

Performance Analysis and Components Irreversibilities of a (25 MW) Gas Turbine Power Plant Modeled with a Spray Cooler

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Abstract: In this study the effect of inlet air cooling system and components irreversibilities on the performance of an active 25MW gas turbine power plant was investigated. The objective of this study was to establish the potential benefits of improving the performance of the current gas turbine plant into a more advanced cycle with high efficiency and power output through inlet air cooling. **Problem statement:** The hypothesis was that the low performance of the gas turbine plant was caused by high ambient temperature, the use of spray cooler was adopted to bring the air condition temperature close to ISO condition. **Approach:** In this study, performance characteristics were determined for a set of actual operational conditions including ambient temperature, relative humidity, turbine inlet temperature and pressure ratio. **Results:** The results obtained show that the use of a spray cooler on the existing gas turbine cycle gives a better thermal efficiency and less irreversibility rate in the components system and the entire plant. The power output of the gas turbine plant with spray cooler was found to have increased by over 7%, accompanied by 2.7% increase in machine efficiency with a reduction in specific fuel consumption of 2.05 and 10.03% increase in the energy of exhaust. Furthermore, a 0.32% reduction in the total irreversibility rate of the plant for the cooled cycle was obtained and a 0.39, 0.29 and 0.17% reduction in the irreversibility rate of compressor, turbine and combustion chamber respectively, were also obtained. **Conclusion:** The results show that retrofitting the existing gas turbine plant with inlet air cooling system gives a better system performance and may prove to be an attractive investment opportunity.

Key words: Gas Turbine (GT), Dry Bulb Temperature (DBT), Engineering Equation Software (EES), Specific Fuel Consumption (SFC), Relative Humidity (RH)

INTRODUCTION

Gas turbines are widely used for electricity generation in most countries of the world. They can be started and stopped easily allowing them to be brought into service as required to meet energy demand at peak conditions (Jaber *et al.*, 2007). Because of natural gas availability and at low prices compared to distillate fuels, many countries of the world, example Nigeria utilizes large conventional gas turbines as based load units. The average efficiencies of gas turbine plants in the Nigerian energy utility sector over the past decade was in the range 27-30% (Abam *et al.*, 2011). The low efficiencies of the gas turbine plants are tied to many factors which include: operation mode, poor maintenance procedures, age of plant, discrepancies in operating data, high ambient temperature and relative humidity. Power output and efficiency of a gas turbine plant depends largely on the condition of the compressor

inlet air temperature (Cortes and William, 2003). The performance output during hot conditions is less compared to the performance at high air temperature and humid environment, so cooling the inlet air temperature to gas turbine, increases the air density, which enhances the mass flow rate of air and gives better power output (Jaber *et al.*, 2007). Research has shown that gas turbine power output decreases by 18% for a 10°C increase in compressor inlet air temperature (Cortes and William, 2003). It has also been reported that high ambient temperature increases the turbine's heat rate resulting in gas turbine plants producing 25-35% less power in summer than winter at an average increase of 6% in fuel consumption (McCracker, 1991).

Many researchers have presented different methods of power augmentation (Landry and Sarian, 1994; Johnson, 1998) presented a discussion of the theory and operation of evaporative coolers for industrial gas

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turbine system applications. Ondryas *et al.* (2007) demonstrated the impact of using chillers at the air intake system to enhance gas turbine power output in a cogeneration plants they obtained over 20% increase in net power output. The study of (Elliot, 2001) shows that a 1% gain of the output power was obtained for every 1.6°C drop in compressor inlet air temperature using water chillers. Kolp *et al.* (2001) demonstrated the effect of various forms of inlet air cooling and supper charging of 40MW GE LM6000 gas turbine plant. Their results indicate that a 28°C reduction in the compressor inlet air temperature increased the net power output by 30%. In addition, (Mercer, 2002) reported in his study that chillers utilizing thermal storage systems would increase the gas turbine power output by over 25% during peak periods. Similarly evaporative coolers gives an increase of about 10-15% power output, while water fogging a cooling technique will enhance power output by 10-20% (Amell and Cadavid, 2002; Mohanty and Paloso, 1995).

