

Thermodynamics Modelling and Performance Assessment of Gas Turbine Power Plant

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Abstract

In this study, thermodynamics modelling of gas turbine engine is made based on thermodynamic association. 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 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 evaluation shows that the turbine has the maximum 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 rise in the gas turbine inlet temperature (GTIT) cause fall in 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: ambient temperature, energy analysis, exergy analysis, gas turbine engine, simulation

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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. ^[1] 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 has made gas turbine power plants of scientific interest for the efficient utilization of energy resources. ^[2]

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 module 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.^[3]

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 variation in performance, price, fractional load modelling, as well as different performance variations with the ambient temperature.^[4]

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 meeting energy demand peaks.^[5] 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.^[6]

In gas turbines, since the air for combustion is received directly from the environment, their performance is strongly exaggerated by both external factors (ambient temperature and relative humidity) and core 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 expanse of these variations greatly affects electricity production, fuel consumption and plant incomes.^[7] 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.^[8,9] One way of restoring operating conditions is to add an air cooler at the compressor inlet.^[10]

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.^[11, 12] 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),^[13] 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.^[14] 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.^[15, 16]

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.^[17] 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.^[18–21]

The prime objectives of this study are:

- (i) to simulate gas turbine engine using MATLAB R2010a
- (ii) to investigate the effect of variation of operation conditions on performance of gas turbine engine
- (iii) to model and evaluate performance of gas turbine power plant using first and second laws of thermodynamics
- (iv) to investigate the effect of variation of ambient temperature and TIT on energy loss and exergy destruction of the selected gas turbine power plant

MATERIALS AND METHODS

Data Collection

In this work, AES barges gas turbine plant unit PB204 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 analysed 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 modelling. Mass, energy conservation laws and exergy balance were applied to each component and the performance of the plant was determined for the system.

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 network output was calculated.

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

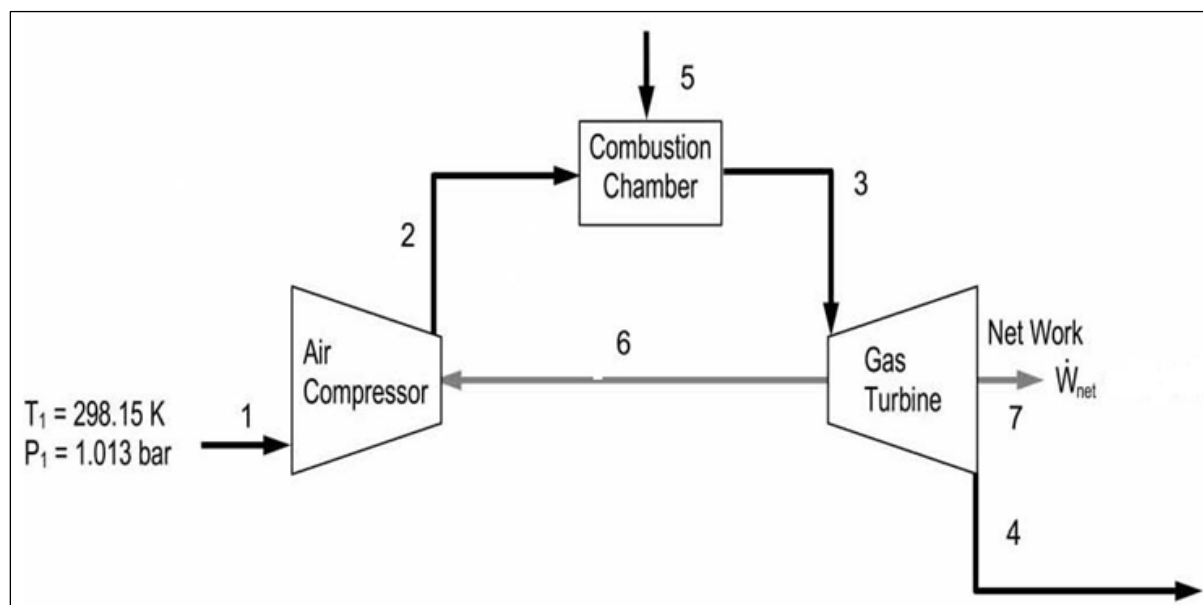


Fig. 1. A Schematic Diagram for a Simple GT Cycle.

Power Plant Component Energetic and Exergetic Analyses

The energetic and exergetic efficiencies of the whole unit that makes up the particular 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.

