21] is:

$$\Delta P \times O_h \times unit\ cost\ of\ energy\ generated \quad 17$$

The total savings resulting from the evaporative cooling (Cs) is:

$$FSPA + Power\ saving\ cost \quad 18$$

Operation and maintenance (O&M) cost is taken as 8% per annum of capital cost. The total cost (Ct) therefore, is expressed as:

$$Total\ cost = capital\ cost + O\&M\ cost \quad 19$$

The specific heat capacity of moist air according to [23] can be calculated from:

$$C_{pma} = C_{pa} + WC_{ps} \quad 20$$

The net profit (Np) for installing the evaporative cooling is then expressed as:

$$Total\ savings\ from\ the\ cooling - Total\ cost \quad 21$$

The cooling heat (Qc) required is given by the expression:

$$Q_c = m_a \times C_{pma}\,[T3 - Tb2] \quad 22$$

The capital cost (Cc) for incorporating the cooling system [21] is:

$$Capital\ cost = Q_c \times unit\ capital\ cost \quad 23$$

III. RESULTS AND DISCUSSION

The parameters used for this study are the ambient temperature, turbine compressor inlet temperature from the evaporative cooler, fuel savings per annum, the total savings resulting from employing evaporative cooling, net profit for installing an inlet cooler, power gained 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 1. Summary of overall average of the working parameters (from 2002 to 2008) [6].

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* *Maximum Temperature, T_5* *Inlet Pressure, P_4* *Outlet Pressure, P_5* *Mass flow rate of fuel, m_g*	*K* *K* *Bar* *Bar* *Kg/s*	*509.08* *1055.40* *6.43* *6.30* *3.05*
Turbine	*Inlet Temperature, T_5* *Outlet Temperature, T_6* *Inlet Pressure, P_5* *Outlet Pressure, P_6* *Mass flow rate, m_g* *Isentropic Efficieny of Turbine, η_{ts}*	*K* *K* *Bar* *Bar* *Kg/s* *%*	*1055.40* *668* *6.30* *1.013* *85.19* *0.87*
Exhaust	*Exhaust gases temperature, T_6* *Exhaust gases pressure, P_6* *Mass flow rate, m_g*	*K* *Bar* *Kg/s*	*688* *1.013* *85.19*
Others	*Load factor* *Cost of fuel* *Lower Heating Value, LHV*	*%* *$/kg* *MJ/kg*	*62.14* *1.934/22.10* *47.14*

Table 2 shows the performance of the plant without the evaporative cooler. The parameters in this table are obtained with the application of the equations derived in the thermodynamic modeling and the values in table 1.

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

YEAR	*Tb2(K)*	*T4(K)*	*Wc(KJ/kg)*	*Wt(KJ/kg)*	*AFR*	*SFC(kg/KWhr)*	*Qadd(KJ/kg*	*P_1(MW)*	*η_1(%)*
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*

The parameters in table 3 were generated after the compressor inlet air has been made to pass through the evaporative cooling process. The important difference in the tables 2 and 3 as can be seen from a close observation is that Tb2, P_1 and η_1 in table 2 are the ambient air temperature that goes into the turbine and the resulting power and thermal efficiency respectively. Whereas in table 3, although Tb2 represents the ambient air temperature, T3 is the actual air inlet temperature that goes into the compressor after the air from the evaporative cooler has been cooled and therefore, the consequent power and thermal efficiency are P_2 and η_2.

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

YEAR	*Tb2*	*T3*	*T4*	*Wc*	*Wt*	*AFR*	*SFC*	*Qadd*	*P_2*	*η_2*
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*

3.1 TURBINE PARAMETERS IMPROVEMENT

Increase efficiency with decrease compressor inlet temperature is clearly demonstrated in fig. 6. The least ambient temperature of 300.05 K recorded is in the year 2002 and this gave the greatest efficiency of 21.76%. In comparing fig. 6 with fig. 7. it can be seen that the turbine efficiency can be made to improve by reducing the compressor inlet temperature.

