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Thermo – Economic Analysis of Compressor Inlet Air Precooling Techniques of a Gas Turbine Power Plant Operational in Nigeria Energy Utility Sector

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Abstract— **The thermo-economic analysis of precooling techniques of a gas turbine power plant is based on the economic investigation of various precooled inflow air methods used to improve the performance of gas turbine operational in Nigeria energy utility sector using thermodynamic principles. The various inlet air pre-cooling methods utilized in this study are evaporative, vapour compression and vapour absorption. This study compared the economics and thermodynamic analysis of the various gas turbine inlet air precooling techniques taking into consideration the prevailing heterogeneous climatic conditions in Nigeria (Northern and Southern regions of Nigeria). The Total Annual Cost and system profitability for evaporative, vapour compression and Vapour absorption are N4590236.1/KWh and N159149400/KWh; N12576095/KWh and N104124200/KWh and N213218642/KWh and N1418927000/KWh respectively. Owing to the heterogeneous climatic conditions in Nigeria, this study recommends that evaporative precooling technique should be utilized in regions where there is availability of water while in region of water scarcity the other pre-cooling methods can be utilized.**

Keyword: Evaporative precooling, vapour compression, vapour absorption, system profitability, net revenue value.

I. INTRODUCTION

Nigeria as a case study does not have a homogeneous climate and weather condition. However, the pre-cooling method utilized in the southern region of Nigeria may not be favourable in the Northern region where the ambient temperature is usually high. This study compares the different pre-cooling methods (evaporative, vapour compression and Vapour absorption) used and recommends which method will be appropriate for a particular region with regard to the prevailing climatic conditions.

The inflow air temperature has a significant effect on the performance of a gas turbine plant [1]. High inlet air temperature lowers the performance of a gas turbine. To overcome the negative effect of high inlet air temperature during hot climate, different pre-cooling methods have been are utilized to reduce the air temperature before passing through the gas turbine The cost of operating and maintaining these pre-cooling methods add additional problems to the gas turbine operation in Nigeria energy utility sector. For a given gas turbine plant, the cost of the pre-cooling method is a function of operating variables [1, 2]. During hot climates, for a gas turbine to generate greater power output, it will require greater pre-cooling loads and this will result to increased cost of the pre-cooling system. The evaporative pre-cooling method has the advantage of reduced investment charge, simplicity of designs, low power consumption, low operation cost and is friendly to our environment as it uses water for its operation [3, 4]. However, its limitations are high water consumption, limited power gain and capacity improvement.

High capital, maintenance and operational cost, as well as high heat rejection are some of the problems of absorption pre-cooling method while higher parasitic load, high maintenance cost and ice formation at compressor inlet as well as causing global warming are all drawbacks associated with vapour compression pre-cooling technique [5].

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This study analyze the economics of the various pre-cooling methods used to enhance the Gas Turbine Power output operational in Nigeria Energy utility sector.

II. MATERIALS AND METHODS

For this research, a HITACHI – MS – 7001B Gas turbine plant located in Nigeria Agip Oil Company plant yard in Obrikom, Omoku, Rivers State, Nigeria was considered. The plant consists of three units - the compressor, the turbine and the pre-cooling units.

The performance of the plant without pre-cooling method was determined as well as its performance with the various pre-cooled system Table I shows key design stipulations in consideration.

TABLE I
Range of Parameters for the Present Analysis.

Parameters	Range
Ambient air	
Ambient air temperature	27 – 50^0 C
Ambient air relative humidity	18 – 84%
Gas Turbine Model Hitachi – MS – 7001b	
Pressure ratio (P_2/P_1)	10
Net power (I.S.O)	52.4MW
Site Power	37MW
Turbine Inlet temperature	1274K
Air mass flow rate	141.16kg/s
Fuel calorific value	46000kJ/kg
Turbine isentropic efficiency	82%

A. *Description of the Plant*

The diagrammatic representation of a Hitachi –MS – 7001B is shown in Figure 1. As the ambient air entered the compressor, and compressed by the compressor, its temperature increases. Upon entering the combustion chamber fuel is sprayed into the compressed air stream resulting in the fuel combustion. The expansion of gas resulting from the combustion of the fuel generate the mechanical work that turns the shaft of the turbine.

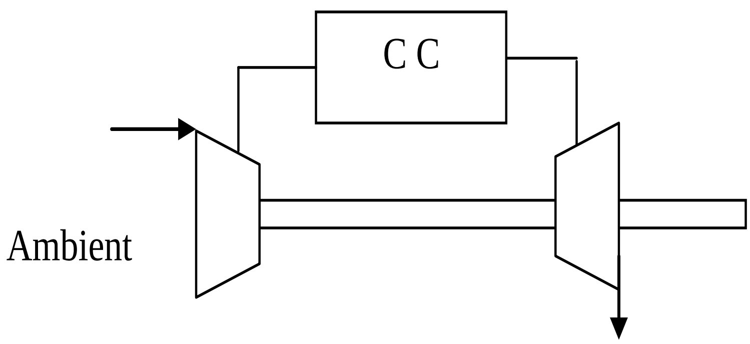

Fig. 1. Diagram of the typical gas turbine.

