International Journal of Energy, Environment, and Economics ISSN: 1054-853X
Volume 24, Number 2-3

ASSESSMENT OF EFFECT OF OPERATION PARAMETERS ON GAS TURBINE POWER PLANT PERFORMANCE USING FIRST AND SECOND LAWS

S. O. Oyedepo[1,*], R. O. Fagbenle[2], S. S. Adefila[3], and M. M. Alam[4]

[1]Mechanical Engineering Department, Covenant University, Ota, Nigeria
[2]Mechanical Engineering Department, Obafemi Awolowo University, Ile-Ife, Nigeria
[3]Chemical Engineering Department, Covenant University, Ota, Nigeria
[4]Institute for Turbulence-Noise-Vibration Interaction and Control, Shenzhen Graduate School, Harbin Institute of Technology, Shenzhen, China

ABSTRACT

In this study, thermodynamics modelling of gas turbine engine is performed based on thermodynamic relations. The thermodynamic model reveals that the influence of operation parameters including the compression ratio, turbine inlet temperature and ambient temperature has significant effect on the performance of gas turbine engine. Energy and exergy analyses were conducted to evaluate performance of the selected power plant and to assess the effect of operating parameters on energy loss and exergy destruction in the plant. Energy analysis shows that the turbine has the highest proportion of energy loss (31.98%) in the plant. The exergy analysis results reveal that the combustion chamber is the most exergy destructive component compared to other cycle components. In addition, it was found that increase in the gas turbine inlet temperature (GTIT) decreases the exergy destruction of this component. The effects of design parameters on exergy efficiency show that an increase in the compression ratio and TIT increase the total exergy efficiency of the cycle due to a rise in the output power of the turbine and a decrease in the combustion chamber losses. The overall exergetic efficiency of the plant decreased with increased ambient temperature. It was found that a 5 K rise in ambient temperature resulted in a 1.03% decrease in the overall exergetic efficiency of the plant. Based on the results of this research work, the possible economical methods and technologies to improve performance of the selected gas turbine power plant are suggested.

Keywords: thermodynamics model, GTIT, exergetic efficiency, ambient temperature, gas turbine engine, simulation

* Correspondence Email: Sunday.oyedepo@covenantuniversity.edu.com

NOMENCLATURE

Symbols

c_p	specific heat at constant pressure, [kJ/kg]
$\dot{E}$	exergy rate, [kW]
$\dot{E}_L$	exergy loss rate
$\dot{E}_D$	exergy destruction rate
$\dot{m}$	mass flow, [kg/s]
P	Power output, [kW]
pe	potential energy[kJ]
r_p	pressure compression ratio
R	gas constant [kJ/mol – K]
y_D	exergy destruction rate ratio

Greek Symbols

γ	adiabatic index
Δp_{cc}	pressure drop in combustion chamber (bar)
η_c	isentropic efficiency of compressor
η_T	isentropic efficiency of turbine
η_{th}	thermal efficiency
ϵ	exergetic efficiency
$\emptyset$	rational efficiency
δ	component efficiency defect
ψ	overall exergetic efficiency
ξ	exergetic performance coefficient

Subscripts

i	inlet
e	exit or outlet
p	pressure
a	air
pg	combustion product
f	fuel
T	turbine
cc	combustion chamber
th	thermal
sys	system
0	ambient
cv	control volume
D	destruction

gen	generation
ac	air compressor
gt	gas turbine
k	Component

Superscripts

tot	total
PH	physical
KN	kinetic
PT	potential
CHE	chemical
T	thermal
P	mechanical

1. INTRODUCTION

Energy is an important factor in wealth generation, economic and social development in any nation. Based on historical data, there is a strong relationship between economic activities and availability of energy resources (Reddy et al., 2013). Gas turbines have come to play a significant role in distributed energy systems due to its multi-fuel capability, compact size and low environmental impact and reduced operational and maintenance cost. Growing demand of power and degradation of environment have made gas turbine power plants of scientific interest for the efficient utilization of energy resources (Ghazikhami et al., 2014).

A gas turbine as a device designed to convert the heat energy of fuel into useful work is different from steam turbine in the sense that there is no change of phase in the working fluid used in gas turbine, whereas there is change of phase (liquid to steam) in working fluid used in steam turbine. Gas turbines are steady- flow power machines in which a gas (usually air) is compressed, heated and expanded for the purpose of generating power. The term turbine is the component which delivers power from the gas as it expands; it is also called an expander. The term gas turbine refers to a complete power machine (Kreith and Goswaini, 2005).

Gas turbines utilized in electric-power generation are manufactured in two classes which are heavy-duty and aero derivative. These two classes of turbines have different performance, cost, partial load modelling, as well as different performance variations with the ambient temperature (Chaker et al., 2003).

Gas turbines (GTs) have been used for electricity generation in most countries around the world. In the past, their use has generally been limited to generate electricity in periods of peak electricity demand. Gas turbines are ideal for this application as they can be started and stopped quickly, enabling them to be brought into service as required to meet energy demand peaks (Jaber et al., 2007). However, due to availability of natural gas at relatively cheap prices compared to distillate fuels, many countries around the world, e.g., Nigeria, use large conventional GTs as base load units (Oyedepo and Kilanko, 2014).

In gas turbines, since the air for combustion is taken directly from the environment, their performance is strongly affected by both external factors (ambient temperature and relative

humidity) and internal factors (components efficiencies, turbine inlet temperature, compression pressure ratio etc.). Efficiency and electric-power output of gas turbines vary according to the ambient conditions. The amount of these variations greatly affects electricity production, fuel consumption and plant incomes (Erdem and Sevilgen, 2006). Power rating can drop by as much as 20 to 30%, with respect to International Standard Organization (ISO) design conditions, when ambient temperature reaches 35 to 45°C (Mahmoudi et al., 2009; Guinee, 2001). One way of restoring operating conditions is to add an air cooler at the compressor inlet (Sadrameli and Goswami, 2007). The air cooling system serves to raise the gas turbine performance to peak power levels during the warmer months when the high atmospheric temperature causes the turbine to work at off-design conditions with reduced power output (Kakaras et al., 2004; Kamal and Zubair, 2006). From external factors point of view, the effect of turbine inlet temperature (TIT) is predominant. According to Rahman et al. (2011) and Ameri et al., (2007), for every 56°C increase in TIT, the power output increases approximately 10% and gives about 1.5% increase in efficiency. Overall efficiency of the gas turbine cycle depends primarily upon the pressure ratio of the compressor.

For comparative purpose, the International Standards Organization (ISO) has established standard conditions which are universally accepted and used for gas turbine performance. Standard air conditions in gas turbine designing at sea level, 25°C temperature and 60% relative humidity (Hall et al., 1994). Power output while operating in these conditions is termed as the standard power. Analyses performed by previous researchers showed that operating below this temperature improved performance and operating above this temperature degraded performance (Bassily, 2001; Gareta et al., 2005).

The performance of thermal power plant operating at off design, from the thermodynamic viewpoint, can be evaluated by the first (energy) and second (exergy) laws. The energy based criteria provides a quantitative interpretation of the thermodynamic analysis, while the exergy based criteria is associated to qualitative information, describing the system in its critical points by the irreversibilities (losses) occurred in the process (Lior and Zhang, 2007). Hence, exergy-based criteria are considered more appropriate for assessing energy system performance as they account better for use of energy resources and give much better guidance for system improvement. They also can be converted to exergy cost efficiencies if the exergy values of the useful outputs and paid inputs can be rationally priced.

In recent years numerous and extensive researches have been conducted to evaluate thermal power systems from both energy and exergy analyses point of view (Bilgen, 2000, Ray et al., 2007, Khaldi and Adouane, 2011, Chen et al., 2014).

The prime objectives of this study are:

1. to model and simulate gas turbine engine using MATLAB R2010a
2. to investigate the effect of variation of operation conditions on performance of gas turbine engine
3. to evaluate performance of selected gas turbine power plant using first and second laws of thermodynamics
4. to investigate the effect of variation of ambient temperature and TIT on energy loss and exergy destruction of the selected gas turbine power plant

2. MATERIALS AND METHODS

2.1. Data Collection

In this work, AES barges gas turbine plant unit PB204 with installed capacity of 31.5 MW was selected for study. The plant is situated on the lagoon jetty, at the PHCN Egbin Thermal Station premise, in Ijede, a suburb of Ikorodu Town in Lagos, Nigeria.

Operating data for the gas turbine unit were collected from the daily turbine control log sheet for a period of five years (2006-2010). The daily average operating variables were statistically analyzed and mean values were computed for the period of January to December, followed by an overall average. A summary of the operating parameters of the PB204 unit used for this study is presented in Table 1. The analysis of the plant was divided into different control volumes and performance of the plant was estimated using component-wise modeling. Mass, energy conservation laws and exergy balance were applied to each component and the performance of the plant was determined for the system.

Table 1. Average operating data for the selected gas turbine power plant

S/No	Operating Parameters	Unit	Value
1	Ambient Temperature, T_1	K	303.63
2	Compressor outlet temperature, T_2	K	622.31
3	Turbine inlet temperature, T_3	K	1218.62
4	Turbine outlet temperature, T_4	K	750.00
5	Exhaust gas temperature, T_{exh}	K	715.00
6	Compressor inlet pressure, P_1	bar	1.013
7	Compressor outlet pressure, P_2	bar	9.80
8	Pressure ratio	-	9.67
9	Mass flow rate of fuel	kg/s	2.58
10	Inlet mass flow rate of air	kg/s	125.16
11	Power output	MW	29.89
12	LHV of fuel	kJ/kg	47,541.57

2.2. Gas Turbine Plant Simulation with MATLAB R2010a

Gas turbine engine was simulated using MATLAB codes developed by the authors. This was done to investigate the effect of variation of operating conditions on performance of gas turbine engine. Gas turbine cycle was modelled by using each component (compressor, combustion chamber and turbine) governing thermodynamics and chemical relations. Cumulative performance indices such as thermal efficiency, power output, specific fuel consumption, heat supplied and net - work output were calculated.

2.3. Power Plant Component Energetic and Exergetic Analyses

The energetic and exergetic efficiencies of the entire unit that makes up the selected gas turbine plant were evaluated using MATLAB R2010a and Microsoft Excel 2010. For the purpose of investigating the effect of interaction of the plant‘s units on the energetic and exergetic efficiencies, the thermal power plant unit was then grouped into subsystem and overall system, as clearly marked out in Figure 1. The energy and exergy balances on inlet and exit streams of each process unit were used in the estimation of their energetic and exergetic efficiencies.

