THERMODYNAMIC ANALYSIS OF COMPRESSOR INLET AIR PRECOOLING TECHNIQUES OF A GAS TURBINE PLANT OPERATIONAL IN NIGERIA ENERGY UTILITY SECTOR

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ABSTRACT

Thermodynamic analysis of inlet air pre-cooling techniques of compressor is among the methods for the enhancement of gas turbine performance. This study compared the effect of using evaporative cooling system pre-cooling method, vapour compression refrigeration precooling and vapour absorption refrigeration precooling techniques to the gas turbine Net Power Output, Thermal efficiency, Thermal Efficiency Change factor (TEC) and Power Gain Ratio (PGR) taking into recognition the prevalent weather and climatic conditions of Nigeria and as well as optimization parameters for the reference system (i.e. without precooling techniques). The results show that at air temperature of 311K, the reference system, evaporative precooling, vapour compression refrigeration and vapour absorption refrigeration precooling methods recorded Net power Outputs of 23.143MW, 25.39MW, 31.84MW and 34.90MW respectively. The Thermal Efficiency Change factor recorded by the precooling systems at an ambient temperature of 311K is 8.68%, 37.4% and 51% respectively.

Keywords: Thermodynamics; Inlet air; Turbine; Efficiency; Refrigeration; Energy.

1. INTRODUCTION

Nigeria as a case study does not have a homogeneous climate and weather conditions. This indicates that the condition of climate in the Northern part is relatively different from Southern part. Based on this, the precooling techniques employed in the southern region of Nigeria may not be favourable in the Northern region. This study is important because it compares different pre-cooling methods and recommends the method that will be appropriate for a particular region with regard to the particular climatic conditions. It is a well-known fact that gas turbines produce less power when the ambient temperature is hot. According to Wash and Fletcher (2004), at high inlet air temperature (i.e. 26^{o}C and above), the air density is significantly reduced and consequently, the mass flow-rate of the inlet air to the compressor is reduced. Turbine inlet air precooling is defined as the precooling of the air before entering the compressor. The main purpose of turbine inlet air precooling is to increase the net power output of the gas turbine when the ambient air temperature is higher than the standard conditions. According to Al-Tori (2009), the rated capacities of all gas turbine are usually based on the standard ambient conditions of 15^{o}C and 14.7psi at sea level. Alok et al. (2012), poses that greatest losses in output power of a gas turbine is usually during the periods when the electricity demand is greatest. A gas turbine can lose about 7% of its nominal power when the intake temperature ranges from 15^{o}C to 25^{o}C, and when the ambient temperature is above 25^{o}C, the loss may reaches about 15%. By cooling the air at the compressor inlet below that of the ambient the plant performance during high ambient temperatures will be improved. The compressor work decreases as the inlet air temperature decreases, thus increasing the overall cycle efficiency. However, a drop in the inlet air temperature increases air density which in turn increases the mass flow-rate of air entering the compressor resulting in enhanced power output. The inlet air precooling techniques investigated in this study are evaporative precooling and mechanical precooling (vapour compression and vapour absorption refrigeration precooling) techniques. In the evaporative pre-cooling method, the inlet air cools as it comes in contact with a cooling fluid, such as fog, water sprays, or a combination of both, (Wang, 2009).

2. MATERIALS AND METHODS

For this research, an open cycle HITACHI – MS – 7001B Gas turbine plant located in Nigeria Agip Oil Company plant yard in Obrikom, Omoku, Rives State, Nigeria was considered. Its operating data were collected from the daily turbine control log sheet for a period of ten years. The plant was divided into different control units namely compressor, turbine and precooling units.

Thermodynamic principles were used on each of the units to determine the gas turbine performance. The performance of the plant without pre-cooling technique was determined first and then its performance when each of the cooling methods (evaporative cooling, vapour compression and vapor absorption cooling) were used to pre-cooled the inlet air before entering the plant.

2.1. Description of the Plant

Fig. 3.1 shows the diagram of a single shaft gas turbine (HITACHI –MS – 7001B) cycle, the type considered in this work. The compressor compressed the ambient air thus raising its temperature before it enters the combustion chamber. Fuel is sprayed into the high temperature air in the combustion chamber. Combustion of fuel takes place and the resulting hot gas expands resulting in mechanical work which drives the turbine shaft.