The gas turbine unit

Based on Fig. 1, the compressor inlet air temperature is equal to the ambient air temperature when precooling effect is neglected. Using the polytropic relations for ideal gas and knowing the isentropic efficiency of the compressor, the compressor inlet temperature (T_1) is given by [2]:

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{r-1/r} \quad (1)$$

Where: T_{2s} is the isentropic temperature at the compressor outlet (K).

$\left(\frac{P_2}{P_1}\right)$ is the pressure ratio, r is the specific heat ratio, and, T_1 is the compressor inlet temperature (K)

The power required to drive the compressor can be calculated by [2-4]

$$\dot{W}_c = \dot{m}_a C_{pa}(T_2 - T_1) \quad (2)$$

$\dot{m}_a$ is the air mass flow-rate (kg/s),

C_{pa} is the specific heat capacity of dry air at constant pressure (KJ/Kg K) and

T_2 is the compressor exits temperature (K).

The heat delivered by combustion chamber is determined as [3-5]:

$$Q_{in} = C_{pa}(T_3 - T_2) \quad (3)$$

Where: T_2 is the combustion chamber inlet temperature (K), and T_3 is the combustion chamber outlet temperature (K).

The turbine discharge temperature (T_4) is defined as [6]:

$$T_4 = T_3 - \eta_T T_{4s}\left[1 \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{r-1}{r}}}\right] \quad (4)$$

Where:

η_T is the turbine isentropic efficiency (%),

$\left(\frac{P_3}{P_4}\right)$ is the pressure ratio, T_{4s} is the turbine discharge isentropic temperature (K).

The power generated by the turbine is defined as:

$$\dot{W}_T = \dot{m}_T C_{pg}(T_3 - T_{4s}) \quad (5)$$

Where:

$\dot{m}_T$ is the mass flow-rate of the charge and it is given as:

$$\dot{m}_T = \dot{m}_a + \dot{m}_f \quad (6)$$

Where**:**

$\dot{m}_a$ and $\dot{m}_f$ are mass flow-rate of air and fuel respectively,

C_{pg} is the specific heat capacity of the gas at constant pressure (KJ/Kg K).

The net power generated by the gas turbine plant is given as:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C \tag{7}$$

The specific fuel consumption (SFC) is given as:

$$SFC = \frac{3600\,\dot{m}_f}{\dot{W}_{net}} \tag{8}$$

The thermal efficiency of the gas turbine plant is given as:

$$\eta_{th} = \frac{3600}{SFC\ X\ NCV} \tag{9}$$

Where:

NCV is the Net fuel Calorific Value (KJ/Kg).

B. Description of the Gas Turbine Coupled with Inlet Air Precooling Techniques

Fig. 2. displays the schematics of the gas turbine attached to a precooling arrangement. It consists of a typical gas turbine plant and an inflow air pre-cooling system.

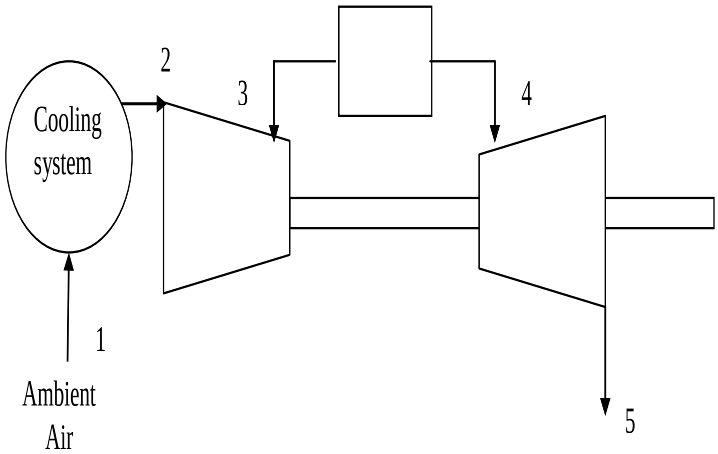

Fig. 2. Schematic diagram of a gas turbine with precooling techniques.

Evaporative pre-cooling techniques

The evaporative pre-cooling method uses the latent heat of vaporization to reduce ambient temperature. Air filter is placed before the evaporative pre-cooling equipment to prevent dust from entering the pre-cooling equipment and the compressor as shown in Figs. 3 (a) and (b).