This was proved when the inlet temperature was brought down to 297.62 K with a resultant increase in efficiency to 21.90%.

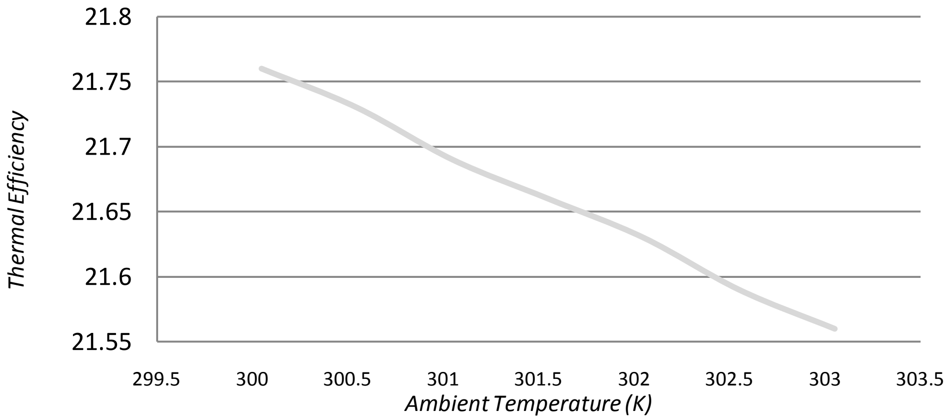

Fig. 6: Thermal Efficiency versus Ambient Temperature, (K) Before Cooling

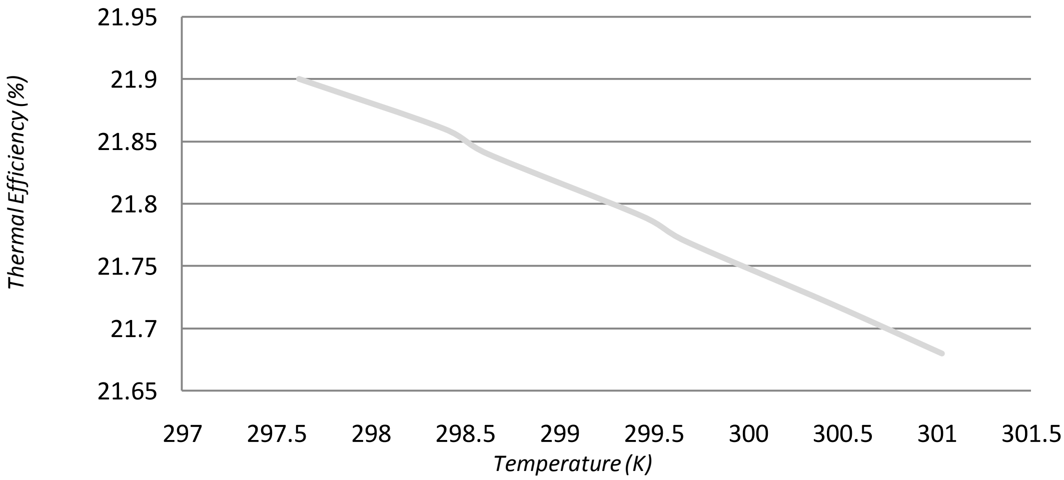

Fig. 7: Thermal Efficiency versus Ambient Temperature, (K) After Cooling

The effects of the variations of ambient temperatures and the power outputs of a GT plant are illustrated in figures 8 and 9. Both figure 8 and figure 9 show the fact that the lower the environmental temperature, the higher the power output. A thorough observation of fig. 9 shows that temperature reduction of 2^{o}C can result to an increase in power output of 2.02MW.