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 (1–40) depict the leading thermodynamics models for simple gas turbine engine. In this study, energy and exergy models are

well-thought-out for gas turbine performance assessment.

Energy (First Law of Thermodynamics) Model

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 ^[22]:

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

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

Air Compressor

The compression ratio (r_p) can be defined as:

$$r_p = \frac{p_2}{p_1}, \quad (2)$$

Where, p_1 and p_2 are the compressor inlet and outlet air pressure, respectively. The isentropic efficiency for the compressor is expressed as:

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

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:

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

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:

$$W_c = \frac{\dot{m}_a c_{pa} T_1}{\eta_c} \left((r_p)^{\frac{\gamma_a-1}{\gamma_a}} - 1 \right), \quad (5)$$

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 $200\text{K} < T < 800\text{K}$ (Rahman *et al.*, 2011)^[23]:

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

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

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

Where, T_a is the ambient temperature. Energy input to the air compressor at specific inlet temperature (T_1) is given as:

$$\dot{Q}_{c2} = \dot{m}_a (c_{pa} T_2 - c_{pa} \bar{T}_1) \quad (8)$$

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

$$\dot{Q}_{c\text{ loss}} = \dot{m}_a (c_{pa} T_2 - c_{pa} T_a) - \dot{m}_a (c_{pa} T_2 - c_{pa} \bar{T}_1) \quad (9)$$

Where, $\bar{T}_1 = \frac{T_a + T_2}{2}$.

Combustion Chamber

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

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

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

$$\dot{Q}_{CC\text{ loss}} = \dot{m}_a c_{pa} T_2 + \dot{m}_f LHV - \dot{m}_f c_{pg} T_3 \quad (11)$$

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 $1000\text{ K} < T < 1500\text{ K}$ ^[23,24]:

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

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

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

Where,

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

The pressure drop across the combustion chamber (ΔP_{cc}) is usually around 2%.^[22, 25] The Turbine inlet pressure (P_3) can be calculated as:

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

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.

Gas Turbine

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

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

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:

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

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

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

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

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

Where,

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad (20)$$

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:

$$P = \dot{m}_g \left[c_{pg} T_3 \eta_T \left(1 - (r_p)^{\frac{1-\gamma_g}{\gamma_g}} \right) - \frac{c_{pa} T_1}{\eta_c} \left((r_p)^{\frac{\gamma_a-1}{\gamma_a}} - 1 \right) \right] \quad (21)$$

Energy input in the turbine is given as:

$$\dot{Q}_T = \dot{m}_g c_{pg} T_3 \quad (22)$$

Energy utilized for turbine work is calculated as follows:

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

Where, T_3 is the combustion chamber exit temperature and TET is turbine exhaust temperature. Energy loss from the turbine is given as:

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

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

$$\dot{Q}_{TTloss} = \dot{Q}_{Closs} + \dot{Q}_{CCloss} + \dot{Q}_{Tloss} \quad (25)$$

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

$$\eta_{th} = \frac{\dot{W}_n}{\dot{m}_f LHV} \quad (26)$$

Equation (25) is also known as the first law efficiency of gas turbine. [26] The specific fuel consumption (SFC) is determined by:

$$SFC = \frac{3600}{\dot{W}_n} f, \quad (27)$$

Where, f (fuel – air mass ratio) 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:

$$HR = \frac{3600}{\eta_{th}} \quad (28)$$

Exergy (Second Law of Thermodynamics) Model

The second law of thermodynamics complements and enhances the examination of energy system by enabling calculation of the real thermodynamic inadequacies 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. [27]

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. [28, 29] The thermo-mechanical exergy stream may be decomposed into its thermal and mechanical components so that the balance in rate form gives:

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

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 [29, 30]:

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

And

Air Compressor

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

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

Combustion Chamber

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

$$\begin{aligned} \dot{E}_{DC} = T_0 \left[\dot{S}_3 - \dot{S}_2 + \dot{S}_5 + \frac{\dot{Q}_{CC}}{T_0} \right] \\ = \dot{m}T_0 \left\{ \left(c_{p2-3} \ln \left(\frac{T_3}{T_2} \right) - R \ln \left(\frac{P_3}{P_2} \right) \right) + \right. \\ \left. \left(c_{p5} \ln \left(\frac{T_5}{T_0} \right) - R \ln \left(\frac{P_5}{P_0} \right) \right) + \frac{c_{p2-3}(T_3 - T_2)}{T_{in CC}} \right\} \end{aligned} \quad (35b)$$