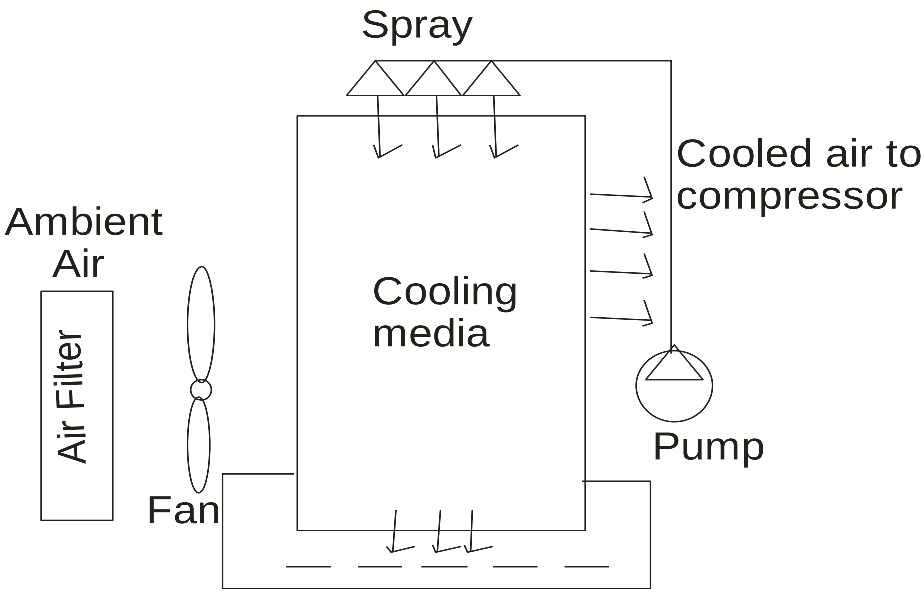

Fig. 3 (a) Typical Architecture of the evaporative precooling system.

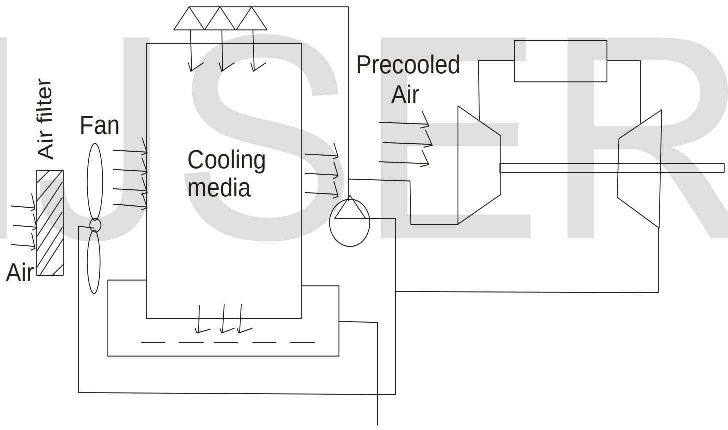

Fig. 3 (b). Schematic diagram of an evaporative precooling system integrated to a gas turbine plant.

The inlet air temperature after precooling is calculated by [4]

$$T_1 = T_{b2} - (T_{b2} - T_{w2})\varepsilon \quad (10)$$

Where:

T_{b2} is the dry-bulb temperature (K)

T_{w2} is the wet-bulb temperature (K),

ε is the evaporative precooling effectiveness (%)

Vapour Compression Precooling Techniques

Another option to provide gas turbine intake air precooling is the vapour compression precooling technique.

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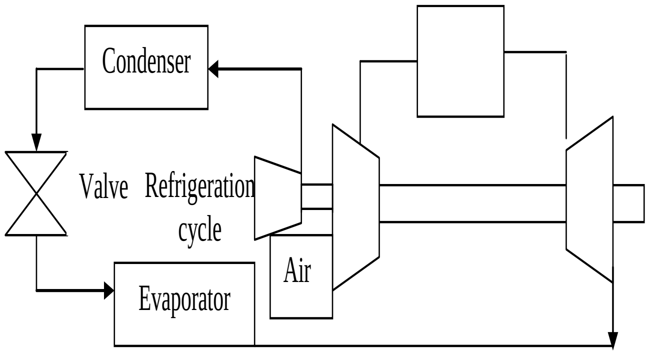

Fig. 4. Schematic diagram of a gas turbine plant integrated with vapour compression system.

The Net power output for the gas turbine integrated with vapour compression system is given as:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C - \dot{W}_C - \dot{W}_{mc} \tag{11}$$

Where:

$\dot{W}_T$, $\dot{W}_C$ and $\dot{W}_{mc}$ are the power generated by the turbine, power consumed by the compressor and power consumed by the mechanical chiller respectively.

The power consumption of the mechanical chiller is expressed as:

$$\dot{W}_{mc} = \frac{\dot{Q}_{CL}}{COP} \tag{12}$$

Where**:**

COP is the coefficient of performance of the mechanical chiller and it is equal to 32.2 for the plant studied.

$\dot{Q}_{CL}$ is the cooling load of the mechanical chiller and it is given as:

$$\dot{Q}_{CL} = \dot{m}_a \left(h_2 - h_3\right) \tag{13}$$

h_2 and h_3 is the enthalpy at the mechanical chiller inlet and outlet respectively.

Vapour absorption precooling techniques

The vapour absorption system consist of a pump, an absorber, a pressure reducing valve, a generator, condenser, receiver, expansion valve and evaporator. As the refrigerant leave the evaporator it is absorbed in a low temperature absorbing medium, rejecting some heat in the process. The refrigerant absorbent solution is then pumped at higher pressure and is heated in the generator thus causing the refrigerant vapour to separate from the solution as a result of the reduced solubility of the refrigerant – absorbent solution at high temperature and pressure,. The vapour passes to the condenser and the weakened refrigerant – absorbent solution is throttled back to the absorber.