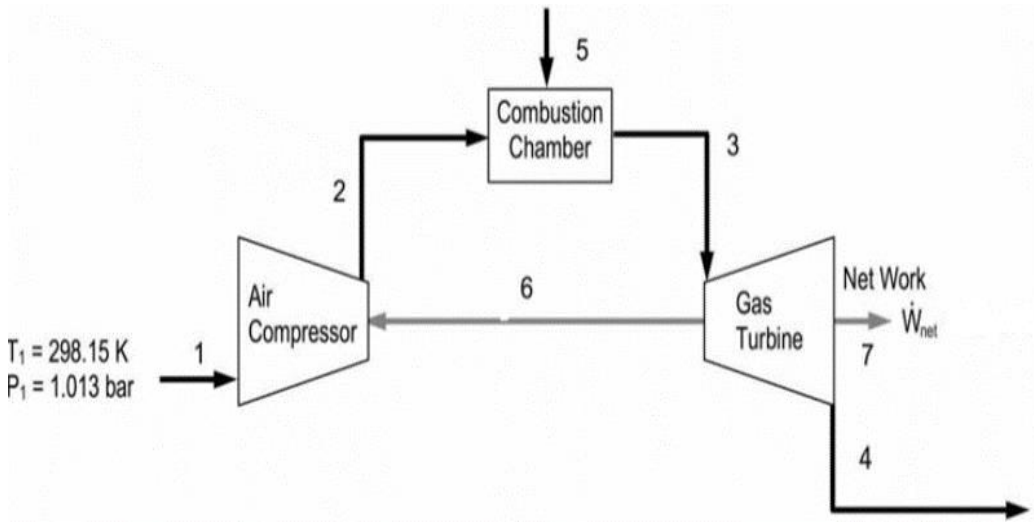

Figure 1. A schematic diagram for a simple GT cycle.

2.4. Thermodynamic Modelling of Gas Turbine Engine

The major components of simple gas turbine (GT) are 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. Equations (1) to (40) depict the governing thermodynamics models for simple gas turbine engine. In this study, energy and exergy models are considered for gas turbine performance assessment.

2.4.1. Energy (First Law of Thermodynamics) Analysis

Using the first law of thermodynamics for a thermal system, it is possible to calculate the cycle thermal efficiency, which is the ratio of the work output to the heat input. Also, the energy loss in each component and the entire plant can be computed using energy balance.

For any control volume at steady state with negligible potential and kinetic energy changes, energy balance reduces to (Barzegar et al., 2011):

$$\dot{Q} - \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \tag{1}$$

The energy balance equations for various components of the gas turbine plant shown in Figure 1 are as follows:

2.4.1.1. Air Compressor Model

The compression ratio (r_p) can be defined as:

$$r_p = \frac{P_2}{P_1}, \tag{2}$$

where p_1 and p_2 are the compressor inlet and outlet air pressure, respectively

The isentropic efficiency for the compressor is expressed as:

$$\eta_C = \frac{\left[(r_p)^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right]}{\left(\frac{T_2}{T_1} - 1\right)}, \tag{3}$$

where T_1 and T_2 are the compressor inlet and outlet air temperatures respectively.

Compressed air temperature can be written in terms of the pressure ratio and the inlet compressor temperature as:

$$T_2 = T_1\left[1 + \frac{(r_p)^{\frac{(\gamma_a - 1)}{\gamma_a}} - 1}{\eta_c}\right], \tag{4}$$

where T_2, is the temperature in K of the compressed air entering combustion chamber and η_c, is the compressor's isentropic efficiency.

At full load, the compressor work rate, W_c can be written in terms of the pressure ratio and the inlet compressor temperature as:

$$\dot{W}_c = \frac{\dot{m}_a C_{pa} T_1}{\eta_c}\left((r_p)^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right), \tag{5}$$

where c_{pa} is the specific heat capacity of air which is considered in this study as a function of temperature and can be fitted by Equation (6) for temperature in the range of 200K<T<800K (Rahman et al., 2011; Kurt et al., 2009):

$$c_{pa}(T) = 1.04841 - \left(\frac{3.8371T}{10^4}\right) + \left(\frac{9.4537T^2}{10^7}\right) - \left(\frac{5.49031T^3}{10^{10}}\right) + \left(\frac{7.9298T^4}{10^{14}}\right) \tag{6}$$

The energy input to the air compressor at ambient temperature is calculated by:

$$\dot{Q}_{c1} = \dot{m}_a\left(c_{pa}T_2 - c_{pa}T_a\right), \tag{7}$$

where T_a is the ambient temperature

Energy input to the air compressor at specific inlet temperature (T_1) is given as:

$$\dot{Q}_{c2} = \dot{m}_a\left(c_{pa}T_2 - c_{pa}\overline{T}_1\right) \tag{8}$$

The energy loss in the compressor due to the inlet air temperature difference is given as:

$$\dot{Q}_{c\,loss} = \dot{m}_a\left(c_{pa}T_2 - c_{pa}T_a\right) - \dot{m}_a\left(c_{pa}T_2 - c_{pa}\overline{T}_1\right) \tag{9}$$

where $\overline{T}_1 = \frac{T_a + T_2}{2}$

2.4.1.2. Combustion Chamber Model

The energy balance in the combustion chamber is given by (Ibrahim and Rahman, 2012):

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f\left(LHV + C_{pf}T_f\right) = (\dot{m}_a + \dot{m}_f)C_{pg}T_3 \tag{10}$$

Energy loss in the combustion chamber is determined using equation (11):

$$\dot{Q}_{CC\,loss} = \dot{m}_a C_{pa} T_2 + \dot{m}_f LHV - \dot{m}_f C_{pg} T_3 \tag{11}$$

where $\dot{m}_f$, is fuel mass flow rate (kg/s), $\dot{m}_a$ is air mass flow rate (kg/s), LHV is low heating value, T_3 is turbine inlet temperature (K) C_{pf} is specific heat of fuel and T_f is temperature of fuel (K). C_{pg} is the specific heat capacity of combustion product (gas) which is considered in this work to be a temperature variable function and can be fitted by Equation (12) for temperature in the range of 1000K <T< 1500K (Tahouni et al., 2012; Kurt et al., 2009):

$$c_{pg}(T) = 0.991615 + \left(\frac{6.99703T}{10^5}\right) + \left(\frac{2.7129T^2}{10^7}\right) - \left(\frac{1.22442T^3}{10^{10}}\right) \tag{12}$$

From equation (9), the fuel – air ratio (f) is expressed as:

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg}T_3 - C_{pa}T_1(1 + r_{pg})}{LHV + C_{pf}T_f - C_{pg}T_3}, \tag{13}$$

where,

$$r_{pg} = \frac{(r_p)^{\frac{(\gamma_a - 1)}{\gamma_a}} - 1}{\eta_c} \tag{14}$$

The pressure drop across the combustion chamber (ΔP_{cc}) is usually around 2% (Adrian and Dorin, 2010; Barzegar et al., 2011). The Turbine inlet pressure (P_3) can be calculated as:

$$P_3 = P_2(1 - \Delta P_{cc}), \tag{15}$$

where P_3, turbine entry level pressure in Pa; P_2 is the combustion chamber inlet temperature, ΔP_{cc}, is pressure drop across the combustion chamber.

2.4.1.3. Gas Turbine Model

The isentropic efficiency for turbine can be written in terms of the turbine pressure ratio, the turbine inlet temperature and turbine exit temperature as:

$$\eta_T = \frac{1-\left(\frac{T_4}{T_3}\right)}{1-(r_T)^{\frac{1-\gamma_g}{\gamma_g}}}, \tag{16}$$

where r_T is the turbine pressure ratio: $r_T = P_3/P_4$.

The exhaust gases temperature from the gas turbine is given as:

$$T_4 = T_3\left\{1-\eta_T\left[1-\left(\frac{P_3}{P_4}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right]\right\} \tag{17}$$

The shaft work rate of the turbine is written in terms of the pressure ratio and the turbine inlet temperature as:

$$\dot{W}_T = \dot{m}_g c_{pg} T_3 \eta_T\left[1-(r_T)^{\frac{1-\gamma_g}{\gamma_g}}\right] \tag{18}$$

The network rate of the gas turbine is given in terms of the pressure ratio, compressor inlet temperature and turbine inlet temperature as:

$$\dot{W}_n = \dot{m}_g c_{pg} T_3 \eta_T\left[1-(r_T)^{\frac{1-\gamma_g}{\gamma_g}}\right] - \frac{\dot{m}_a c_{pa} T_1}{\eta_C}\left[(r_p)^{\frac{\gamma_a-1}{\gamma_a}}-1\right] \tag{19}$$

where

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{20}$$

c_{pg} is the specific heat capacity of combustion product (gas) and it is given as in (12).

The power output is expressed in terms of the pressure ratio, compressor inlet temperature and turbine inlet temperature as:

$$P = m_g\left[c_{pg} T_3 \eta_T\left(1-(r_p)^{\frac{1-\gamma_g}{\gamma_g}}\right) - \frac{c_{pa} T_1}{\eta_C}\left((r_p)^{\frac{\gamma_a-1}{\gamma_a}}-1\right)\right] \tag{21}$$

Energy input in the turbine is given as:

$$\dot{Q}_T = \dot{m}_g c_{pg} T_3 \tag{22}$$

Energy utilized for turbine work is calculated as follows:

$$\dot{Q}_{Tw} = \dot{m}_g\left(c_{pg} T_3 - c_{pgTET} TET\right), \tag{23}$$

where T_3 is the combustion chamber exit temperature and TET is turbine exhaust temperature.

Energy loss from the turbine is given as:

$$\dot{Q}_{Tloss} = \dot{m}_g(c_{pgTET}TET) \tag{24}$$

The total energy loss in the turbine system is calculated by:

$$\dot{Q}_{TTloss} = \dot{Q}_{C\ losss} + \dot{Q}_{CC\ loss} + \dot{Q}_{T\ loss} \tag{25}$$

The gas turbine thermal efficiency (η_{th}) can be determined by Equation (26):

$$\eta_{th} = \frac{\dot{W}_n}{\dot{m}_f LHV} \tag{26}$$

Equation (25) is also known as the first law efficiency of gas turbine (Mozafari, et al., 2010).

The specific fuel consumption (*SFC*) is determined by:

$$SFC = \frac{3600}{W_n} f, \tag{27}$$

where f (fuel –air mass ratio) is given is given by Equation (12).

The heat rate (HR) (i.e the consumed thermal energy to generate unit energy of electrical energy) can be expressed as:

$$HR = \frac{3600}{\eta_{th}} \tag{28}$$

2.4.2. Exergy (Second Law of Thermodynamics) Analysis

The second law of thermodynamics complements and enhances the analysis of energy system by enabling calculation of the real thermodynamic inefficiencies and losses from the system being considered. The exergy method is based on the second law of thermodynamics according to which complete transformation of heat into work is not possible (Alcides, 1999).