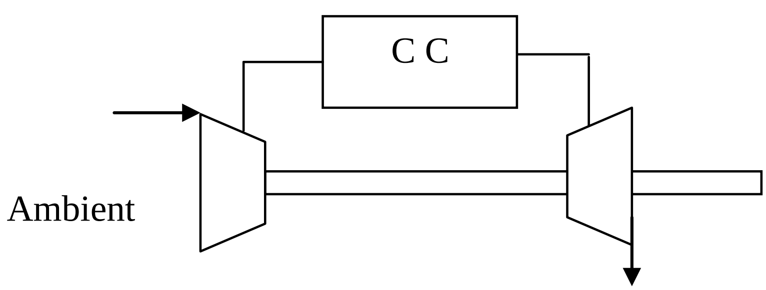

Figure 3.1: Diagram of a gas turbine cycle.

2.2. The Gas Turbine Unit

Based on Fig. 3.1 above, the compressor inlet air temperature is the ambient air temperature when precooling effect is neglected. Applying the polytropic relations for ideal gas and knowing the isentropic efficiency of the compressor, the compressor inlet temperature (T_1) is given by:

This section should provide enough detail to allow full replication of the study by suitably skilled investigators. Protocols for new methods should be included, but well-established protocols may simply be referenced.

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

Where:

T_{2s} is the isentropic temperature at the compressor outlet (K); $\frac{P_2}{P_1}$ represents the ratio of the pressure; r is the specific heat ratio, and T_1 is the compressor inlet temperature (K).

The power to drive the compressor can be calculated by

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

Where:

$\dot{m}_a$ is mass flow-rate of air (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 generated during combustion by is determined as:

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

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

The turbine discharge temperature (T_4) is defined as (Rahem *et al.*, 2006):

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

Where:

η_T is isentropic efficiency of the turbine (%),

$\left(\frac{P_3}{P_4}\right)$ is the ratio of the pressure,

T_{4s} is the turbine discharge isentropic temperature (K).

The power generated by the turbine is defined as:

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

Where:

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

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

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:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C \qquad (7)$$

The specific Fuel Consumption (SFC) is given as:

$$\text{SFC} = \frac{3600\,\dot{m}_f}{\dot{W}_{net}} \qquad (8)$$

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

$$\eta_{th} = \frac{3600}{SFC\ X\ NCV} \qquad (9)$$

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

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

Fig. 3.2. Shows a schematic diagram of the gas turbine with a precooling system. It comprises of a standard gas turbine plant and an inlet air precooler.

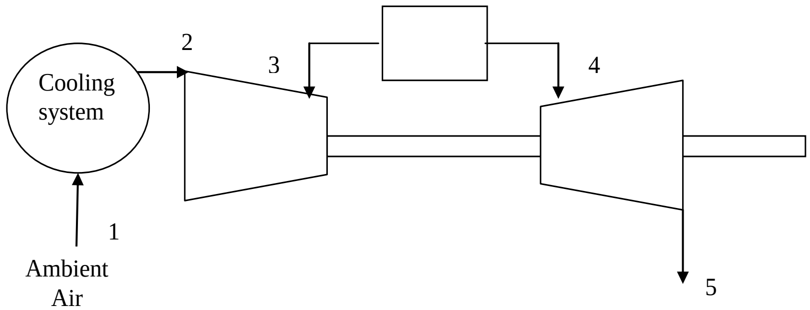

Figure 3.2: Schematic diagram of a gas turbine cycle with precooling techniques.