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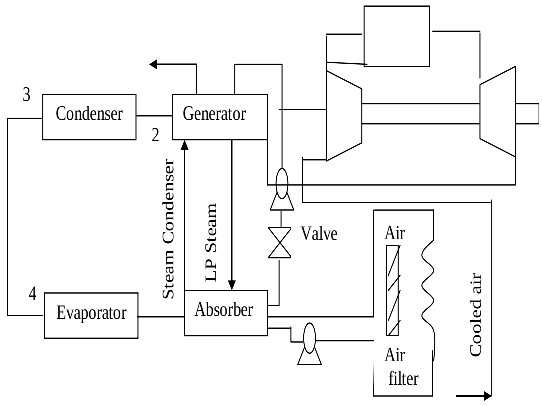

Fig. 5 Schematic diagram of a gas turbine plant integrated with a vapour absorption refrigeration system.

Assuming that the power required to drive the liquid solution pump is negligible; the net power generated by the gas turbine plant is given as [6]:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C - \dot{Q}_G \qquad (14)$$

Where:

$\dot{Q}_G$ is the power required to drive the generator.

Applying an energy balance equation to the vapour absorption cycle in Fig. 3, and neglecting pump work, we have

$$\dot{Q}_G + \dot{Q}_\varepsilon + \dot{Q}_c + \dot{Q}_A = 0 \qquad (15)$$

Where:

$\dot{Q}_\varepsilon$, $\dot{Q}_c$ and $\dot{Q}_A$ are heat supplied in the evaporator, condenser and absorber respectively and they are given as:

$$\dot{Q}_\varepsilon = \dot{m}_r \left(h_4 - h_1\right) \qquad (16)$$

$$\dot{Q}_c = \dot{m}_r \left(h_2 - h_3\right) \qquad (17)$$

$$\dot{Q}_A = \dot{m}_r \left(h_1 - h_2\right) \qquad (18)$$

Where:

h_1, h_2 , h_3 and h_4 are enthalpies at evaporator outlet, condenser outlet, condenser inlet, condenser outlet and evaporator inlet respectively.

Substituting Equations (16), (17) and (18) into equation (14), we have:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C - \dot{m}_r \left(h_4 - h_3\right) \qquad (19)$$

Where:

$\dot{m}_r$ is the mass flow-rate of the refrigerant – absorbent solution given as 0.2876Kg/s.

C. Economic Analysis

Intake air pre-cooling increases the power output which will also increase the revenue of the Gas Turbine plant. But, this increase in the revenue will be partially offset due to the installation and operational cost for the various pre-cooling systems.

The annual operation cost is a function of the operation period, t_{op}, cost of installation of the chiller, $\overset{c}{c}_{ch}$, and cooling coil, $\overset{c}{c}_{cc}$ and the electricity rate, $\overset{c}{c}_{el}$.

If the various precooling techniques consumes electrical power, T_e, and the electricity rate is $\overset{c}{c}_{el}$ (N/kWh) then the total annual cost can be expressed as [6]:

$$c_{total} = \overset{c}{a}\left[\overset{c}{c}_{el} + \overset{c}{c}_{cc}\right] + \int_{o}^{top} c_{el} T_e \, dt \tag{20}$$

Where:

T_e represents the electrical power consumed by the various pre-cooling systems. The capital recovery factor, $\overset{c}{a}$, is calculated as [6].

$$\overset{c}{a} = \frac{i(I+i)^n}{(I+i)^{n-1}} \tag{21}$$

Where: n is the specific period (years), and i is the interest rate (%).

D. Economic Analysis for Evaporative Precooling System

For evaporative precooling techniques, $\overset{c}{c}_{ch}$ and $\overset{c}{c}_{cc}$ are negligible, and T_e is taken as the electrical power consumed by the fan and pump; thus equation (20) is reduced to:

$$c_{total} = \overset{c}{a} + \int_{o}^{top} c_a \dot{W}_{pf} \, dt \tag{22}$$

Where:

$\dot{W}_{pf}$ is the power consumed by both pump and fan and it is given as:

$$\dot{W}_{pf} = \dot{W}_p + \dot{W}_f \tag{23}$$

$\dot{W}_p$ and $\dot{W}_f$ is power consumed by pump and power consumed by fan respectively.

According to Shanbghazani et al. [4], electrical power consumed by pump is given as:

$$\dot{W}_p = \dot{m}_w V_f \, \Delta p / \eta_{pump} \tag{24}$$

And that consumed by the fan is given as 10% of the power required to drive the pump

$$\dot{W}_f = 0.1 \dot{W}_p \tag{25}$$

Where**:**

$\dot{m}_w$ is the mass flow-rate of water (Kg/s), Δp is the pressure change in the flow of water (N/m^2), η_{pump} is the mechanical efficiency of the pump, (%) and V_f is the velocity of water flowing into the evaporative precooling header media.

E. Economic Analysis for Vapour Compression Precooling Technique

For vapour compression pre-cooling method, the power consumed by the compressor ($\dot{W}_{mc}$) replaces T_e in equation (25) thus the total cost is given as:

$$C_{total} = a^{c}\left[C^{c}_{ch} + C^{c}_{cc}\right] + \int_{o}^{top} c_{el}\, \dot{W}_{mc}\; dt \tag{26}$$

Where**:** $\dot{W}_{mc}$ is the electrical power required to drive the compressor.