A general exergy - balance equation applicable to any component of a thermal system may be formulated by utilizing the first and second laws of thermodynamics (Oh et al., 1996; Ebadi and Gorji – Bandpy, 2005). The thermo-mechanical exergy stream may be decomposed into its thermal and mechanical components so that the balance in rate form gives:

$$\dot{E}_i^{PH} - \dot{E}_e^{PH} = \left(\dot{E}_i^T - \dot{E}_e^T\right) + \left(\dot{E}_i^P - \dot{E}_e^P\right), \tag{29}$$

where the subscripts i and e represent inlet and exit states; $\dot{E}^{PH}$ is the physical exergy of a material stream, $\dot{E}^T$ is the thermal component of the exergy stream, $\dot{E}^P$ is the mechanical component of the exergy stream, the term on the left – hand side of the equation represent the change in exergy of the flow stream, the first and second terms on the right – hand side of the equation represent the changes in the thermal and mechanical components of the exergy stream respectively.

The thermal and mechanical components of the exergy stream for an ideal gas with constant specific heat may be written respectively as (Ebadi and Gorji – Bandpy, 2005, Abam et al., 2011):

$$\dot{E}^T = \dot{m}\, c_p \left[(T - T_0) - T_0 ln \frac{T}{T_0}\right] \tag{30}$$

and

$$\dot{E}^P = \dot{m} R T_0 ln \frac{P}{P_0}\,, \tag{31}$$

where P_0 and T_0 are the pressure and temperature, respectively, at standard state; $\dot{m}$ is the mass flow rate of the working fluid; R is the gas constant, c_p is the specific heat at constant pressure.

In steady state, exergy balance for control volume is given as (Bejan et al., 1996; Kotas, 1995):

$$\dot{E}_x = \sum_j \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_j + \dot{W}_{CV} + \sum_i m_i e_i - \sum_e m_e e_e \tag{32}$$

The subscripts i, e, j and 0 refer to conditions at inlet and exits of control volume boundaries and reference state. Equation (32) can be written as:

$$E_i^{tot} - E_e^{tot} - E_D = 0 \tag{33}$$

Equation (33) implies that the exergy change of a system during a process is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of irreversibilities.

The exergy-balance equations and the exergy destroyed during the process taking place in each component of the power plant are written as follows:

Air Compressor

$$\dot{E}^{WAC} = \left(\dot{E}_1^T - \dot{E}_2^T\right) + \left(\dot{E}_1^P - \dot{E}_2^P\right) + T_0\left(\dot{S}_1 - \dot{S}_2\right) \tag{34a}$$

$$\dot{E}_{DAC} = T_0\left(\dot{S}_2 - \dot{S}_1\right) = \dot{m} T_0 \left[c_{p1-2} ln\left({T_2}/{T_1}\right) - R ln\left({P_2}/{P_1}\right)\right] \tag{34b}$$

Combustion Chamber

$$\dot{E}^{CHE} + \left(\dot{E}_2^T + \dot{E}_5^T - \dot{E}_3^T\right) + \left(\dot{E}_2^P + \dot{E}_5^P - \dot{E}_3^P\right) + T_0\left(\dot{S}_3 - \dot{S}_2 + \dot{S}_5 + \frac{\dot{Q}_{CC}}{T_0}\right) = 0 \tag{35a}$$

$$\dot{E}_{DC} = T_0\left[\dot{S}_3 - \dot{S}_2 + \dot{S}_5 + \frac{\dot{Q}_{CC}}{T_0}\right]$$

$$=\dot{m}T_0\left\{\begin{matrix}\left(c_{p2-3}ln\left({T_3}/{T_2}\right)-Rln\left({P_3}/{P_2}\right)\right)+\left(c_{p5}ln\left({T_5}/{T_0}\right)-Rln\left({P_5}/{P_0}\right)\right)+\\ \frac{c_{p2-3}(T_3-T_2)}{T_{in\,CC}}\end{matrix}\right\} \quad (35b)$$

Gas Turbine

$$\dot{E}^{WGT}=\left(\dot{E}_3^T-\dot{E}_4^T\right)+\left(\dot{E}_3^P-\dot{E}_4^P\right)+T_0\left(\dot{S}_3-\dot{S}_4\right) \quad (36a)$$

$$\dot{E}_{DGT}=\dot{m}T_0\left[c_{p3-4}ln\left({T_4}/{T_3}\right)-Rln\left({P_4}/{P_3}\right)\right] \quad (36b)$$

For a control volume at steady state, the exergetic efficiency is

$$\varepsilon=\frac{\dot{E}_P}{\dot{E}_F}=1-\frac{\dot{E}_D+\dot{E}_L}{\dot{E}_F} \quad (37)$$

where the rates at which the fuel is supplied and the product is generated are denoted by $\dot{E}_F$ and $\dot{E}_P$ respectively. $\dot{E}_D$ and $\dot{E}_L$ denote the rates of exergy destruction and exergy loss, respectively.

The ith component efficiency defect denoted by δ_i is given by Equation (38) (Abam et al., 2011):

$$\delta_i=\frac{\sum\Delta\dot{E}_{Di}}{\sum\Delta\dot{E}_{xin}} \quad (38)$$

where, $\sum\Delta\dot{E}_{Di}$ is the sum of change in total rate of exergy destruction and $\sum\Delta\,\dot{E}_{xin}$ is the sum of change in total rate of exergy flow into the plant.

The overall exergetic efficiency of the entire plant is given as:

$$\psi_i=\frac{\dot{W}_{net}}{\dot{E}_{x\,fuel}} \quad (39)$$

The amount of exergy loss rate per unit power output as important performance criteria is given as:

$$\xi=\frac{\dot{E}_{D\,Total}}{\dot{W}_{net}} \quad (40)$$

where ξ is the exergetic performance coefficient.

Exergy destruction rate and efficiency equations for the gas turbine power plant components and for the whole cycle are summarized in Table 2.

Table 2. The exergy destruction rate and exergy efficiency equations for gas turbine

Component	Exergy Destruction Rate	Exergy Efficiency
Compressor	$\dot{E}_{DC} = \dot{E}_{in} - \dot{E}_{out} + \dot{W}_C$	$\varepsilon = \frac{\dot{E}_{out} - \dot{E}_{in}}{\dot{W}}$
Combustion Chamber	$\dot{E}_{DCC} = \dot{E}_{in} - \dot{E}_{out} + \dot{E}_{fuel}$	$\varepsilon = \frac{\dot{E}_{out}}{\dot{E}_{in} - \dot{E}_{fuel}}$
Gas Turbine	$\dot{E}_{DT} = \dot{E}_{in} - \dot{E}_{out} - (\dot{W}_{net} + \dot{W}_C)$	$\varepsilon = \frac{\dot{W}_{net} + \dot{W}_C}{\dot{E}_{in} - \dot{E}_{out}}$
Total exergy destruction rate	$\dot{E}_{DTotal} = \sum \dot{E}_D = \dot{E}_{DC} + \dot{E}_{DCC} + \dot{E}_{DT}$	

3. Results and Discussion

3.1. Effects of Operating Conditions on Performance of Gas Turbine Plants

The simulation results of the effect of operation conditions on gas turbine power plant performance are presented in this section. The effects of operation conditions on the power output, heat rate, specific fuel consumption and efficiency are obtained by the computational model developed by the energy balance utilizing the MATLAB codes (MATLAB R2010a software). In this study, the effect of operating atmospheric conditions on gas turbine is considered purposely to show sensitivity of gas turbine performance to the environmental condition. The results of this section as presented in Figures 2 to 13 are based on theoretical relationships earlier presented.

3.1.1. Effect of Compression Ratio

Figure 2 shows the relation between cycle thermal efficiency and compression ratios for turbine inlet temperatures (TITs) between 900K and 1500K. It can be seen that the thermal efficiency linearly increases at lower compression ratio as well as higher TIT until certain value of compression ratio. The maximum cycle temperature (TIT) is limited by metallurgical considerations. The blades of the turbine are under great mechanical stress and the temperature of the blade material must be kept to a safe working value (Eastop and McConkey, 2009). The temperature of the gases entering the turbine can be raised, provided a means of blade cooling is available. In aircraft practice where the life expectancy of the engine is shorter, the maximum temperatures used are usually higher than those used in industrial and marine gas turbine units; more expensive alloys and blade cooling allow maximum temperatures of above 1600K to be attained.

The variation of thermal efficiency is more significant at higher compression ratio and lower turbine inlet temperature. However, at lower TITs, there is decrement in thermal efficiency drastically with increase in compression ratio. Further, as the turbine inlet temperature increases, the peaks of the curves flatten out giving a greater range of ratios of pressure optimum efficiency.

Figure 3 presents the effect of compression ratio and ambient temperature (T1) on the thermal efficiency of gas turbine engine. It is observed that thermal efficiency increases with increase compression ratio but decreases with ambient temperature (T1). The variation in thermal efficiency with increase in compression ratios at different ambient temperatures is not

significant. A comparison between the results from the present study and standard thermodynamics text books (Eastop and McConkey, 2009; Rajput, 2007) reveals an acceptable agreement.

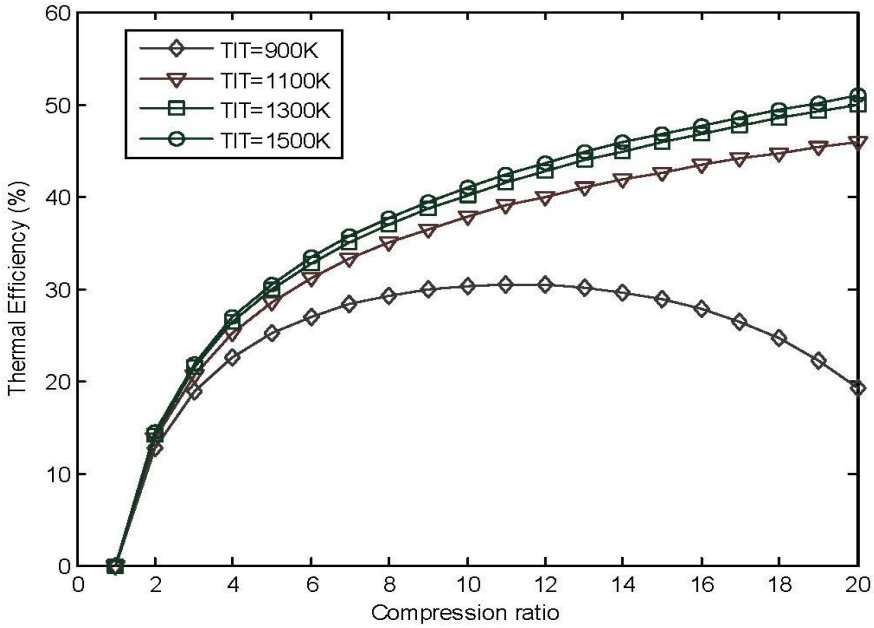

Figure 2. Effect of Compression ratio and TIT on thermal efficiency.