2.4. Evaporative Precooling Techniques

In evaporative method of pre-cooling, the sensible heat energy of air is utilized to evaporate water and in the process the air temperature drops below the ambient. According to Meher-Homji (2000), in evaporative pre-cooling of turbine inlet air, droplets of water, 5 – 20 microns in diameter, are sprayed into the air inlet ducts at 1000 – 3000 psia. As the droplets evaporate, the air is cooled to the wet bulb temperature. Air filter are incorporated to prevent dust from entering the pre-cooling equipment and the compressor as shown in Figs. 3.3 (a) and (b)

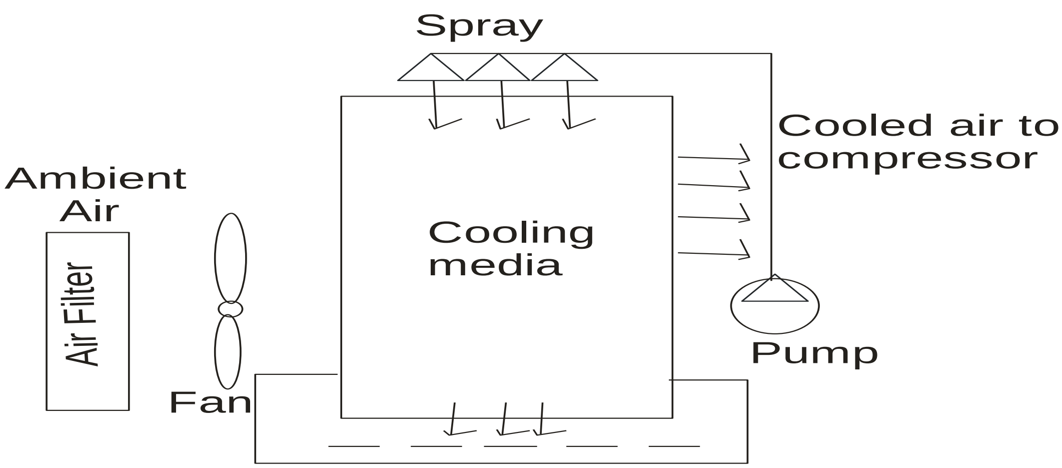

Figure 3.3 (a): Typical Architecture of the evaporative precooling system.

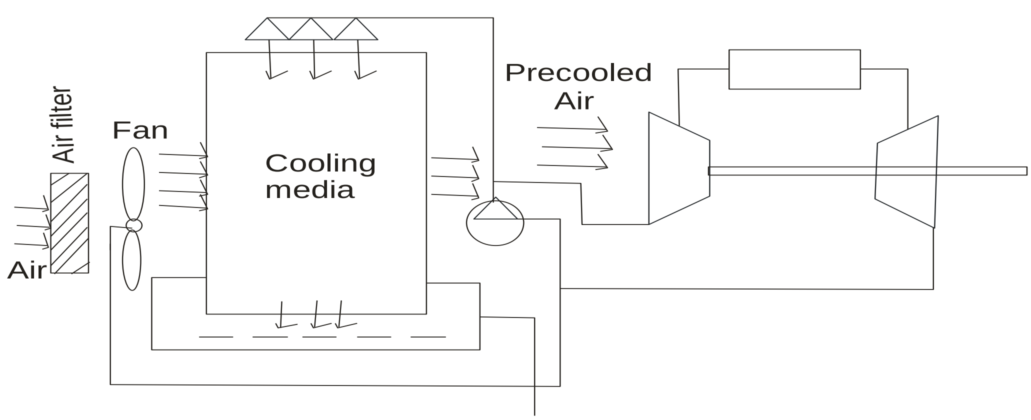

Figure 3.3 (b): Schematic diagram of an evaporative precooling system integrated to a gas turbine plant.

The inlet air temperature after precooling is calculated by: (Shanbghazani et al., 2008)

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

Where:

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

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

ε is the evaporative precooling effectiveness (%)

2.5. Vapour Compression Precooling Techniques

Vapour compression refrigeration inlet air precooling system utilizes vapour compression refrigeration equipment (compressor). Its operation/maintenance cost are lower than that of vapour absorption pre-cooling system but the high capital costs and power requirements is approximately 30% of the power gain. In this cycle the refrigerant, evaporates and condensates at suitable pressure for practical equipment design. The refrigerant used for the vapour compression refrigeration analysis in this research is ammonia (NH_3), R22.