F. Economic Analysis for Vapour Absorption Precooling Technique

For vapour absorption pre-cooling method, the power consumed is that required to drive the generator.
The total cost required to run the vapour absorption precooling system is given as:

$$C_{total} = a^{c}\left[C^{c}_{ch} + C^{c}_{cc}\right] + \int_{o}^{top} c_{el}\, \dot{W}_{Ge}\; dt \tag{27}$$

Where:

$\dot{W}_{Ge}$ is the power required to drive the generator (KW).

The chiller's purchase cost (C^{c}_{ch}) and cost of the cooling coil (C^{c}_{cc}) is obtained from vendors data or mechanical equipment cost index. For this study,

C^{c}_{cc} = N329, 868, 308 approximately

C^{c}_{ch} = N66, 897, 204 approximately.

G. Criteria for Evaluation of Gas Turbine Pre-cooling Method

To determine the feasibility of the system with a pre-cooling method, the plant performance with and without any pre-cooling techniques will be examined.
This can be achieved by determining the plant performance using different pre-cooling methods and the total additional cost resulting from using a pre-cooling system. The power gain ratio (equation 28) and the thermal efficiency change (equation 29) as proposed by Rahem et al. [6] and Achazmy et al [7], respectively was used for the analysis:

$$\text{PGR} = \frac{\dot{W}_{net}\,with\ cooling - \dot{W}_{net}\,without\ cooling}{W_{net}\,without\ cooling} x100 \tag{28}$$

TEC = $$\frac{\eta_{th}\ with\ cooling - \eta_{th}\ without\ cooling}{\eta_{th}\ without\ cooling} x\,100 \quad (29)$$

H. System Profitability

To evaluate the economic feasibility of the system with air intake pre-cooling method, the total cost of the pre-cooling technique and the increase in annual income resulting from the energy savings obtained from using a pre-cooling system is estimated.

The annual exported energy by the power plant fitted with a pre-cooling system is given by Rahem et al [6] as:

E(kWh) = $$\int_{o}^{top} W_{net}\,dt \quad (30)$$

If the gas turbine's annual electricity generation without a precooling system is $\varepsilon_{without\,cooling}$ and the precooling system increases the power generation to $\varepsilon_{with\,cooling}$, then the net increase in revenue due to the addition of the precooling system is

Net Revenue = $(\varepsilon_{with\ cooling} - \varepsilon_{without\ cooling})c_{els}$ (31)

The profitability of the power plant with a precooling system is defined as the increase in revenue due to the increase in electricity generation and it is given by:

Profitability = $(\varepsilon_{with\,cooling} - \varepsilon_{without\,cooling})C_{els} - C_{total}$ (32)

$(\varepsilon_{with\,cooling} - \varepsilon_{without\,cooling})C_{els}$ Gives the increase in revenue and C_{total} gives the annual expenses of the precooling system.

The profitability could be either positive which means an economic incentive for adding the precooling system, or negative meaning that there is no economic advantage despite the increase in the electricity generation of the plant.

III. RESULTS AND DISCUSSION

In order to establish a systematic comparison between the effects of the various air precooling techniques studied, the performance of the gas turbine unit is examined for a restricted set of operational and design conditions of the HITACHI – MS – 7001B Gas turbine taking into account real climatic conditions prevalent in Obrikom town, Omoku, Rivers State, Nigeria.

Effect of Ambient Temperature to the Net Power Generated by the Gas Turbine without any Precooling Method

Fig. 6 shows that, the Net Power Output decreases as the ambient temperature increases However, employing a pre-cooling techniques to cool the inlet air entering the compressor will enhance the Net Power Output of the gas turbine.

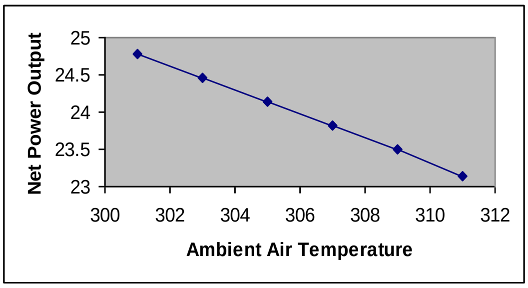

Fig. 6.Variation of Net power Output with respect to changes in ambient air temperature.

Effect of Ambient Temperature to the Power Consumed by the Compressor without any Precooling Technique

Fig. 7 shows the effect of increasing ambient air temperature to the work required to drive the compressor. This shows that as the ambient air temperature increases, the compressor consumes more power hence the performance (i.e. net power output) of the gas turbine reduces. To compensate this reduction in power output and higher work consumed by the compressor, there is a need for the compressor inlet air to be precooled to a desirable temperature so that the gas turbine will generate enough power required in the hot season.

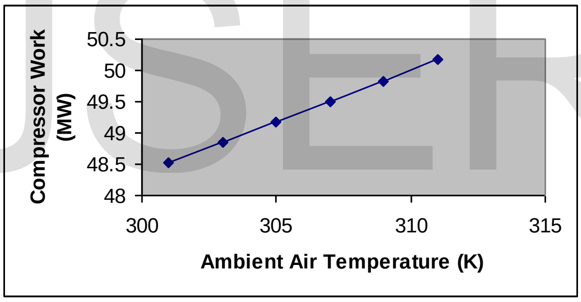

Fig. 7. Variation of Ambient Air temperature to Compressor work.