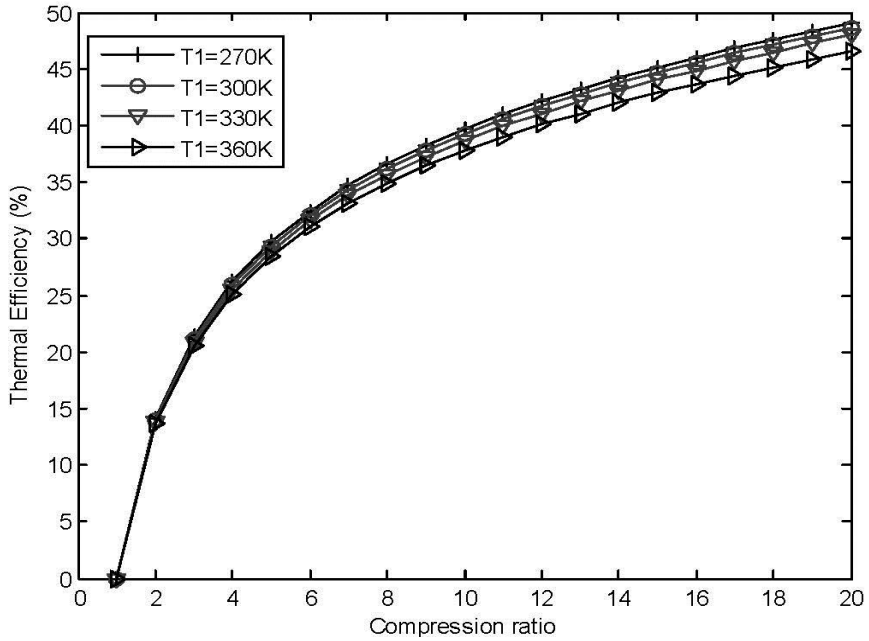

Figure 3. Effect of Compression Ratio and ambient temperature (T1) on Thermal Efficiency.

Figure 4 shows the effect of compression ratio r_p and turbine inlet temperature TIT on specific fuel consumption (SFC). It is observed that the SFC decreases linearly with increase of compression ratio up to about $r_p = 3$ and that within this r_p range SFC also decreases with increase in TIT. As r_p increases, both the compression work (input) and turbine work (output) increase, the net power output rises upward and then declines.

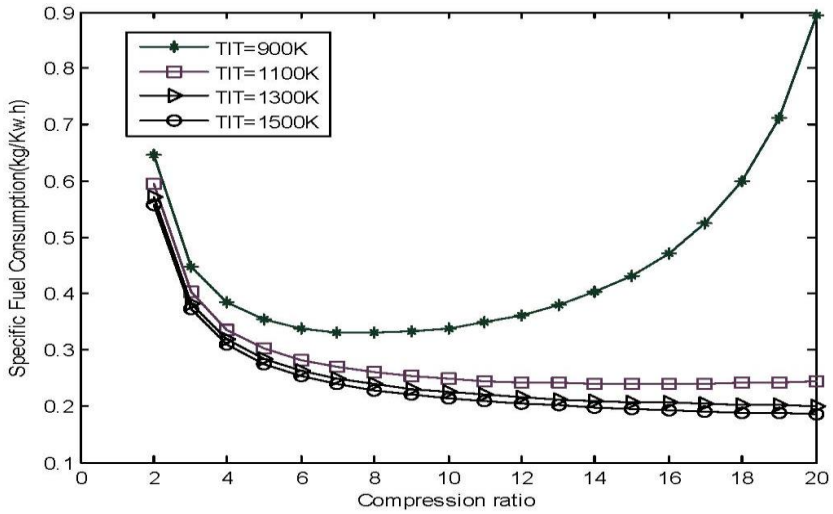

Figure 4. Effect of Compression ratio and TIT on Specific Fuel Consumption.

Figure 5 presents the variation in the maximum power output with compression ratio at different TITs. It is observed that at lower compression ratios the power output increases linearly with TIT. The peaks (maximum power output) of the curves vary with TIT such that at higher TITs, the peaks flatten out.

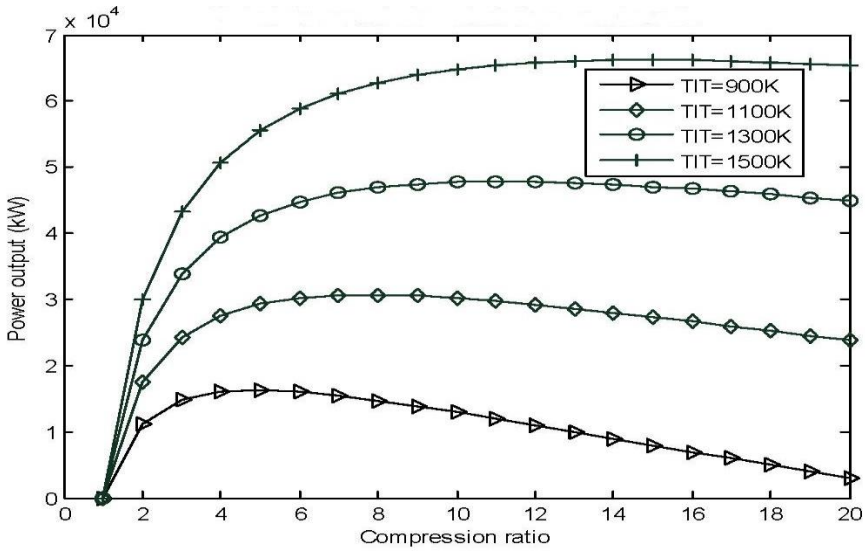

Figure 5. Effect of Compression ratio and TIT on Power Output.

Figure 6 shows the variation of net - work output with compression ratio and TIT. With increase in compression ratio, the net-work output decreases. At high turbine inlet temperature, the peaks of net - work output flatten out.

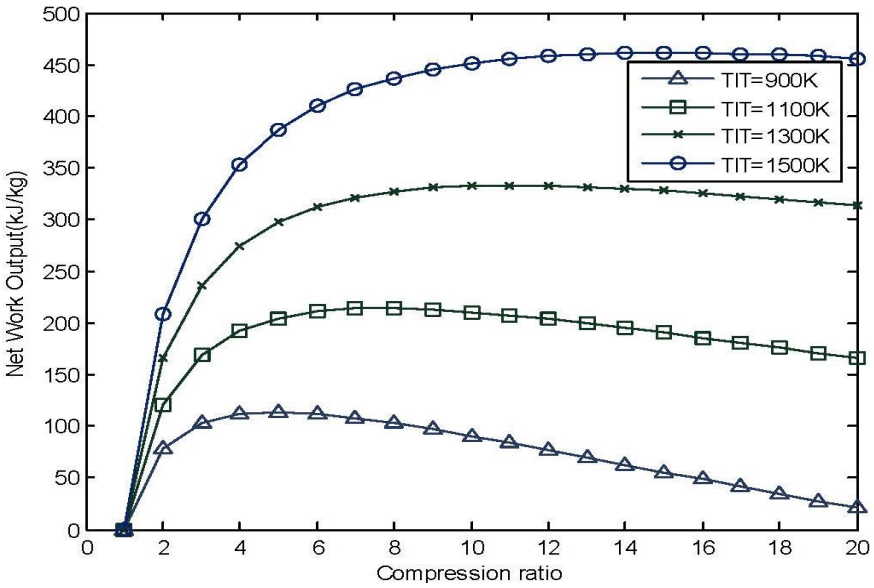

Figure 6. Effect of Compression ratio and TIT on Network Output.

Variation of heat supplied with compression ratio and TIT is presented in Figure 7. It is observed that heat supplied increases with TIT but decreases with compression ratio. Since compression ratio increases the temperature of the air entering the combustion chamber this implies that less heat is needed for combustion to take place in the combustion chamber. Therefore, increase in compression ratio for the same turbine inlet temperature decreases the heat supplied.

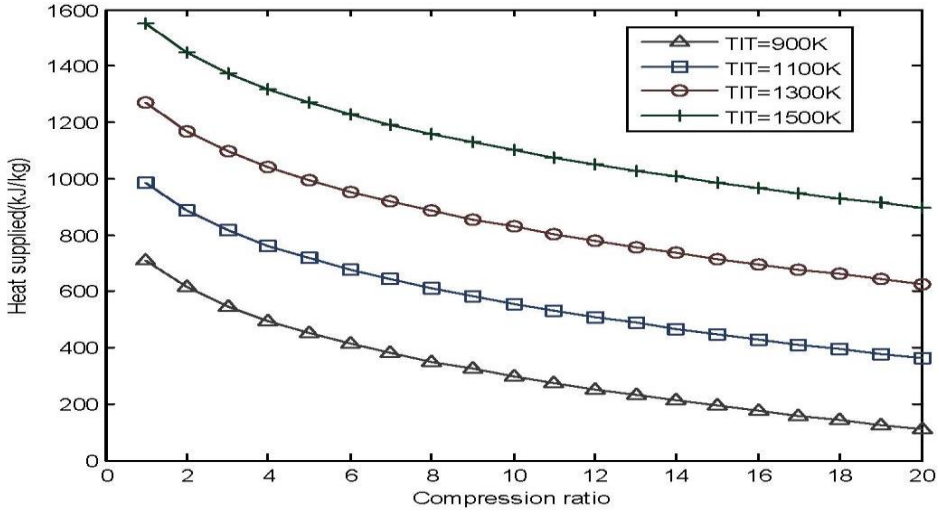

Figure 7. Effect of Compression ratio and TIT on Heat Supplied.