Gas Turbine

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

$$\dot{E}^P = \dot{m}RT_0 \ln \frac{P}{P_0}, \quad (31)$$

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 [31, 32]:

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

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:

$$E_i^{tot} - E_e^{tot} - E_D = 0 \quad (33)$$

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:

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

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

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

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 i^{th} component efficiency defect denoted by δ_i is given by Equation (38)^[30]:

$$\delta_i = \frac{\sum \Delta \dot{E}_{Di}}{\sum \Delta \dot{E}_{xin}} \tag{38}$$

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:

$$\psi_i = \frac{W_{net}}{\dot{E}_{x fuel}} \tag{39}$$

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

$$\xi = \frac{\dot{E}_{D Total}}{W_{net}} \tag{40}$$

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	$\dot{E}_{DC} = \dot{E}_{in} - \dot{E}_{out} + W_C$	$\varepsilon = \frac{\dot{E}_{out} - \dot{E}_{in}}{W}$
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	$\dot{E}_{DT} = \dot{E}_{in} - \dot{E}_{out} - (W_{net} + W_C)$	$\varepsilon = \frac{W_{net} + W_C}{\dot{E}_{in} - \dot{E}_{out}}$
Total exergy destruction rate	$\dot{E}_{D Total} = \sum \dot{E}_D = \dot{E}_{DC} + \dot{E}_{DCC} + \dot{E}_{DT}$	

RESULTS AND DISCUSSION

Effects of Operating Conditions on Performance of Gas Turbine Plants

The simulation results of the outcome 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 outcome of operating atmospheric situations on gas turbine is measured purposely to show

sensitivity of gas turbine performance to the environmental condition. The results of this section as presented in Figures 2–13 are based on theoretical relationships earlier presented.

Effect of Compression Ratio

Figure 2 shows the relation between cycle thermal efficiency and compression ratios for turbine inlet temperatures (TITs) between 900 and 1500 K. 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. [33] 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 1600 K 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.

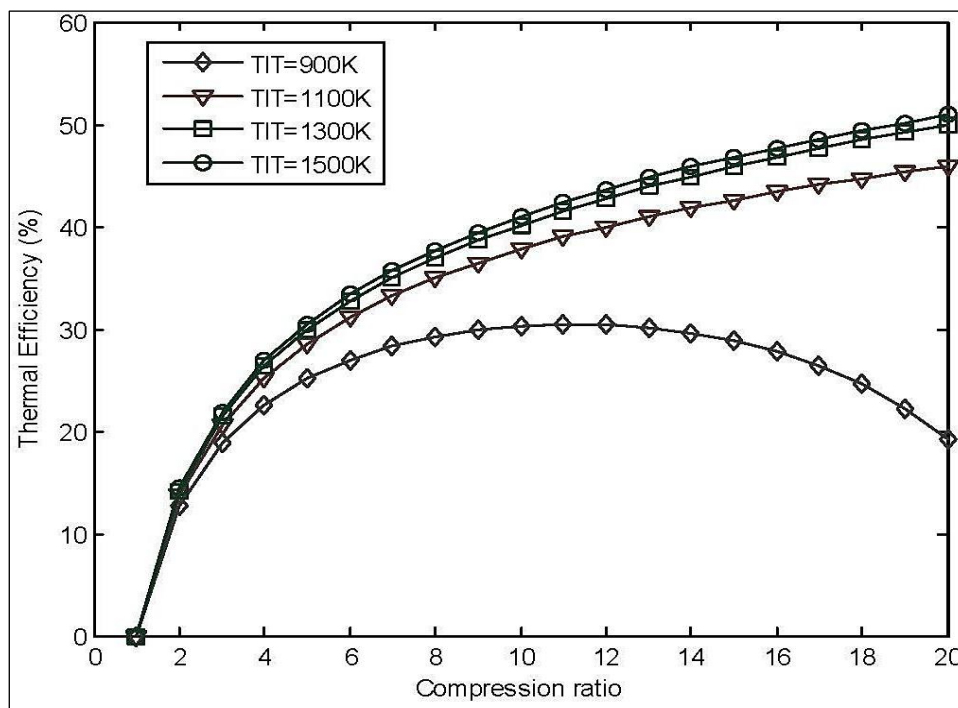


Fig. 2. Effect of Compression Ratio and TIT on Thermal Efficiency.