Evaporative Precooling Technique

Table II shows that evaporative pre-cooling method decreases the ambient temperature by about 2% and also reduces the power required to drive the compressor. Also the net power output increased by about 3%.

TABLE II
Output Parameters Obtained At Different Ambient Temperature Temperatures for Evaporative Precooling Techniques.

Ambient Temperature (K)	Precooled Temperature (K)	Compressor Work (Mw)	Specific Fuel Consumption (Kg/KWh)	Net Power Output (MW)	Power Gain Ratio (%)	Thermal Efficiency Change (%)
301	292.9	47.25	243700	26.07	5.24	3.55
303	294.4	47.45	245639	25.86	5.75	6.30
305	295.1	47.52	246239	25.80	6.92	7.0
307	296.2	47.66	247565	25.67	7.73	7.73
309	296.2	47.82	249157	25.50	8.53	8.53
311	297.3	47.93	250276	25.39	9.7	9.68

Effect of precooled air to power consumption of compressor

Fig. 8 shows that as air temperature is pre-cooled by about 2%, the power required to drive the compressor is reduced to about 1.23%. This increases the gas turbine efficiency compared to the reference case.

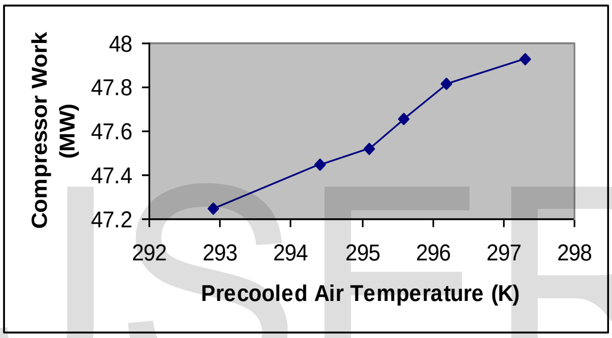

Fig. 8. Effect of precooled air temperature to power consumed by compressor for evaporative precooling technique.

Effect of Precooled Air Temperature to Power Gain Ratio and Thermal Efficiency Change for Evaporative Precooling

The Fig. 9 shows how the precooled air affects both the power Gain Ratio (PGR) and Thermal Efficiency Change (TEC). This shows that for every 2% reduction in temperature using the evaporative precooling techniques the gas turbine power Gain Ratio (PGR) and Thermal Efficiency Change (TEC) are approximately 5% and 3.2% respectively.

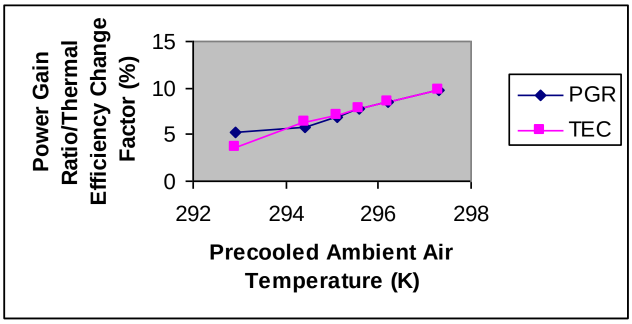

Fig. 9. Effect of precooled air temperature to power Gain Ratio (PGR) and thermal efficiency charge factor (TEC) with evaporative precooling techniques.

Vapour Compression Precooling Techniques

Table III shows the different output parameters for different precooled temperatures using the vapour compression precooling techniques. This that for vapour compression precooling techniques, more power gain ratio and thermal efficiency change factor is obtained as compared to evaporative precooling techniques.

Effect of Precooled Air to both Net Power Output and Compressor Work for Vapour Compression Precooling Techniques

TABLE III

Output Parameters Obtained At Different Ambient Temperatures For Vapour Compression Precooling Techniques.

Ambient Temperature (K)	Precooled Temperature (K)	Compressor Work (MW)	Specific Fuel Consumption (Kg/KWh)	Net Power Output (MW)	Power Gain Ratio (%)	Thermal Efficiency Change (%)
301	229	36.97	155354.5	33.17	33.8	34.1
303	230.9	37.23	193189.4	32.89	34.5	35.1
305	232.4	37.50	196262.2	32.56	35.61	35.64
307	233.9	37.72	196353.5	32.36	35.8	35.8
309	235.4	37.95	197906.9	32.11	36.6	36.8
311	236.9	38.2	199503.8	31.84	37.6	37.4

Figure 10 and 11 shows the effect of pre-cooled air using vapour compression pre-cooling techniques on net power output and compressor work respectively. Figure 10 shows that vapour compression pre-cooling method increased the Net Power Output by about 28% compared to that without pre-cooling. Figure 11 also shows that for pre-cooled air, the compressor work reduced as compared to the one without pre-cooled air.

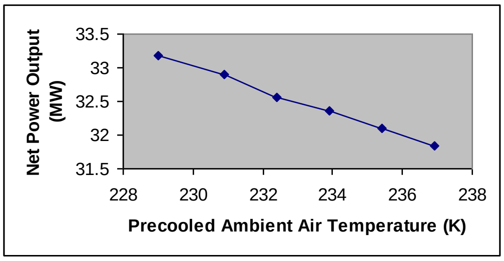

Fig. 10. Effect of precooled air to net power output using vapour compression precooling techniques.