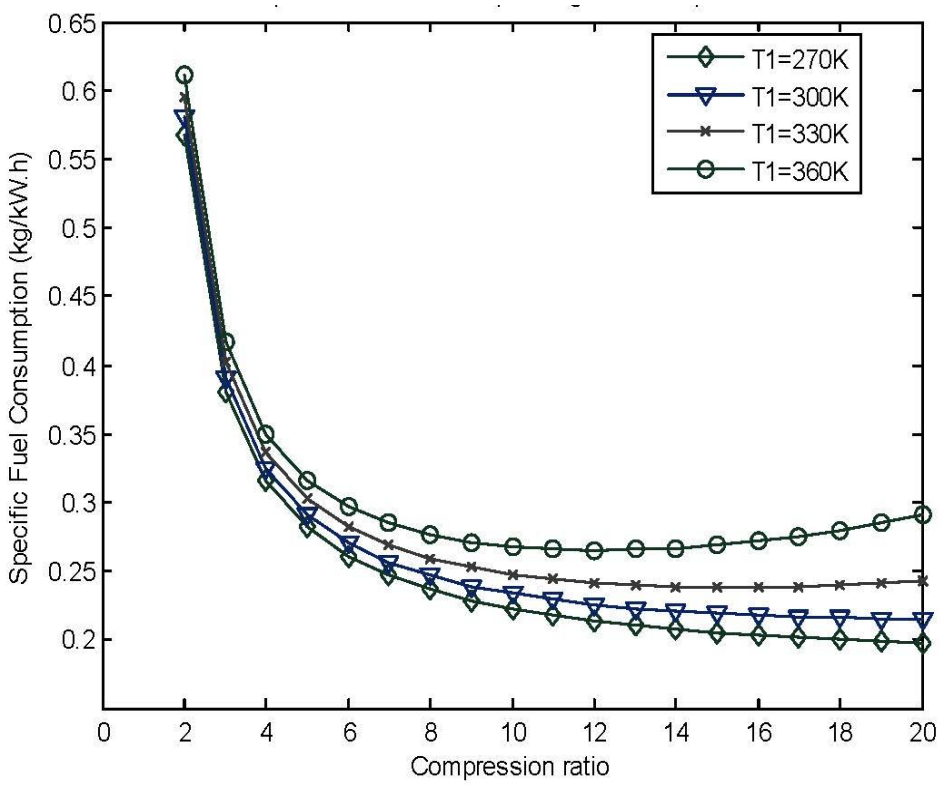

Figure 8. Effect of ambient temperature and compression ratio on specific fuel consumption.

Figure 8 shows the effect of compression ratio and ambient temperature on specific fuel consumption. It is observed that the specific fuel consumption increases with increased ambient temperature. At higher T1, the air density decreases, resulting in a decrease in air mass low rate. Thus, the fuel mass flow rate increases since the air to fuel ratio is kept constant. Therefore, the specific fuel consumption increases with the increase of ambient temperature due to the flue gas losses. The increase in compression ratio for gas turbine power plant leads to a continuous decrease of specific fuel consumption.

3.1.2. Effect of Ambient Temperature

Figure 9 shows the effect of ambient temperature and TIT on the power output. It is observed that at lower ambient temperature the power output increases linearly with TIT. The power output increases with turbine inlet temperature but decreases with increase in ambient temperature. As the ambient temperature increases, the specific work of the compressor increases (Nag, 2008), thus reducing the net- work output and invariably reducing the power output of the gas turbine. Also, increasing the TIT leads to an increase in the turbine work output, hence increase in the net power output.

Figure 10 shows variation of specific fuel consumption with ambient temperature and turbine inlet temperature. At lower ambient temperatures, the specific fuel consumption decreases linearly with ambient temperature. The specific fuel consumption increases with increasing ambient temperature and also with lower TIT. The effect of variation of SFC is more significant at higher ambient temperature and lower TIT.

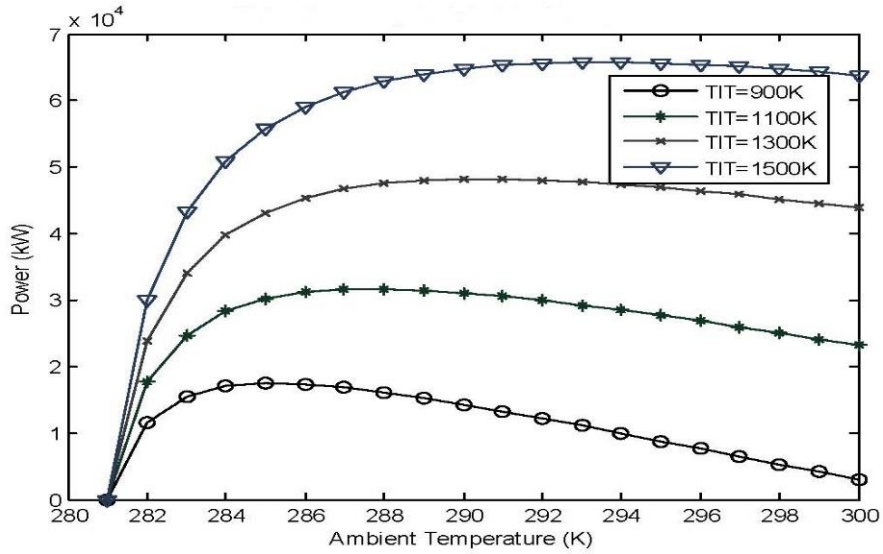

Figure 9. Effect of ambient temperature and TIT on Power Output.

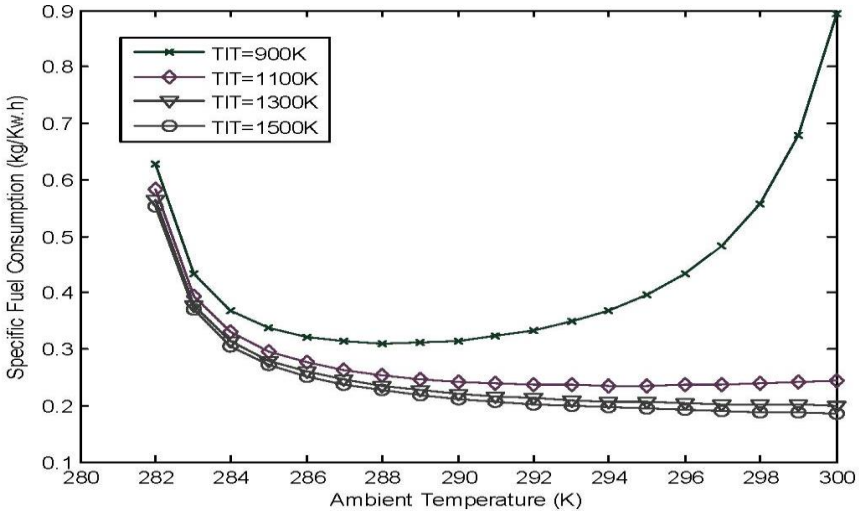

Figure 10. Effect of ambient temperature and turbine inlet temperature on specific fuel consumption.

3.1.3. Effect of Turbine inlet Temperature

The variation of power output with ambient temperature and TIT is shown in Figure 11. It can be seen that the power output increases linearly with TIT while it decreases with increase in the ambient temperature T1. The increase in power output due to turbine inlet temperature is as a result of the net - work output increase. Figure 11 also shows that the gas turbine power output is affected by ambient temperature due to the change of air density and compressor work. Since a lower ambient temperature leads to a higher air density and a lower compressor work that in turn gives a higher gas turbine output power. However, when the ambient temperature increases, the specific work of the compressor increases thus, reducing power output for gas turbine.

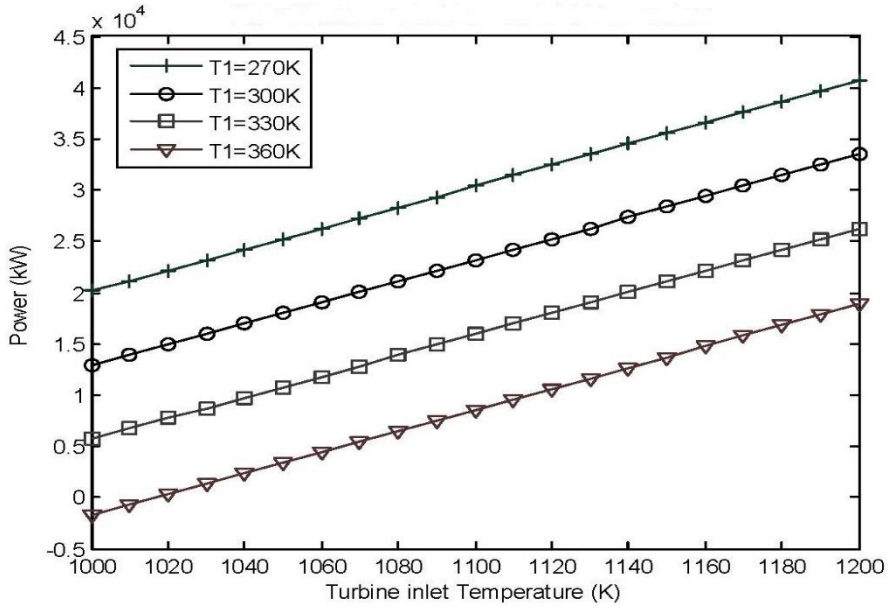

Figure 11. Effect of turbine inlet temperature and ambient temperature on power output.

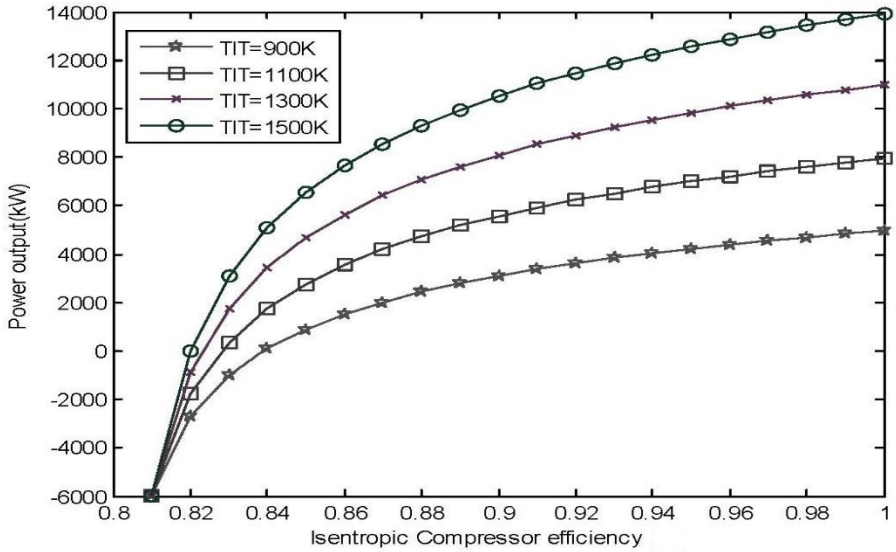

Figure 12. Effect of Turbine inlet temperature and Isentropic Compressor Efficiency on Power Output.

3.1.4. Effect of Compressor and Turbine Efficiencies

Figures 12 and 13 present the effect of the compressor and turbine isentropic efficiencies on the power output for various TITs. The power output increases with increase in the compressor and turbine isentropic efficiencies. This implies that the thermal losses have been reduced in compressor and turbine. This leads to increased power output. The rate of increase in power output is more significant at higher TIT and higher isentropic compressor and turbine efficiencies.