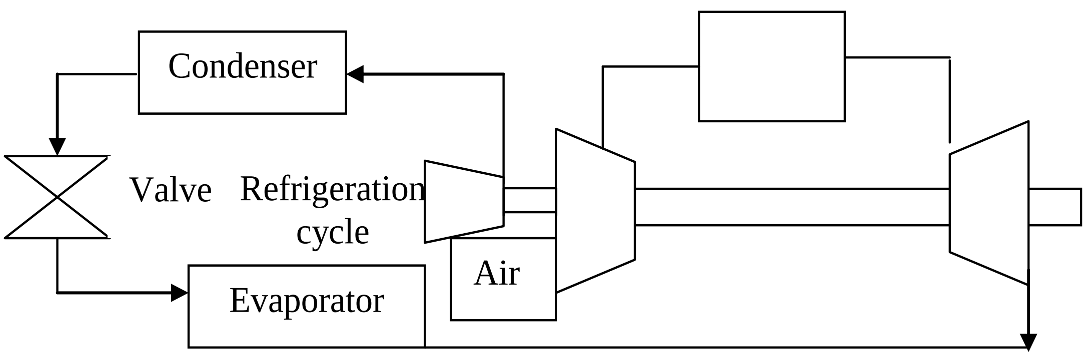

Figure 3.4: Schematic diagram of a gas turbine plant integrated with vapour compression system.

The Net Power Output for the gas turbine integrated with vapour compression system is given as:

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

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 (i.e. refrigeration cycle compressor) respectively. Power consumed by the mechanical chiller is given as:

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

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:

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

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

2.6. Vapour Absorption Precooling Techniques

This system consists of a pump, an absorber, a pressure reducing valve and a generator. Other components are condenser, receiver, expansion valve and evaporator. The refrigerant leaves the evaporator and enters a low temperature absorbing medium. During this process heat is given off and the absorbent solution is pumped at higher pressure. As a result of the reduced solubility of the refrigerant – absorbent solution at high temperature and pressure, refrigerant vapour is separated from the solution. The vapour passes to the condenser and the weakened refrigerant – absorbent solution is throttled back to the absorber. In this study, the refrigerant is ammonia while the absorbent is water.

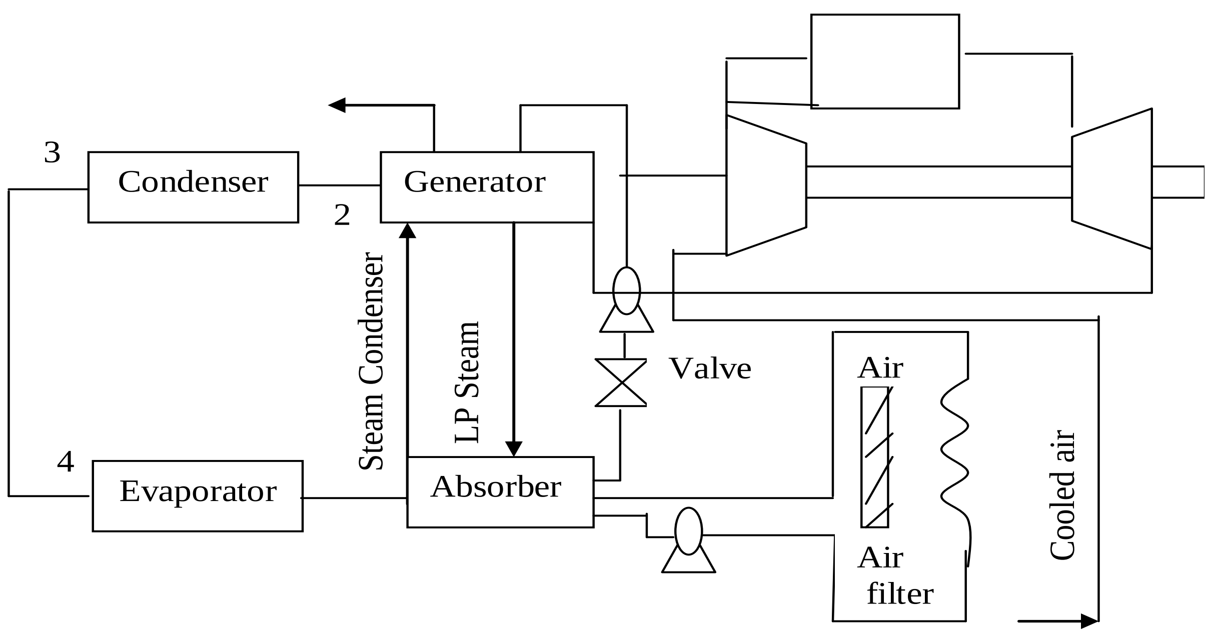

Figure 3.5: Schematic diagram of a gas turbine plant integrated with a vapour absorption refrigeration system.