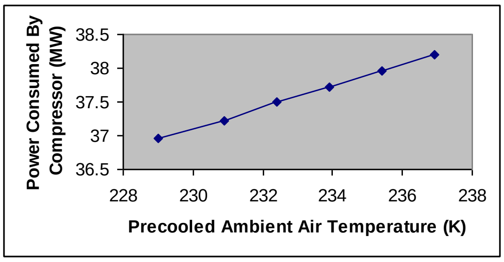

Fig. 11. Effect of precooled air to compressor work using vapour compression precooling techniques.

Fig. 12 explains that for every 3% pre-cooling of ambient air using vapour compression pre-cooling techniques, there is about 13% increment in both power gain ratio (PGR) and thermal efficiency change factor (T E C), respectively.

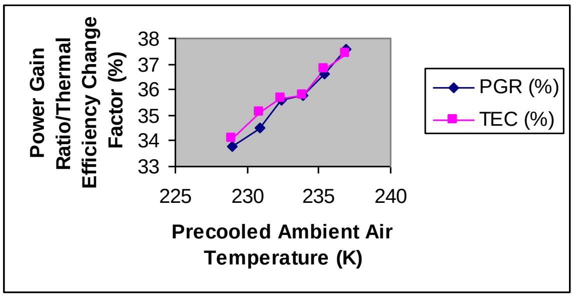

Fig. 12. Effect of pre-cooled air to power gain ratio (pgr) and efficiency change factor (tec) using vapour compression pre-cooling technique.

Vapour Absorption Precooling Techniques

Table IV shows the variation of power consumed by compressor (W_c), net power output (W_{net}), specific fuel consumption (SFC), power Gain Ratio (PGR) and Thermal Efficiency change factor (TEC) for different ambient temperatures (T_a).

TABLE IV
Output parameters obtained at different ambient temperatures for absorption precooling techniques.

Ambient Temperature (K)	Precooled Temperature (K)	Compressor Work (MW)	Specific Fuel Consumption (kg/kWh)	Net Power Output (MW)	Power Gain Ratio (%)	Thermal Efficiency Change (%)
301	229	36.97	1755324.9	36.21	46.0	46.2
303	231	37.23	179647.2	35.97	47.1	47.7
305	232	37.45	177734.27	35.75	48.1	48.1
307	233.9	37.72	179120.13	35.47	48.9	48.9
309	235.4	37.95	180225.3	35.25	50.0	50.9
311	236.9	38.2	182011.5	34.9	51.0	51.0

Figs. 13 and 14 show that as the pre-cooled temperature of the air increases, net power output decrease and the power required to driving the compressor increases.

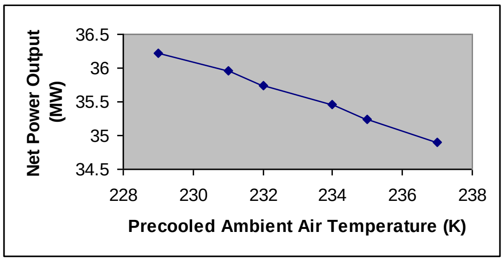

Fig. 13. Effect pre-cooled air to net power output using vapour absorption pre-cooling techniques

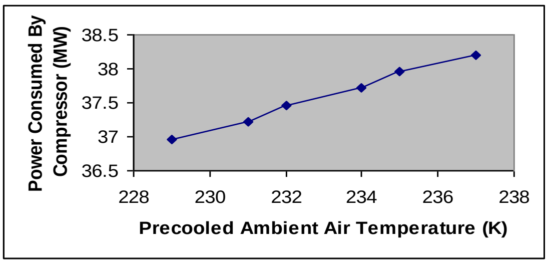

Fig. 14. Effect of precooled air to power consumed by compressor using vapour absorption precooling techniques.

Fig. 15 shows that the power gain ratio (PGR) and thermal efficiency charge factor (TEC) increase at approximately the same rate for vapour absorption pre-cooling techniques.

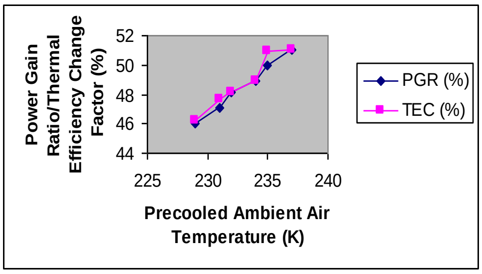

Fig. 15. Effect precooled air temperature to power gain ratio (PGR) and thermal efficiency change factor (TEC).

Comparative Studies for Evaporative Precooling, Vapour Compression Precooling and Vapour Absorption Precooling Techniques

Figs. 16 and 17 show the variation of net power output and power gain ratio for the different pre-cooling systems. Vapour absorption pre-cooling system records the highest Power output and power gain ratio compared to other precooling methods.