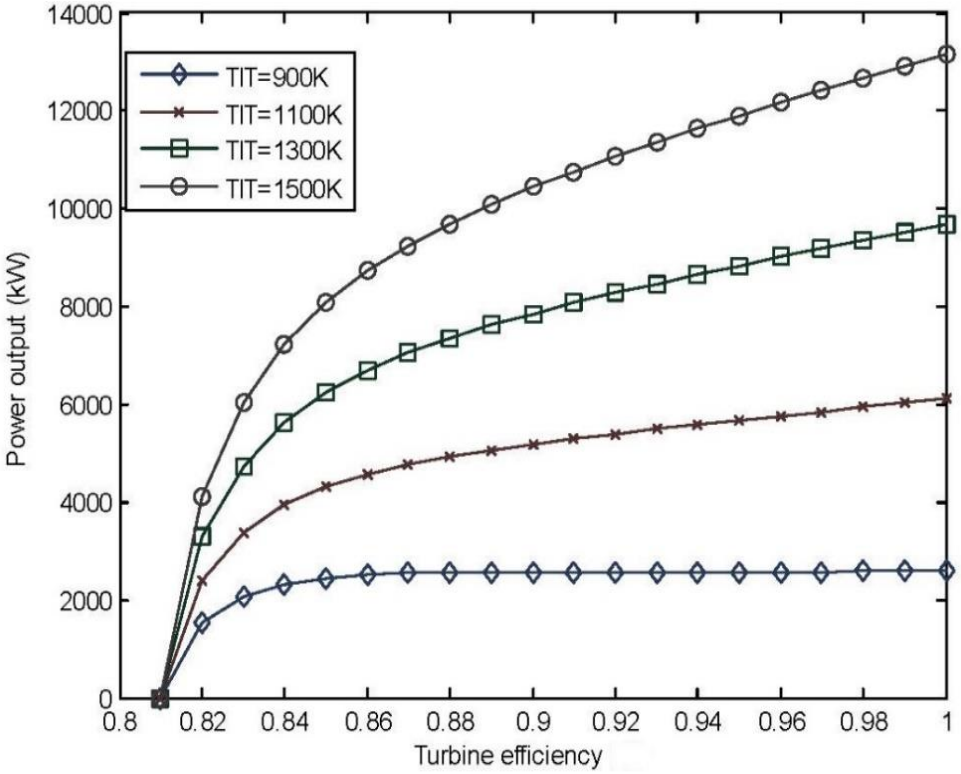

Figure 13. Effect of Turbine inlet temperature and Isentropic Turbine Efficiency on Power Output.

3.2. Results of Performance Evaluation of Selected Gas Turbine Power Plant Using Energy and Exergy Analyses

Energy and exergy analysis are important to explain how energy flows interact with each other and how the energy content of resources is exploited. The energetic efficiency (1st law efficiency) supplies information about the efficiency in using energy resources to get the products. The exergetic efficiency (2nd law efficiency) is used to explain efficiency from the exergetic point of view. These two indicators (1st law efficiency and 2nd law efficiency) have wide range of application at system and component level (Mirandola et al., 2000, Oyedepo, 2014). A complete analysis of the thermodynamic performance of a process generally requires the use of both energy and exergy analyses.

3.2.1. Energy Analysis

The average operating data of the selected gas turbine power plant for the period of six years (2005 to 2010) is presented in Table 1. The energy loss experienced in the gas turbine components are shown in Table 3. The average operating data for period of six years (2005 to 2010) presented in Table 1 were used as inputs to the analytical models (Eqns. 9, 11, 24, 25 and 26). For the period of six years, the thermal efficiency is 36.68% (see Table 3). Energy performance analysis also shows that the turbine has the highest proportion of energy loss (31.98%) in the selected plant.

Table 3. Results of Energy Performance Analysis

S/No	Energy Performance Indicator	Unit	Value
1	Installed rated power	MW	33.5
2	Energy loss of compressor	MW	0.71
3	Energy loss of combustion chamber	MW	11.35
4	Energy loss of turbine	MW	103.78
5	Total energy loss in the plant	MW	415.84
6	Network of turbine	MW	44.99
7	% Energy loss of compressor	%	1.50
8	% Energy loss of combustion chamber	%	5.48
9	% Energy loss of turbine	%	31.98
10	% Total energy loss in the plant	%	38.96
11	Energy input	MW	207.20
12	Thermal efficiency	%	36.68

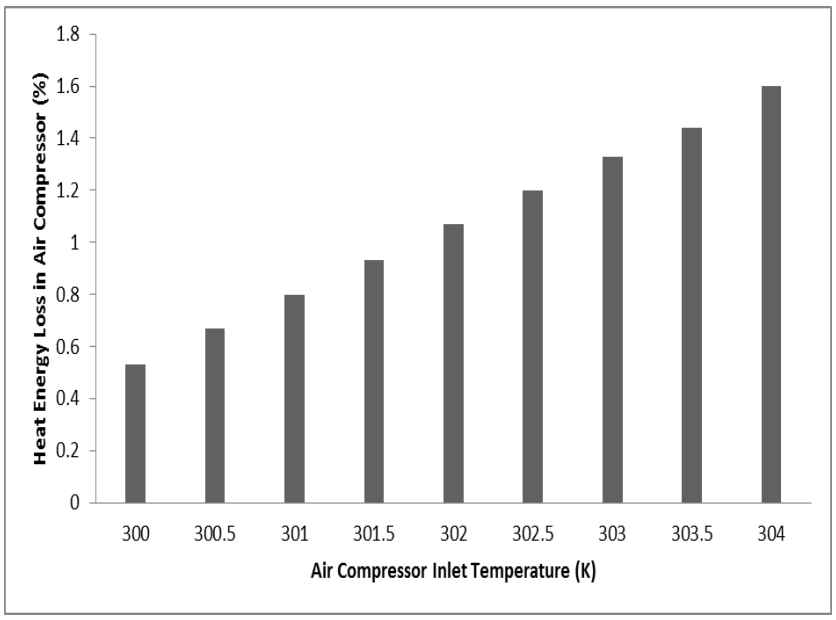

Figure 14. Heat Energy Loss in Air Compressor (%) Against Air Compressor Inlet Temperature (K).

As earlier mentioned, the operating parameters have significant effect on gas turbine engine performance. Figure 14 presents the effect of variation in air compressor inlet temperature on heat energy loss in the air compressor. The energy losses in air compressor increase at high ambient temperature. The air compressor work increases as inlet air temperature increases which leads to a decrease in net work of the gas turbine. Air

compressor work can be minimized when the air inlet temperature and mass flow rate are reduced. This shows that compressor work can be managed by the compressor inlet air temperature.

Compression ratio is another parameter that affects performance of gas turbine power plant. Figure 15 shows the effect of compression ratio on energy loss in air compressor. Increase in pressure ratio brings about decrease in energy loss in air compressor. This shows that the compressor work can be reduced by decreasing the compression ratio.

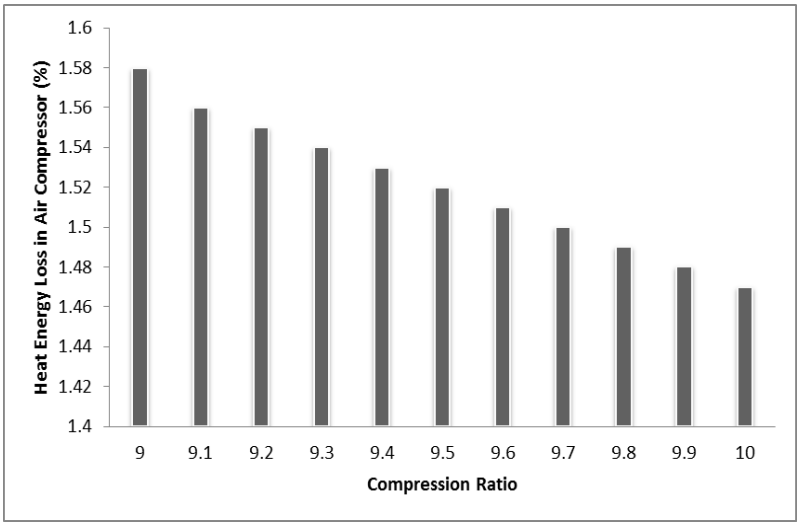

Figure 15. Heat Energy Loss in Air Compressor (%) Against Compression Ratio.

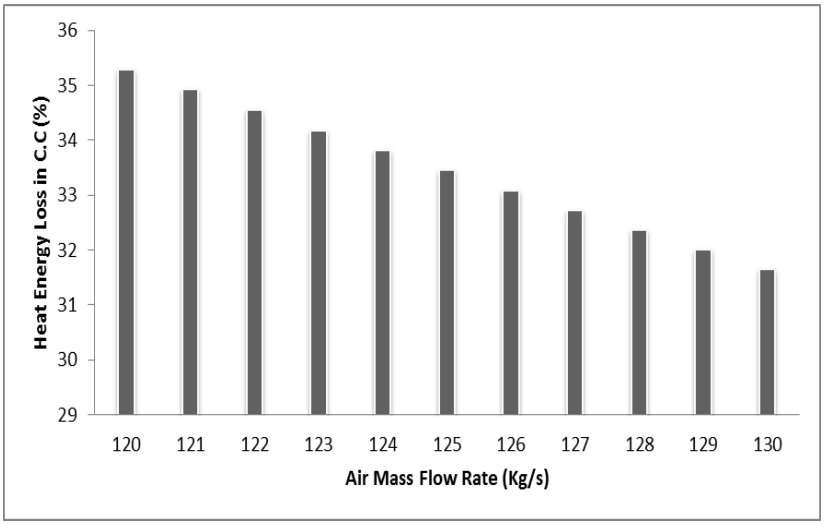

Figure 16. Heat Energy Loss in Combustion Chamber (%) Against Air Mass Flow Rate (kg/s).

Figure 16 shows the effect of air mass flow rate on heat energy loss in the combustion chamber. Heat energy losses in the combustion chamber decrease with increase in air mass flow rate. This implies that high mass flow rate of air can minimize the energy losses in

combustion chamber as this would introduce more air for combustion. The unburnt air in combustion chamber serves as coolant. Therefore, the energy losses decrease as the temperature of the hot gases is decreased. This is due to high quantity of air mass flow which lowers the temperature of the hot gases.

From above discussion, it is obvious that gas turbine performance is affected by operating parameters. The magnitude of effects of these parameters on performance of gas turbine varies from power plant to power plant based on either technical deficiencies within the system or changes in ambient conditions. Arriving at a decision for plant performance improvement based on energetic performance results only may not be sufficient. For complex systems like gas turbine plant with multiple components this may be misleading as quantifying actual losses in the different system control volumes might not be accurately achieved. Using only energetic analysis for decision making is lopsided, since it does not reveal explicit presentation of plant performance. Therefore, the results obtained from energetic performance analysis should be considered with those of exergetic analysis allowing an improved understanding by quantifying the effect of irreversibility occurring in the plant and the locations.