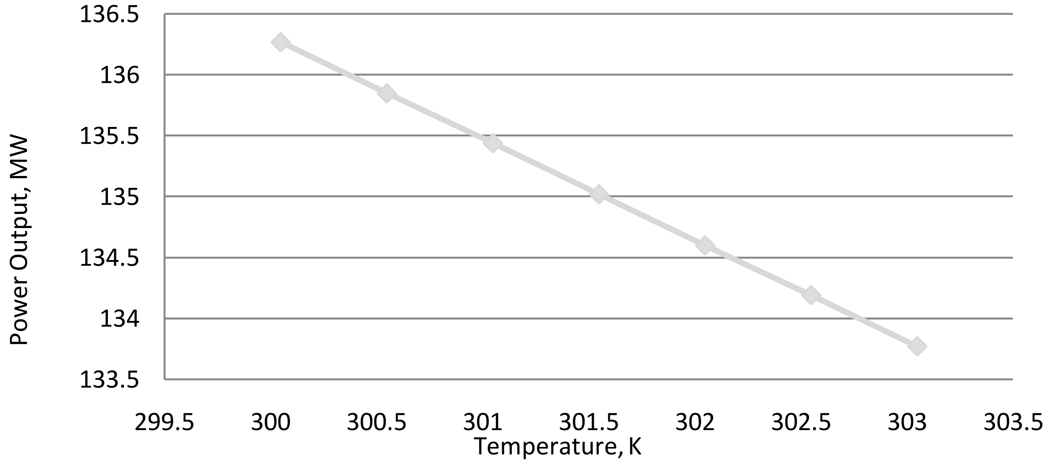

Fig. 8: Power Output versus Ambient Temperature, (K) After Cooling

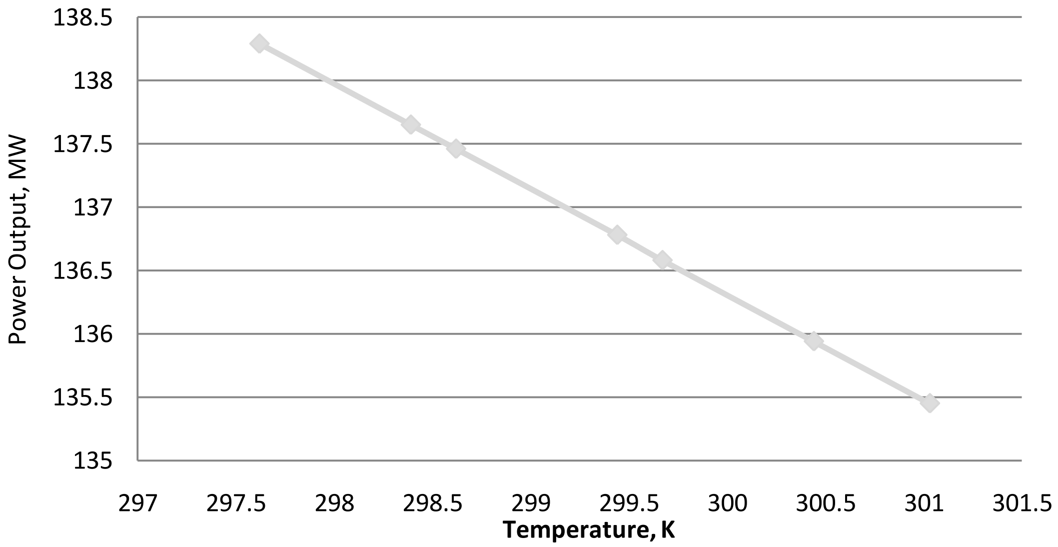

Fig. 9: Power Output versus Ambient Temperature, (K) After Cooling

3.2 ECONOMIC ANALYSIS

The results of the values obtained for the purpose of analysis of the economic benefits of employing the inlet cooling are shown in table 4 which was generated from the values in tables 1, 2 and 3 with the application of relevant equations derived earlier.