Furthermore, With the rapid increase in electricity demand in Nigeria and the expected shortages in power supply due to delays in implementation of the major power projects, retrofitting the existing Gas Turbine (GT) power plants in the power sector with inlet-air pre-cooling system is an attractive investment opportunity for the Power Holding Company of Nigeria (PHCN). This study studies the performance enhancement and modification of an active 25MW gas turbine plant (a grid-connected GT) operating at Delta Power Station in Nigeria using the prevailing climatic conditions. The performance of the proposed modified Bryton cycle is compared with base case condition (without inlet air cooling). The irreversibility rate in the components system of the gas turbine plant for the two scenario will be considered.

Gas turbine cycle parameters: Data used for the analysis are obtained from maintenance data sheet of H-25MW Hitachi single shaft turbine plant. The nominal performance data at ISO condition (15°C and 60% RH), power output 26.9MW, fuel Low Heating Value (LHV) 4699 kJ kg⁻¹, rated revolution speed 7,280rpm, gas intake flow rate 88 kg sec⁻¹, exhaust gas temperature 555°C, Pressure Ratio (PR) 14.7, compressor and turbine, 17 and 3 stages axial flow types.

Model description:The schematic diagram of the spray cooler and gas turbine plant are shown in Fig. 1 and 2. Cold water is sprayed into the dry atmospheric air entering the system with the help of injectors. The water spraying system is counter flow in evaporative cooler and modeled as an adiabatic saturator. The dry air mixes with water becoming saturated where the Dry Bulb Temperature (DBT) of inlet air reaches initial Wet Bulb Temperature (WBT).

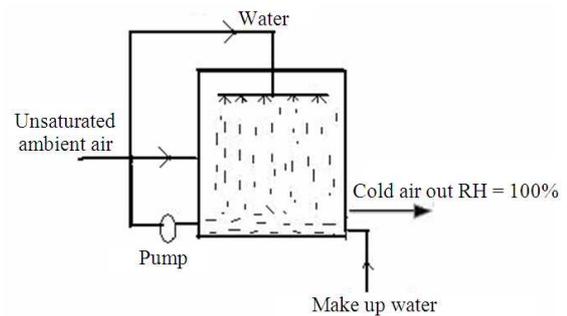


Fig. 1: Schematic diagram of spray cooler

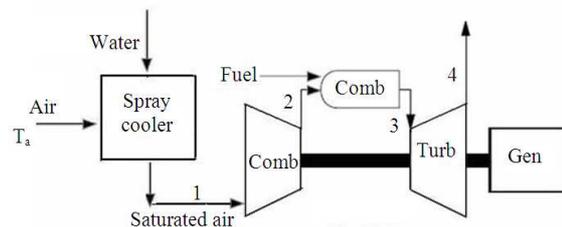


Fig. 2: Schematic diagram of gas turbine with spray cooler

It is assumed that the relative humidity of outlet air from the cooler will not exceed 100% despite the inlet air condition. The humidifying efficiency is assumed to be 100%. The spray cooler lowers the ambient air temperature by 13°C for hot and dry ambient conditions, 50°C and 40% relative humidity and by 7.2°C for cold and humid conditions, 25°C and 50% relative humidity (Kumara *et al.*, 2007). The gas turbine power plant models are developed on the following assumptions: (i) all components have adiabatic boundaries, (ii) plant performance at ISO conditions, (iii) the air and the combustion products are assume ideal characteristics, (iv) kinetic and potential components of energy are neglected and (v) the ambient conditions of temperature and pressure are 25°C and 1.013 bar. Applying energy balance for the cooler in Fig. 1.

The heat interaction between the ambient air and the saturated air is presented in Eq. 1 (Kumara *et al.*, 2007):

$$C_{p_a}(T_a - T_1) = (\omega_1 - \omega_a) h_{fg} \quad (1)$$

$$\omega = \frac{0.622P_v}{P - P_v} \quad (2)$$

where, ω_a and ω_1 are humidity ratios before and after the saturation respectively. In general ω is related to the water vapour pressure at saturation as presented in Eq. 2 (Cengel and Boles, 2010), P is pressure of air at

intake and P_v is saturated pressure of water at intake. The cooler outlet temperature is obtained in Eq. 3 (Shanbghazani *et al.*, 2008):

$$T_1 = T_a - (T_a - T_{wb}) \times \epsilon \quad (3)$$

Where:

T_a = Ambient temperature
 T_1 = Cooler outlet temperature
 T_{wb} = Wet bulb temperature
 ϵ = Cooler efficiency