Table 4. Results of Exergy Analysis

S/No	Energy Performance Indicator	Unit	Value
1	Fuel exergy flow rate	MW	220.53
2	Exergy destruction rate in compressor	MW	4.58
3	Exergy destruction rate in combustion chamber	MW	55.20
4	Exergy destruction rate in turbine	MW	1.07
5	Total exergy destruction rate in the plant	MW	60.85
6	% Exergy destruction rate in compressor	%	7.52
7	% Exergy destruction rate in combustion chamber	%	90.71
8	% Exergy destruction rate in turbine	%	1.76
9	Exergy efficiency of compressor	%	85.99
10	Exergy efficiency of combustion chamber	%	74.97
11	Exergy efficiency of turbine	%	98.56
12	Overall exergetic efficiency	%	19.06
13	Exergetic performance coefficient	%	1.45
14	Efficiency defect of compressor	%	14.01
15	Efficiency defect of combustion chamber	%	25.03
16	Efficiency defect of turbine	%	1.42
17	Total efficiency defect of the plant		40.46

3.2.2. Exergy Analysis

The exergy flow rates at the inlet and outlet of each component of the plants were evaluated based on the values of measured properties such as pressure, temperature, and mass flow rates at various states. These quantities are used as input data to the computer program (MATLAB) written to perform the simulation of the performance of the components of the gas turbine power plant and the overall plant.

An exergy balance for the components of the gas turbine plant and of the overall plant was performed and the net exergy flow rates crossing the boundary of each component of the plant, together with the exergy destruction, exergy defect and exergy efficiency in each component are calculated and are presented in Table 4. The exergy analysis result shows that the highest percentage exergy destruction occurs in the combustion chamber (CC) (90.71%) and followed by the air compressor (7.52%). Hence, the combustion chamber is the major source of thermodynamic inefficiency in the plant considered and this is due to the irreversibility associated with combustion and the large temperature difference between the air entering the combustion chamber and the flame temperature. These immense losses basically mean that a large amount of energy present in the fuel, with great capacity to generate useful work, is being wasted.

By comparing data in Table 3 and Table 4, the total plant losses for the plant is 38.96% for energetic consideration and 40.46 % for exergetic case. This shows that using only energetic analysis for decision making is lopsided as it does not reveal explicit presentation of plant performance. Therefore, the result obtained from energy analysis should be considered along with those from exergy analysis. This allows an improved understanding by quantifying the effect of irreversibility occurring in the plant and the locations of occurrence.

The effect of turbine inlet temperature on the exergetic efficiency (or second law efficiency) of the plant was investigated. The simulation of the performance of the plant was done by varying the turbine inlet temperature from 1000 to 1400 *K*. Figure 17 shows that the second-law efficiency of the plant increases steadily as the turbine inlet temperature increases. The increase in exergetic efficiency with increase in turbine inlet temperature i s limited by turbine material temperature limit. This can be seen from the plant efficiency defect curve. As the turbine inlet temperature increases, the plant efficiency defect decreases to minimum value at certain TIT (1200K), after which it increases with TIT. This shows degradation in performance of gas turbine plant at high turbine inlet temperature.

The effect of variation in ambient temperature on the second law efficiency (exergetic efficiency) of the gas turbine components was also assessed. The simulation of the performance of the components was done by varying the air inlet temperature from 290 to 320 K. Figure 18 compares the second-law efficiencies of the air compressor, combustion chamber, gas turbine and the overall plant when the ambient temperature varies. The exergy efficiency of the turbine component and the overall exergetic efficiency of the overall plant decreased with increased ambient temperature, whereas the exergy efficiencies of the compressor and combustion chamber increased with increased ambient temperature. The overall exergetic efficiency decreased from 18.53 to 17.26% for ambient temperature range of 290 to 320 K. It was found that a 5 K rise in ambient temperature resulted in a 1.03% decrease in the overall exergetic efficiency of the plant. The reason for the low overall exergetic efficiency is due to large exergy destruction in the combustion chamber (Kotas, 1995).

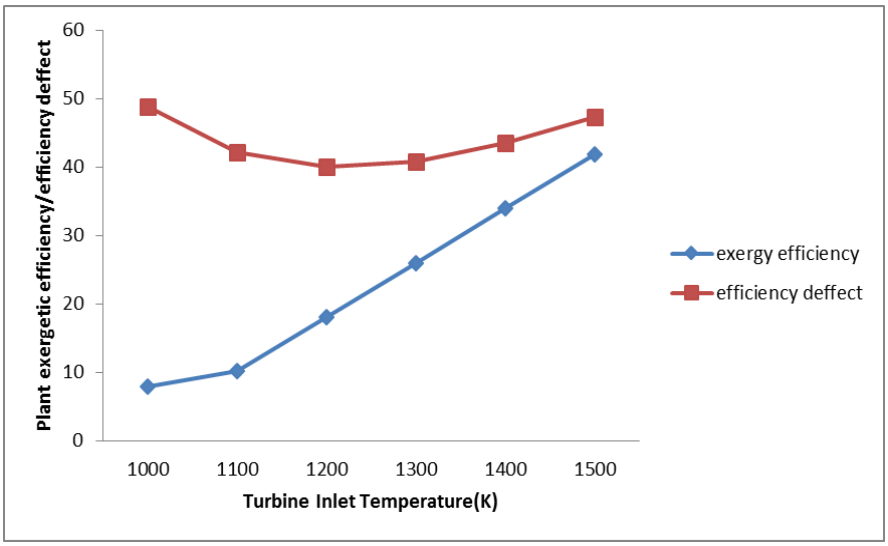

Figure 17. Variation in Plant Exergetic Efficiency and Efficiency defect with Turbine Inlet Temperature.

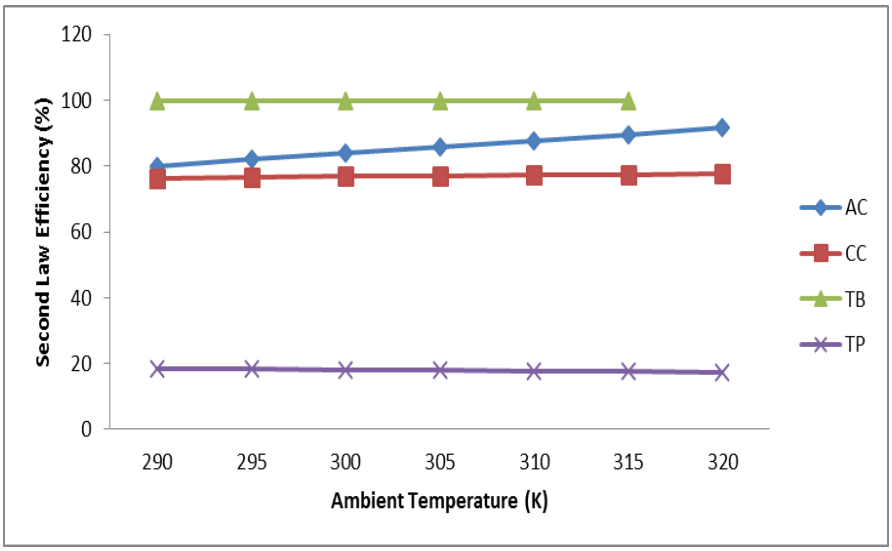

Figure 18. Variation in second-law efficiency with Ambient Temperature. AC - Second law efficiency of Compressor, CC- Second law efficiency of combustion chamber, TB – Second law efficiency of turbine, TP – Second law efficiency of entire plant.

3.3. Validation of Computer Simulation Code

The model developed in this study is validated by the actual data that were taken from the existing gas turbine power plant in Nigeria. Average parameters recorded within the period under review are set as base line for comparison with the calculated results. The parameters considered in this study in gas turbine engine during simulation are inlet temperature of the air compressor, the mass flow rate of fuel and turbine inlet temperature.

The results of thermodynamic properties of the cycle form the modeling part and the power plant data are illustrated in Table 5. The comparison of simulation results and the actual data from the power plant show that the difference in the simulation results and the actual data varies from 1.17 to 5.04 %. The maximum difference is about 5.04 % for mass flow rate of fuel while the minimum difference is about 1.17 % for compressor outlet temperature. This validates the correct performance of the developed simulation code to model the selected gas turbine power plant, as the results of the simulation values are close to the actual operating data of the plant considered in this study.

Table 5. Results between the power plant data and simulation code

Parameter	Unit	Measured data	Simulation code	Difference (%)
T_2	K	622.31	629.59	1.17
T_4	K	750	775.02	3.34
m_f	Kg/s	2.58	2.45	5.04

CONCLUSION AND RECOMMENDATIONS

In this study, comprehensive thermodynamic modelling, energy and exergy analyses were performed for selected gas turbine power plant in Nigeria. To achieve this aim, a simulation code was developed in MATLAB software program. In order to validate the simulation code, the results were compared with the actual data obtained from running selected gas turbine power plant in Nigeria. The results showed a reasonably good agreement between simulation code results and experimental data obtained from actual running gas turbine plants.

The thermodynamic model reveals that the influence of operating parameters including the compression ratio, turbine inlet temperature and ambient temperature has significant effect on the performance of gas turbine power plant. The thermodynamic simulation results are summarized as follows.

- The thermal efficiency and power output decrease linearly with increase of ambient temperature.
- The thermal efficiency and power output increase linearly at lower compression ratio with increase in turbine inlet temperature.
- Heat supplied increases with turbine inlet temperature but decreases with compression ratio.
- Specific fuel consumption increases with increase ambient temperature but decreases with increase compression ratio and turbine inlet temperature.
- The turbine inlet temperature (TIT) significantly affects the performance of gas turbine engine. It should be kept on higher side for minimizing losses in the gas turbine system. Increasing the turbine inlet temperature increases the output power and thermal efficiency as a result of increasing the turbine work.

Energy analysis reveals that thermal efficiency of the selected power plant is 36.68 %. Also, energy performance analysis shows that the turbine has the highest proportion of energy loss (31.98%) in the plant investigated. This is followed by the combustion chamber (5.48%). Results of energy analysis further show that heat energy loss in air compressor increases with air compressor inlet temperature but decreases with compression ratio. In combustion chamber, heat energy loss decreases with increase in air mass flow rate.

The results from the exergy analysis show that the combustion chamber is the most significant exergy destructor in the selected power plant, which is due to the chemical reaction and the large temperature differences between the fluids in different section of the combustion chamber. These immense losses basically mean that a large amount of energy present in the fuel, with great capacity to generate useful work, is being wasted. Moreover, the results show that an increase in the turbine inlet temperature (TIT) leads to an increase in gas turbine exergy efficiency due to a rise in the output power of the turbine and a decrease in the combustion chamber losses. The total efficiency defects and overall exergetic efficiency of the selected power plant are 40.46 % and 19.06 % respectively.