Table 4:*Summary of economic analysis*

YEAR	*FSPA*	*Cp*	*Cs*	*Qc*	*Cc*	*O & M*	*Ct*	*Np*	*ΔP*	Q_h
2002	*204.56*	*15061.00*	*15265.74*	*346.01*	*79582.30*	*6366.58*	*85948.88*	*-70757.90*	*2.02*	*5849*
2003	*189.66*	*17937.45*	*18127.11*	*307.56*	*70739.35*	*5659.15*	*76398.50*	*-58271.39*	*1.80*	*6966*
2004	*219.10*	*20917.65*	*21136.75*	*346.00*	*79581.77*	*6366.54*	*85948.33*	*-64811.60*	*2.02*	*7253*
2005	*189.68*	*19786.30*	*19975.98*	*300.44*	*69101.87*	*5528.15*	*74630.00*	*-54654.04*	*1.76*	*7684*
2006	*204.42*	*17787.28*	*17991.70*	*338.90*	*77944.40*	*6235.54*	*84179.94*	*-66188.24*	*1.98*	*6396*
2007	*189.70*	*10790.50*	*10980.20*	*300.44*	*69101.87*	*5528.15*	*74630.00*	*-63649.80*	*1.75*	*7483*
2008	*175.02*	*17985.45*	*18160.50*	*287.63*	*66154.40*	*5292.35*	*71446.75*	*-53286.25*	*2.17*	*7592*

3.2.1 Fuel Saving per Annum (FSPA)

Table 4 is a summary of the values of the economic analysis of the GT plant in the seven years covered in this research. The FSPA as illustrated in equation 16 depended on the efficiency difference and to a great extent on the power output of the plant. It therefore implies that the wider the efficiency difference and greater the power output, the higher the FSPA. This again can be seen in the year 2002 where the two efficiencies are 21.76% and 21.90% with power gained as 2.02MW The FSPA values as seen in table are small, this is due to the low power generated by the GT mainly owing to the fact that the plant was meant to serve relatively smaller population. Significant FSPA can be obtained in higher power GT plants.

3.2.2 Total Savings for Cooling (Cs)

This is the sum of FSPA and the power cost saving. The power cost saving is generated from the products of the power gained (ΔP), operational hours (Q_h) and the unit cost of energy generated (US$10.30). The total savings for cooling is a function of the operational hours. This is due to the unit cost of energy generated, the longer the operational hours the more fund that will be generated. However, it also depends on the increase in power output caused by employing the cooling medium.

3.2.3 Capital Cost (Cc)

The cooling heat (Qc) required by the evaporative cooling is an important requirements in the estimation of the capital cost of installing an inlet cooler. As can be seen in table 4, the higher the cooling heat the larger the capital cost as the greatest cooling heat of 346.01kW and 346.00kW gave the highest capital cost of US$79582.30 and US$79581.77 respectively. However, the decision made here in calculating the capital cost is pegged by the amount US$230/kW as a unit cost for incorporating the air cooling system [22][23]. Therefore, the total estimation involved in stalling the evaporative cooler is the sum of the capital cost and the operation and maintenance cost.

3.2.4 Net Profit for Installing Inlet Cooling (Np)

This is the net profit that will be generated for incorporating the air cooling system. It is the difference between the total savings resulting from the evaporative cooling and total cost of employing the cooling system. As it can be seen in table 4, the first year of investing as is common to most public owned business is not expected to yield profit possibly because smaller unit cost of the energy generated and less power generated. However, the benefit in terms of profit can be significant when the power output is made to increase, the operational hours increased and the unit cost of energy is also increased.

IV. CONCLUSION

The effects of inlet air cooling on the performance of a GT power plant were studied. The study showed that in installing an air inlet cooler, power output and thermal efficiency of the GT plant can be improve such that a decrease in temperature by 2.43^{o}K can result to an increase power of 2.02MW with a consequent rise in efficiency from 21.76-21.90%. This power increment derived from the application of the inlet cooling determines to a great extent the economic analysis of the plant and the benefits derived in it. The study also showed that apart from the GT performance there are economic benefits such as the fuel savings, total savings and net profit obtained from the application of an inlet cooling technology.

V. ACKNOWLEGDEMENTS

I wish to acknowledge the efforts of Mike, Odiowei. They all, in one way or the other, assisted in providing materials and helped in the formatting and the eventual publishing of this work.

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