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 (Rahem et al., 2006):

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C - \dot{Q}_G \tag{14}$$

Where:
Q_G is the power required to drive the generator.
Applying an energy balance equation to the vapour absorption cycle in Figure 3, and neglecting pump work, we have

$$\dot{Q}_G + \dot{Q}_\varepsilon + \dot{Q}_c + \dot{Q}_A = 0 \tag{15}$$

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:

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

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

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

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

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

Where:

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

2.7. Evaluation Criteria of Gas Turbine Precooling System

In order to evaluate the feasibility of a precooling system coupled to a Gas turbine plant, it is important to examine the performance of the plant with and without any precooling techniques.
In order to study the performance of a gas turbine fitted with a precooling method the power gain ratio (PGR) and the thermal efficiency change (TEC) as proposed by Rahem et al. (2006) and Achazmy et al. (2006) respectively is given as:

$$\textbf{PGR=} \frac{\dot{W}_{net}\ with\ cooling - \dot{W}_{net}\ without\ cooling}{W_{net}\ without\ cooling} x100 \tag{20}$$

$$\textbf{TEC =} \frac{\eta_{th}\ with\ cooling - \eta_{th}\ without\ cooling}{\eta_{th}\ without\ cooling} x\,100 \tag{21}$$

3. RESULTS AND DISCUSSIONS

The performance of the Gas turbine plant integrated with evaporative precooling system, vapour compression precooling system and vapour absorption precooling system are investigated. Fig. 4.1 describes the effect of ambient temperature to the gas turbine Net Power output (N.P.O). At an ambient temperature of 303K, the gas turbine recorded a Net Power Output of 38MW, 36MW, 25MW and 22MW for vapour absorption precooling, vapour compression precooling, evaporative precooling and when no precooling technique is involved respectively. Furthermore, at an ambient temperature of 311K, the Net Power Output for the reference system (i.e. no precooling system), evaporative precooling, vapour compression precooling and vapour absorption precooling are 22MW, 26MW, 31MW and 36MW respectively.

Table 4.1: Net power output obtained at different ambient temperatures for the reference system (without precooling) and the precooled systems.

Ambient temperature (K)	Net Power Output			
	Reference plant	Evaporative precooling	Vapour compression	Vapour absorption
301	24.79	26.07	33.17	36.21
303	24.46	25.86	32.89	35.97
305	24.14	25.80	32.56	35.75
307	23.82	25.67	32.36	35.47
309	23.50	25.50	32.11	35.25
311	23.14	25.39	31.84	34.90

Table 4.2: Power gain ratio obtained at different ambient temperatures for the precooling systems.

Ambient temperature (K)	Power Gain Ratio (%)		
	Evaporative precooling	Vapour compression	Vapour absorption
301	5.24	33.80	46.20
303	5.75	34.50	47.70
305	6.92	32.61	48.10
307	7.73	32.80	48.90
309	8.53	36.60	50.00
311	9.70	37.60	51.00

Table 4.3: Thermal efficiency change factor obtained at different ambient temperatures for the precooling systems.

Ambient temperature (K)	Thermal Efficiency Change Factor (%)		
	Evaporative precooling	Vapour compression	Vapour absorption
301	3.55	34.10	46.20
303	6.30	35.10	47.70
305	7.00	35.64	48.10
307	7.73	35.80	48.90
309	8.53	36.80	50.00
311	8.68	37.40	51.00