The working fluid passing through the compressor is assumed to be an ideal mixture of air and water vapour. The total enthalpy of atmospheric air is given below as in Eq. 4, (Cengel and Boles, 2010):

$$h = h_a + \omega \times h_v \cong C_p T + \omega h_g \quad (4)$$

Where:

h_a = Enthalpy of dry air
 h_v = The enthalpy of water vapour

The enthalpy of water vapour can be evaluated approximately as in Eq. 5, Cengel and Boles (2010):

$$h_v = 2500.9 + 1.82 \times T \quad (5)$$

The total temperature of the fluid leaving the compressor having an isentropic efficiency η_c can be evaluated using ideal gas relation obtained in Eq. 6 (Shanbghazani *et al.*, 2008):

$$T_{02} = T_1 + \frac{T_2 - T_1}{\eta_c} = T_1 + \frac{T_1}{\eta_c} \left[r_p^{\left(\frac{k-1}{k}\right)} - 1 \right] \quad (6)$$

Where:

r_p = Compressor ratio
 k = Specific heat ratio

Similarly the total temperature leaving the turbine having an isentropic efficiency η_t is given as Eq. 7:

$$T_{04} = T_3 - \eta_t (T_3 - T_4) \quad (7)$$

The total mass flow rate of humid air is given as Eq. 8:

$$\dot{m}_{ha} = \dot{m}_{mda} + \omega \dot{m}_{da} = (1 + \omega) \dot{m}_{da} \quad (8)$$

where, \dot{m}_{da} and \dot{m}_{ha} are mass flow rates of humid air and dry air. The compressor work is calculated from the mass flow rate and enthalpy change across the compressor Eq. 9:

$$\dot{W}_c = \dot{m}_{air} (1 + \omega) \times (C_{pa} T_{02} - C_{pa} T_1) + \omega (h_{02} - h_1) \quad (9)$$

Similarly, the turbine work is obtained as Eq. 10:

$$\dot{W}_T = (\dot{m}_{air} + (1 + \omega) \dot{m}_f) \times (C_{pg} T_3 - C_{pg} T_{04}) + \omega (h_3 - h_{04}) \quad (10)$$

The energy of exhaust gas E_G is obtained as Eq. 11:

$$\dot{E}_G = (\dot{m}_{air} + (1 + \omega) \dot{m}_f) \times (C_{pg} T_{04} - C_{pg} T_1) + \omega (h_{04} - h_1) \quad (11)$$

The net power obtained from the gas turbine is Eq. 12:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_c \quad (12)$$

The thermal efficiency of the gas turbine power plant is evaluated as Eq. 13:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{m}_f \times LHV} \quad (13)$$

The specific fuel consumption is expressed in Eq. 14, (Shanbghazani *et al.*, 2008). The Specific fuel consumption compares the ratio of the fuel used by an engine to a certain force such as the amount of power the engine produces. This is a very important economic criteria.

$$Sfc = \frac{3600 \times \dot{m}_f}{\dot{W}_{net}} \text{ Kg / KWh} \quad (14)$$

Components irreversibilities: Equation 15 presents the steady state exergy balance equation for a control volume according to (Kotas, 1995):

$$\dot{E}x = \sum_j \left(1 - \frac{T_0}{T_j} \right) \dot{Q}_j + \dot{W}_{cv} + \sum_i \dot{m}_i e_i - \sum_i \dot{m}_e e_e \quad (15)$$

The subscripts i.e., j and o refer to conditions at inlet and exits of control volume boundaries and reference state. The components irreversibility rates is derived from Eq. 15 written as Eq. 16:

Compressor:

$$\dot{I}_c = \dot{E}_{x1} - \dot{E}_{x2} + \dot{W}_c \quad (16)$$

\dot{W}_c is the total power input rate to the compressor; \dot{I}_c is the total irreversibility rate in the compressor; \dot{E}_{x1} is the exergy rate of the air and \dot{E}_{x2} exergy outflow rate from the compressor unit.

Combustion chamber: The exergy flow rate in the combustion chamber is obtained from Eq. 15. In the combustion chamber, the shaft work is zero and Eq. 15 reduces to to Eq. 17:

$$\sum_i (\dot{E}_{xf})_i + \sum_{in} \dot{E}_x c_2 = \sum_{in} \dot{E}_x c c_3 + \dot{I}_{CC} \quad (17)$$

Where:

\dot{I}_{CC} = The combustion chamber irreversibility.