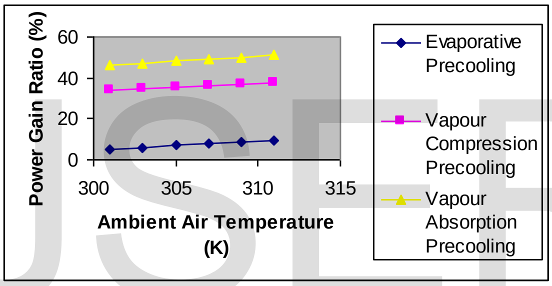

Fig. 16. Variation of power gain ratio (%) for the various precooling techniques.

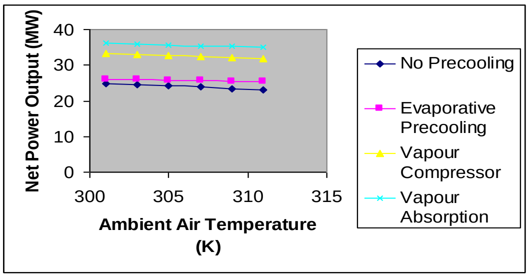

Fig. 17. Comparison of net power output for various precooling techniques at different ambient air temperatures

TABLE V
Power gain ratio (PGR) for the various precooling techniques at different ambient air temperature.

Ambient Temperature (K)	Evaporative precooling system	Vapour compression precooling system	Vapour absorption precooling system
301	5.24	33.8	26.0
303	5.75	34.5	47.1
305	6.92	32.61	48.1
307	7.73	32.8	48.9
309	8.53	36.6	50.0
311	9.70	37.6	51.0

Table V shows the PGR factor for the turbine for the different pre-cooling methods adopted at different in let air temperatures to the pre-coolers. From the table, it shows that vapour absorption precooling methods will give the highest power gain ratio (PGR) for the different pre-cooling techniques studied.

System Profitability and Revenue for the various Precooling Techniques (Economic Analysis)

Figs. 18 and 19 show that vapour absorption precooling technique will record the highest system profitability and Net Revenue respectively than the vapour compression and evaporative precooling technique; though vapour compression precooling technique will record a higher result than the evaporative precooling technique.

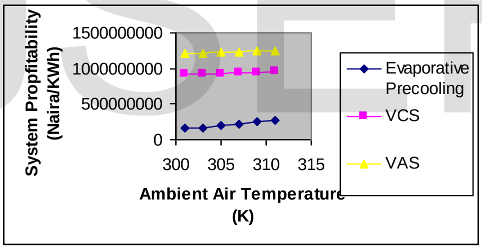

Fig. 18. Variation of the system profitability for the various precooling techniques at different inlet air temperatures.

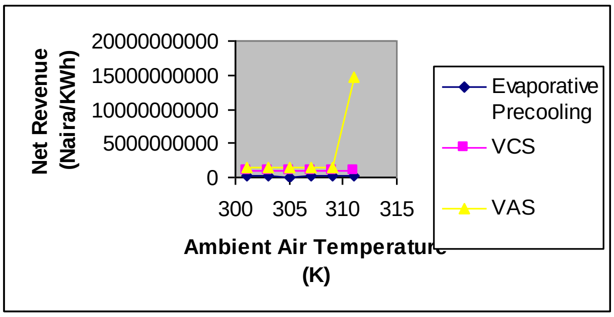

Fig. 19. Variation of net revenue for the different precooling techniques at different ambient air temperatures.

Total Annual Cost Analysis

The total annual cost (C_{total}) for evaporative precooling technique (Fig. 20) is N4590236.1/kWh; that of vapour compression and vapour absorption precooling methods are N12576095.9/kWh and N213218642/kWh respectively.

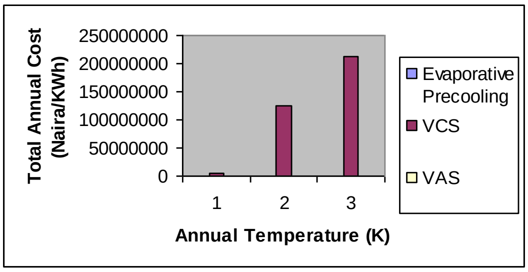

Fig. 20. Bar chart showing the total annual cost for the various precooling techniques

IV. Conclusion

Gas turbine plant performance in Nigerian energy utility sector has been studied using different inlet air pre-cooling techniques from thermodynamics and economics approach. The performance when evaporative precooling technique is used depends mainly on the ambient conditions. The evaporative pre-cooling system works effectively during hot climatic conditions. The results obtained at different ambient temperatures show that evaporative pre-cooling system increased the power output by over 7%, with increase in machine efficiency of about 2.7%. Vapour compression precooling system increased the power output by about 10%, with increase in machine efficiency of about 5.2%. While vapour absorption pre-cooling system increased the power output by 13%, with increase in machine efficiency of about 8.7%.

The total annual gas turbine output power gain and the total annual cost due to cooling by evaporative, vapour compression and vapour absorption inlet air pre-cooling are 5.24% and N4590236.1/KWh; 33.8% and N12576095.9/kWh; 46% and N213218642/kWh, respectively.

Evaporative precooling method is more cost effective; has low maintenance cost and low energy consumption when compared to vapour absorption inlet air precooling. It is therefore recommended that evaporative precooling technique should not be utilized in the Northern part of Nigeria because in this region, there is less abundance of water supply.

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