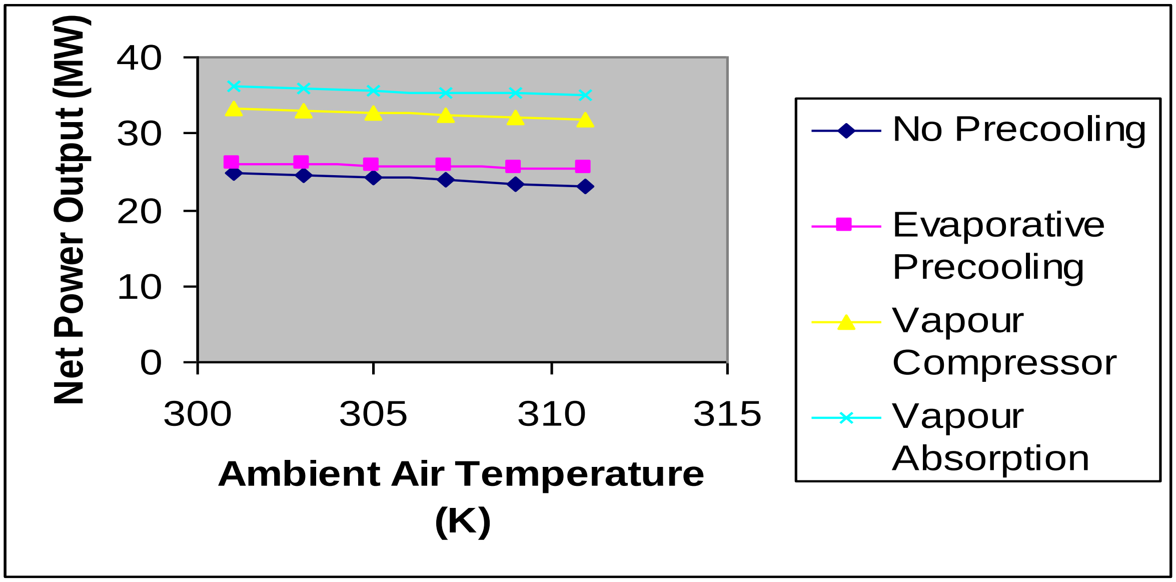

Figure 4.1: Comparison of net power output for various precooling techniques at different ambient air temperatures.

At precooled ambient temperature of 311K, the Power Gain Ratio of the evaporative, vapour compression and vapour absorption are 10%, 38% and 51% respectively (Fig. 4.2). This indicates that as the ambient temperature is reduced, the compressor consumes less work hence the net power output is increased.

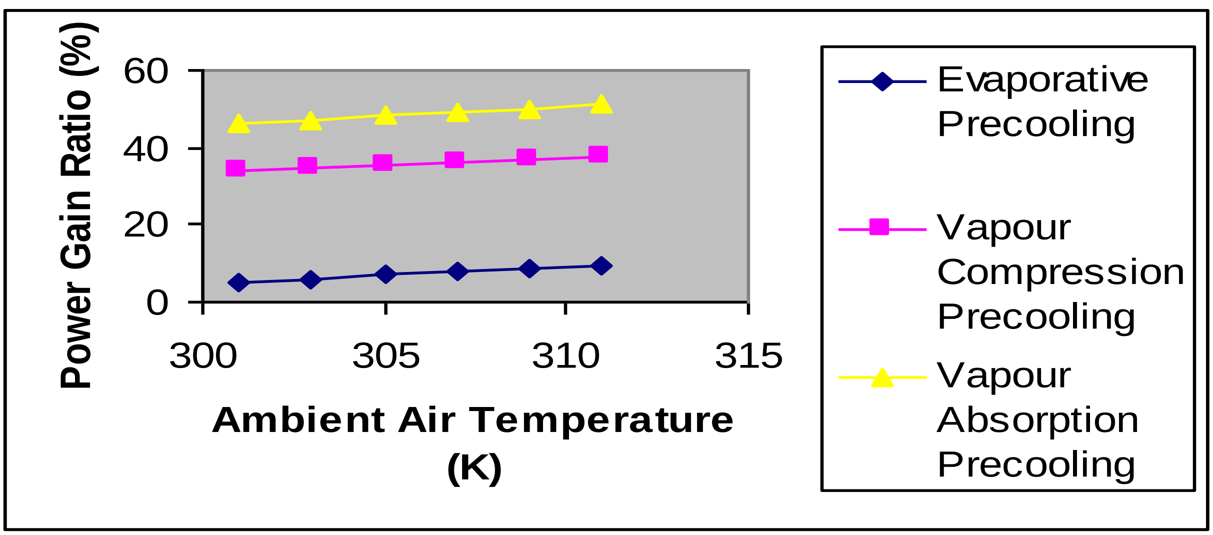

Figure 4.2: A graph of Power Gain Ratio versus Precooled Ambient Temperature

Fig. 4.3 shows the comparison of the thermal efficiency change factor for the various compressor inlet air precooling techniques. At a temperature of 311K, evaporative, vapour compression and vapour absorption precooling techniques measured thermal efficiency changes of 8.68%, 37.4% and 51% respectively. This means that the more the reduction in ambient temperature, the increase in thermal efficiency.