\dot{E}_{xf} = Fuel exergy rate which is expressed in Eq. 18:

$$\begin{aligned} \dot{E}_{xf} = & \dot{E}_{xph} + \dot{E}_{xchem} = \dot{m} c_p^h \\ & \left[(T_3 - T_0) - T_0 c_p^s \ln \left(\frac{T_3}{T_0} \right) \right] + RT_0 \ln \\ & \left(\frac{P_2}{P_0} \right) + \sum_i x_i e_{io} + RT_0 \sum_i \ln \gamma_i x_i \end{aligned} \quad (18)$$

\dot{E}_{xchem} is chemical exergy rate, x_i is mole fraction of constituent's elements and e_{io} is standard molar exergy of each constituent of the natural gas species.

P_0 and P_2 are the compressor inlet and exit pressures, respectively; c_p^h and c_p^s are mean molar isobaric heat capacity for evaluating enthalpy and entropy changes. They are expressed in Eq. 19 and 20:

$$c_p^h = \frac{1}{T - T_0} \int_{T_0}^T c_p^- dT \quad (19)$$

$$c_p^s = \frac{1}{\ln \left(\frac{T}{T_0} \right)} \int_{T_0}^T c_p^- dT \quad (20)$$

c_p^- is expressed or obtained in a polynomial form as a function of temperature as in Eq. 21 (Himmelblau and Riggs, 2012):

$$c_p^-(T) = a + b^2T + cT^2 + dT^3 \quad (21)$$

The values of a, b, c and d are constant characteristics of the gas considered and obtained from standard tables.

Air and constituents of natural gas specific heats for $200K < T < 800K$ and $273K < T < 1500K$ respectively are given as in (Himmelblau and Riggs, 2012):

$$\begin{aligned} C_{pair} = & 28.09 + 0.1961 \times 10^{-2} T \\ & + 0.4799 \times 10^{-5} T^2 - 1.965 \times 10^{-9} T^3 \end{aligned}$$

For methane, ethane, propane e.t.c their specific heats are:

$$\begin{aligned} C_p CH_4 = & 19.87 + 5.021 \times 10^{-2} T \\ & + 0.366 \times 10^{-5} T^2 - 11 \times 10^{-9} T^3 \\ C_p C_2H_6 = & 49.37 + 13.92 \times 10^{-2} T \\ & + 5.816 \times 10^{-5} T^2 - 7.28 \times 10^{-9} T^3 \\ C_p C_3H_8 = & 68.032 + 22.59 \times 10^{-2} T \\ & + 71 \times 10^{-5} T^2 - 31 \times 10^{-9} T^3 \end{aligned}$$

Turbine: The irreversibility rate in the turbine is obtained as in Eq. 22:

$$i_T = \dot{m} \left[T_0 \left(c_p^s \ln \frac{T_3}{T_4} - R \ln \frac{P_3}{P_4} \right) + c_p^h (T_3 - T_4) (1 - \eta_T) \right] \quad (22)$$

MATERIALS AND METHODS

Operating data for Gas turbine unit were collected from the daily turbine control log sheet for a period of two years. The daily average operating variables were calculated using MS Excel worksheets and the thermodynamic properties were determined using an Engineering Equation Software (EES). The analysis of the plant was divided into different control volumes and performance of the plant was estimated using component-wise modeling. Mass and energy conservation laws were applied to each component and the performance of the plant was determine for the simple system (without spray cooler) and for the cooled system (with spray cooler). The irreversibility rate and the efficiencies of the turbine components were evaluated employing Kotas exergy models. Results for the two scenario were compared.

RESULTS

The results of the effect of spray cooler on the performance of the gas turbine plant are presented. Figure 3 and 4 shows the variation in thermal efficiency with ambient temperature and power output for cooled and simple cycles at 60 and 100% Relative Humidity (RH).