Though, gas turbine engines have the advantage of fast startup, but suffer from low power output and thermal efficiency at high ambient temperatures. GT power plants operating in Nigeria are simple GTs, there is a tremendous derating factor due to higher ambient temperatures. No wonder the thermal efficiency of the selected gas turbine power plant is low. Based on the results of this research work, the following possible economical methods and technologies to improve performance of the selected gas turbine power plants are hereby recommended:

Use of Spar – Shell Blade for Turbine Blades

From theoretical analysis of gas turbine engine, the maximum cycle temperature (TIT) is limited by metallurgical considerations. The use of Spar-Shell technology allows the turbine operator to achieve significantly greater temperature increases because the turbine blade material is not the nickel alloy that is currently used, but a higher temperature metal of the refractory type. Refractory materials are a class of metals that are extraordinarily resistant to heat and wear. Examples of refractory metals include Niobium, Molybdenum, Tungsten, and Tantalum. The Spar-Shell Blade allows the metal temperature of the turbine blade to increase by 100°C, saving 50 - 75% of the air required to keep the blades “cool.” These changes allow for the overall turbine to operate 3.5% more efficiently (FTT, 2009).

Advanced Clearance Control Schemes and Sealing Technologies

Gas turbines are constructed with cases around the blades to contain and control the working fluid. Every molecule of working fluid that the blade does not extract work from as it passes by, is called “leakage” which also reduces turbine efficiency. The problem of leakage is common in the simple gas turbine plants in Nigeria. The possible method to control and limit the amount of leakages in turbines can be through advanced clearance control schemes and sealing technologies.

Improvements in the Surface Finish of Turbine Blades and Cases

Improvements in the surface finish of blades and cases help to minimize losses in turbine efficiency and invariably the performance of gas turbine power plant. Surface finish improvement can be accomplished through better blade coatings, improved wear resistance, and other surface treatments.

Retrofitting with Advanced Cycle

Retrofitting the selected GT power plant with advanced cycle would improve its performance significantly. Among many proven technologies are inlet air cooling, intercooling, regeneration, reheating and steam injection gas turbine (STIG) etc. Air inlet cooling system (evaporative cooling, inlet fogging or inlet chilling method) is a useful option for increasing power output of the selected power plant. This helps to increase the density of the inlet air to the compressor. As AES Barge gas turbine plant is very close to lagoon area, the source of cooling water can be obtained from lagoon. The inlet air cooling system is cost effective and can be implemented in the basic system without major modification to the original system integration.

Application of Coatings to Gas Turbine Compressor Blades

The compressor airfoils of older turbines tend to be rougher than a newer model simply because of longer exposure to the environment. In addition, the compressor of older models consumes a larger fraction of the power produced by the turbine section. Therefore, improving the performance of the compressor will have a proportionately greater impact on total engine performance. Application of Coatings to gas turbine compressor blades (the “cold end” of the machine) would improve the selected gas turbine engine performance. Compressor blade coatings provide smoother, more aerodynamic surfaces, which increase compressor efficiency. In addition, smoother surfaces tend to resist fouling because there are fewer “nooks and crannies” where dirt particles can attach. Coatings are designed to resist corrosion, which can be a significant source of performance degradation, particularly if a turbine is located near saltwater. As AES Barge gas turbine plant is located on lagoon, compressor coating technology would improve the plant performance significantly.

ACKNOWLEDGMENTS

The authors appreciate the Management of AES power plant for providing the data used in this study.

REFERENCES

Abam, F. I., Ugot, I. U. and Igbong, D. I. (2011), ‘Thermodynamic Assessment of Grid – Based Gas Turbine Power Plants in Nigeria’, *Journal of Emerging Trends in Engineering and Applied Sciences* 2(6):1026 – 1033.

Adrian, I. and Dorin, L. (2010), Thermodynamic Analysis of Gas Turbine Powered Cogeneration Systems,' *Journal of Scientific and Ind. Res*., Vol. 69: 548 – 553.

Alcides, C. N. (1999), 'Assessment of Novel Power Generation Systems for Biomass Industry,' PhD Thesis, Cranfield University.

Ameri M, Ahmadi P and Khanmohammadi S (2007), 'Exergy analysis of a 420MW combined cycle power plant', *Int. J. Energy Res*. DOI: 10.1002/er.1351.

Barzegar - Avval, H, Ahmadi, P, Ghaffarizadeh, A. R. and Saidi, M. H. (2011), ''Thermo-economic-environmental multi-objective optimization of a gas turbine power plant with preheater using evolutionary algorithm', *Int. J. Energy Res.* 35:389–403.

Bassily, A. M (2001), 'Effects of evaporative inlet and after cooling on the recuperated gas turbine cycle', *Applied Thermal Engineering* 21 (18): 1875–1890.

Bejan, A; Tsatsaronis, G. and Moran, M. (1996), Thermal Design and Optimization, John Wiley & Sons, Inc. New York.

Bilgen, E (2000), 'Exergetic and engineering analyses of gas turbine based cogeneration systems', *Energy* 25: 1215–1229.

Chaker, M; Meher-Homji, C. B ; Mee, T and Nicholson, A (2003) ' Inlet fogging of gas turbine engines detailed climatic analysis of gas turbine evaporation cooling potential in the USA', *Journal of Engineering for Gas Turbine and Power* 125 :300–309.

Chen, Q, Han, W, Zheng, J, Sui, J and Jin, H (2014), 'The exergy and energy level analysis of a combined cooling, heating and power system driven by a small scale gas turbine at off design condition', *Applied Thermal Engineering* 66:590 – 602.

Eastop, T. D and McConkey, A (2009), Applied Thermodynamic for Engineering Technologists (5th Ed.), Pearson Educational Ltd., India, pp. 290.

Ebadi, M. J and Gorji – Bandpy (2005), 'Exergetic Analysis of Gas Turbine Plants,' *Int. J* Exergy**,** 2: 31 – 39.

Erdem H. A. and Sevilgen H. S. (2006), 'Case study: effect of ambient temperature on the electricity production and fuel consumption of simple cycle gas turbine,' applied *thermal engineering* 26: 230-236.

FTT (Florida Turbine Technologies. Inc.) (2009), *Turbine Efficiency Improvements for Existing Power Plants*, pp 1 – 8.

Gareta, R; Romeo, L. M; Gil, A. (2005). Economic optimization of gas turbine inlet air—cooling systems in combined cycle applications, *Center for Power Plant Efficiency Research',* ECOS, Vol. 1: 409–415.

Ghazikhani, M, Khazaee I and Abdekhodaie, E (2014), 'Exergy analysis of gas turbine with air bottoming cycle', *Energy* 72: 599 – 607.

Guinee J. B (2001), (Ed.), Life cycle assessment: an operational guide to the ISO standards; LCA in perspective; guide; operational annex to guide. The Netherlands: Centre for Environmental Science, Leiden University.

Hall A. D; Stover, J. C and Breisch R. B (1994), 'Gas turbine inlet-air chilling at a cogeneration facility', *ASHRE Transactions* 100, Part 1.

Jaber, Q. M; Jaber, J. O. and Khawaldah, M. A. (2007), "Assessment of Power Augmentation from Gas Turbine Power Plants Using Different Inlet Air Cooling Systems", *Jordan Journal of Mechanical and Industrial Engineering,* Vol 1,No 1, pp7-15.

Kakaras E., Doukelis A. and Karellas S. (2004), 'Compressor intake-air cooling in gas turbine plants', *Energy*, 29: 2347-2358.

Kamal N. A. and Zuhair A. M. (2006), 'Enhancing Gas Turbine Output Through Inlet Air Cooling', *Sudan Engineering Society Journal*, 52(46), pp 1-8.

Khaldi, F and Adouane, B (2011), 'Energy and exergy analysis of a gas turbine power plant in Algeria,' *Int. J. Exergy, 399 – 413.*

Kotas, T. J. (1995), 'The exergy method of thermal plant analysis,' Krieger Publishing Company, Malabar, Florida.

Kreith F and Goswanni, Y. D. (Ed.) (2005), The CRC Handbook of Mechanical Engineering (2nd Edition), CRC Press.

Kurt, H., Recebli, Z and Gredik, E (2009), 'Performance analysis of open cycle gas turbines', *International Journal of Energy Research*, 33(2):285–294.

Lior, N. and Zhang, N. (2007), 'Energy, exergy, and Second Law performance criteria, *Energy* 32: 281–296.

Mahmoudi S. M, Zare V., Ranjbar F. and Farshi L. (2009), 'Energy and exergy analysis of simple and regenerative gas turbines inlet air cooling using absorption refrigeration', *J. Appl. Sci.*, 9(13): 2399-2407.

Mirandola, A; Stoppato, A and Tonon, S. (2000), "An Integrated Approaches to the Assessment of Energy Conversion Plants, *Int. J. Applied Thermodynamics* Vol. 3 (3): 111 – 119.

Mozafari, A., Ahmadi, A. and Ehyaei, M. A. (2010) 'Optimisation of micro gas turbine by exergy, economic and environmental (3E) analysis', *Int. J. Exergy*, Vol. 7, No. 1, pp.1–19.

Nag, P. K. (2008), Power plant engineering, New Delhi: Tata McGraw-Hill Publishing Company Limited.

Oh, S. D., Pang, H., Kim, S. and Kwak, H. (1996), 'Exergy analysis for a gas-turbine cogeneration system', *Journal of Engineering for Gas Turbine and Power*, Vol. 118, pp.782 - 791.

Oyedepo, S. O. (2014), Thermodynamic Performance Analysis of Selected Gas Turbine Power Plants in Nigeria, Ph.D Thesis, Covenant University, Ota, Nigeria.

Oyedepo, S. O. and Kilanko, O. (2014), 'Thermodynamic Analysis of a Gas Turbine Power Plant Modeled with an Evaporative Cooler', *International Journal of Thermodynamics* (IJoT) Vol. 17 (No. 1), pp. 14 – 20.

Rajput, R. K. (2007), Engineering Thermodynamics (S.I Units), LAXMI PUBLICATIONS (P) LTD 113, Golden House, Daryaganj, New Delhi-110002.

Ray, T.K, Ganguly, R and Gupta, A (2007), 'Exergy Analysis for Performance Optimization of a Steam Turbine Cycle', IEEE PES Power Africa 2007 Conference and Exposition Johannesburg, South Africa, 16-20 July 2007, pp 1 – 8.

Reddy, V. S., Kaushik, S. C. and Panwar, N. L. (2013), 'Review on power generation scenario of India, *Renewable and Sustainable Energy Reviews* 18: 43–48.

Sadrameli S. M. and Goswami D. Y. (2007), 'Optimum operating conditions for a combined power and cooling thermodynamic cycle', *Appl. Ener.*, 84: 254-265.

Tahouni N., Jabbari B. and Panjeshahi M. H., (2012), 'Optimal design of a cogeneration system in a kraft process using genetic algorithm', *Chemical Engineering Transactions*, 29, 19-24.