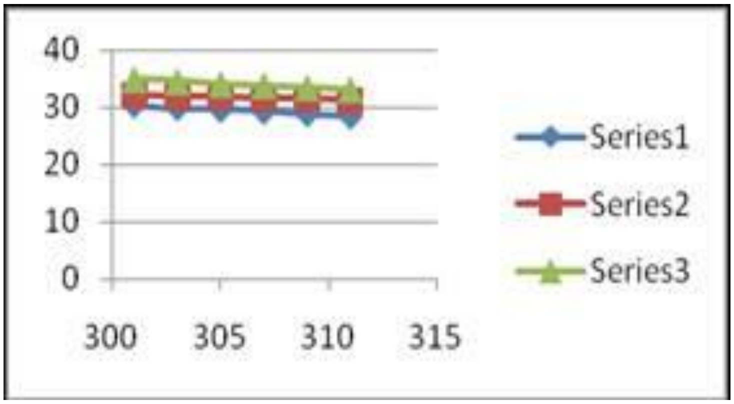

Fig. 4.4 shows that the gas turbine thermal efficiency is affected by ambient air temperature due to the change in air density and compressor work. A lower ambient air temperature will lead to a higher air density and a lower compressor. At a temperature of 311K, the thermal efficiency for evaporative, vapour compression and vapour absorption precooling techniques are 31.28%, 32.94% and 33.09% respectively.

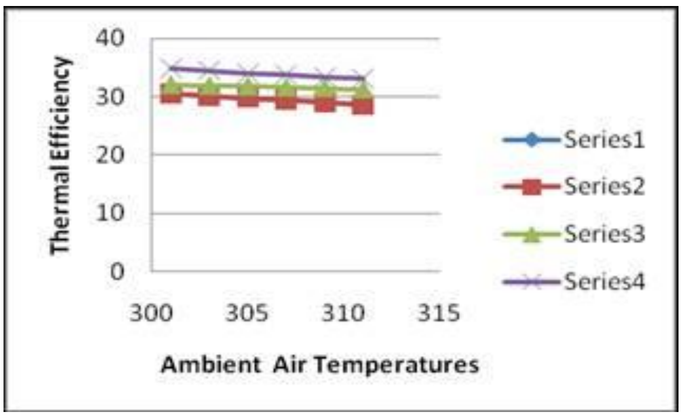

Figure 4.4: A graph of Thermal Efficiency against Ambient Air Temperature.

4. Conclusions and Recommendations

The performance of a gas turbine modeled with inlet air precooling techniques operational in Nigeria energy utility sector was studied and the following inferences were made:

1) The evaporative, vapour compression and vapour absorption precooling systems recorded net power outputs of 25.39W, 31.84MW and 34.90MW respectively at an ambient temperature of 311K.
2) At the ambient temperature of 311K, evaporative, vapour compression and vapour absorption recorded a Power Gain Ratio of 9.70%, 37.6% and 51% respectively.

The above detailed records show that vapour compression and vapour absorption refrigeration precooling techniques will record higher net power output than evaporative precooling because refrigeration precooling techniques cools ambient air beyond the Wet-bulb-temperature (WBT) but vapour absorption refrigeration precooling will record the highest net output power because less parasitic work is consumed unlike in vapour compression precooling system in which more parasitic work is consumed (i.e. power required to drive the refrigeration compressor). This research therefore recommends that since evaporative precooling system is favourable in dry and humid areas, it should be used in the northern part of Nigeria provided that the supply of water will not generate a seasonal bottleneck whereas the refrigeration precooling systems can be utilized in the southern region of Nigeria.

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