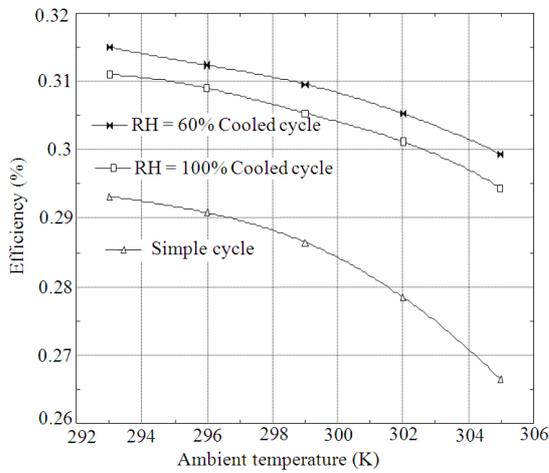


Fig. 3: Efficiency Vs ambient temperature for simple and cooled cycles

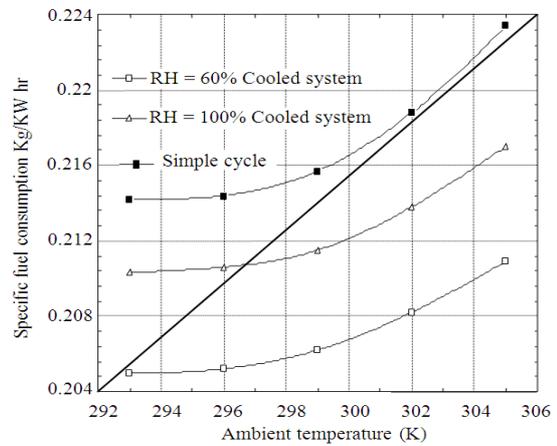


Fig. 5: Specific fuel consumption versus ambient temperature for simple and cooled cycles

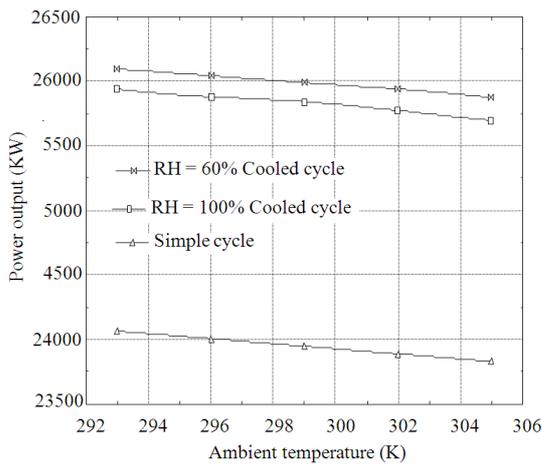


Fig. 4: Power output Vs ambient temperature for simple and cooled cycles

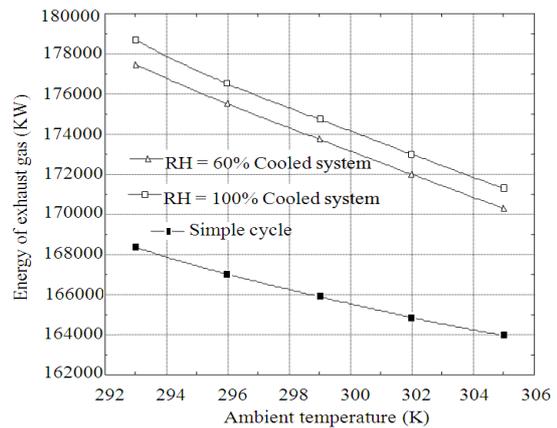


Fig. 6: Energy of exhaust versus ambient temperature for simple and cooled cycles

Table 1: Irreversibility rates and component efficiencies for simple and cooled cycles			
Components	Compressor	Combustion chamber	Turbine
Simple cycle			
Irreversibility rate (KW)	11841.34	66368.76	26352.57
Component efficiency (%)	70.20	30.67	60.35
Cooled cycle			
Irreversibility rate (KW)	11794.27	66256.64	26274.42
Component efficiency (%)	72.37	33.54	62.44

The variation in specific fuel consumption and the energy of exhaust with temperature for 60 and 100% RH are shown in Fig. 5 and 6. While the effect of 20% increase in RH to power output is shown in Figure 7. The components irreversibilities and efficiencies of the gas turbine plant for the two cycles are presented in Table 1.

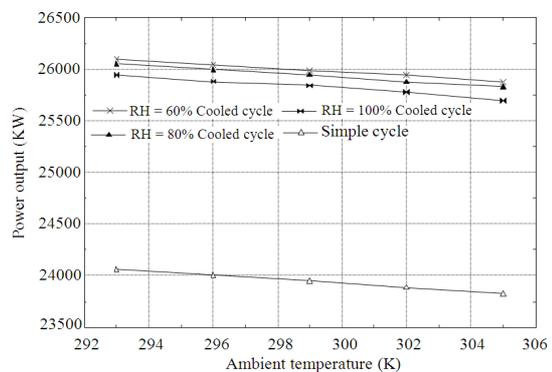


Fig. 7: Power output versus ambient temperature for 20% increase in relative humidity

DISCUSSION

The power output and the efficiency of the gas turbine cycle were calculated for different ambient air temperatures and for relative humidity of 60 and 100%. Thermodynamic properties such as specific heats, humidity ratio of air, enthalpy of steam and combustion gasses were evaluated as a function of temperature, pressure, relative humidity by using an Engineering Equation Software (EES). Figure 3 and 4 shows the variation in thermal efficiency with ambient temperature and power output for cooled and simple cycles at 60 and 100% relative humidity. From the results a degree increase in ambient temperature lead to 0.79% drop in gas turbine rated power. For low ambient temperature of 23°C (296K) and Relative Humidity (RH) of 60% decreased to ISO standard design condition, the turbine output power increased by about 6% with a 2.1% increase in thermal efficiency. At high ambient temperature of 32°C (305K) and 60% relative humidity decreased to ISO standard condition, the power output increased by 7.95% with a 2.7% increase in thermal efficiency. The Specific Fuel Consumption (SFC) of gas turbine decreases as the ambient temperature increases as shown in Fig. 5. At low ambient temperature of 23°C and 60% relative humidity decreased to ISO standard design condition, the specific fuel consumption drops by 0.92% and at high ambient temperature of 32°C (305K) and 60% RH decreased to ISO standard condition the specific fuel consumption drops by 2.0%. Figure 6 shows that at an ambient temperature of 23°C (296K) decrease to ISO standard condition, the energy of exhaust gases increases by 8.19% and at high ambient temperature of 33°C (306K) decrease to ISO standard condition, the energy of exhaust gases increases by 10.03%. This means that lowering the inlet air temperature will lead to an increase in the exhaust gas flow rate with more heat recovered from the exhaust gases (Karim, 2008). Further increase in Relative Humidity (RH) by 20% as shown in Fig. 7 results to a decrease in power output of about 0.32% accompanied by a corresponding drop of 0.32% in thermal efficiency and about 0.33% increase in specific fuel consumption. This trend is in agreement with the results obtained by (Karim, 2008). Better performance of the gas turbine plant can be obtained as shown from the results when the operating conditions are brought close to ISO conditions.

Table 1 presents the irreversibility rate and efficiencies of components system of the gas turbine

plant. The total irreversibility rate for the simple cycle is 155275.91KW while for the cooled cycle is 154786.45KW. The result shows that a drop in ambient temperature caused a 0.32% reduction in the total irreversibility rate of the plant. Similarly, a drop in ambient temperature leads to 0.39, 0.29 and 0.17% decrease in irreversibility rate of compressor, turbine and combustion chamber respectively. The combustion chamber has the lowest efficiency due to large irreversibilities resulting from the differences between flame and combustion fuel. The performance parameters for the cooled cycle at all values of ambient temperature improves the gas turbine system. The results also show a reduction in the rate of irreversibility in gas turbine components for cooled cycle in Table 1, where components efficiencies were improved culminating to a low thermal discharge index of 3.24. Thermal Discharge Index (TDI) is the total number of thermal energy units discharged to the environment for every unit of electrical energy produced by the plant. It is completely dependent on thermal efficiency (Culp, 1979). Low value of TDI conotes better system performance.

CONCLUSION

The performance of spray cooler is dependent on ambient temperature and humidity. The system works efficiently during dry and hot climatic conditions prevalent in some locations in Nigeria. The results obtained show that, the use of an inlet air cooling system as a measure for system enhancement was found to increase the power output of the gas turbine by over 7%, accompanied by 2.7% increase in machine efficiency. A drop in specific fuel consumption of 2.0 and 10.03% increase in energy of exhaust were observed. The irreversibility rate is less for the cooled cycle with improvement in thermal efficiency which is due to less entropy generation. However, comparing the cooled cycle and the simple cycle it was observed that for any ambient temperature and relative humidity, the cooled cycle improves all the performance indices of the gas turbine plant. The payback period for the spray cooler is between 4-5 years. Water needed for operation of sprayer may be a serious constraint in some areas, but condensing water from the exhaust system is one method for recovering partially the make up water for spray cooler especially in extreme dry climates. , in order to select a particular cooling technique for adoption in Nigerian energy utility sector, research including system performance and economic analysis of the cooling methods will appear to be merited.

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