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 [33, 34] reveals an acceptable agreement.

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.

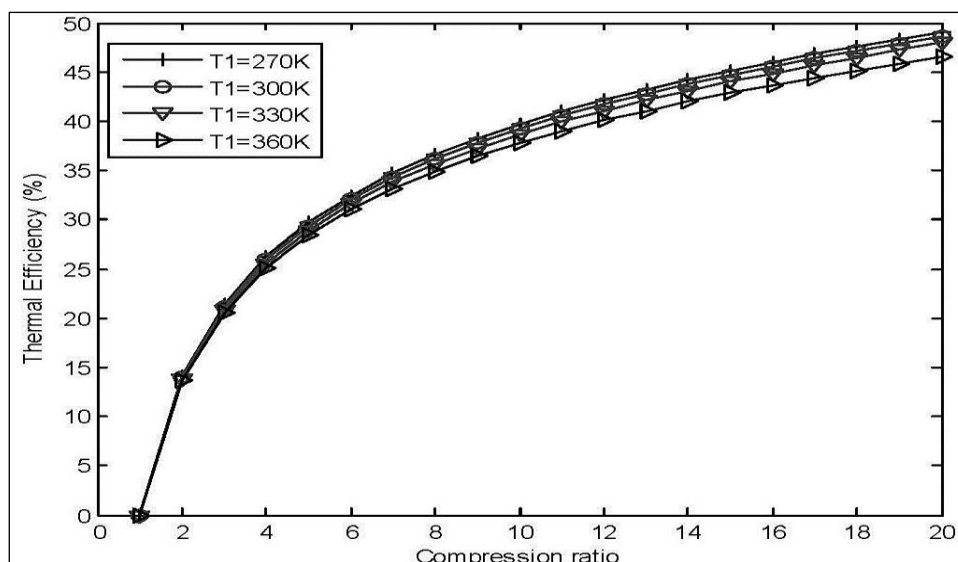


Fig. 3. Effect of Compression Ratio and Ambient Temperature (T_1) on Thermal Efficiency.

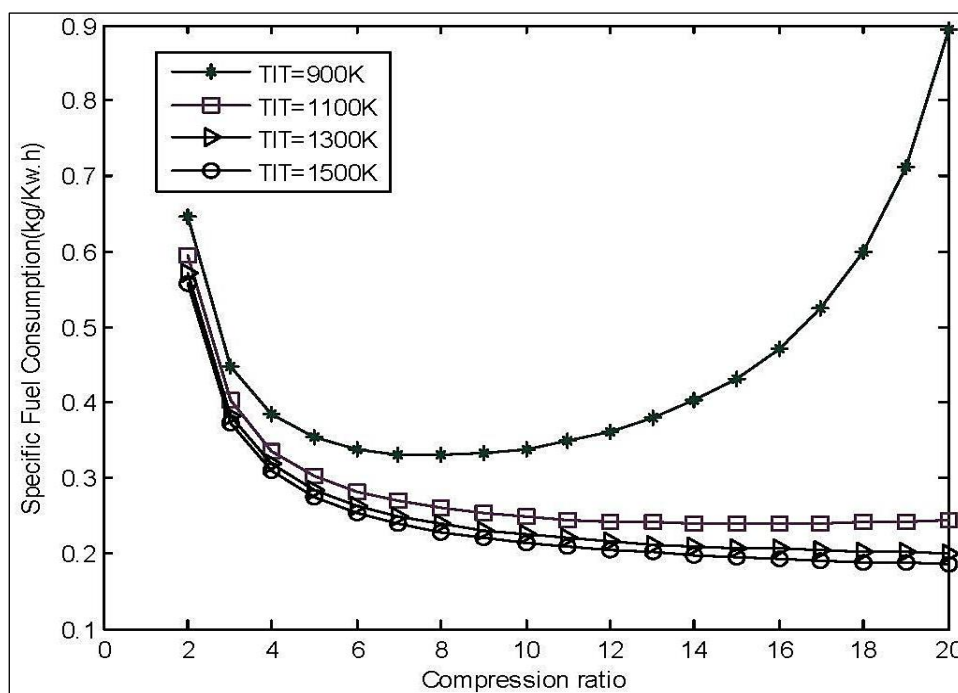


Fig. 4. Effect of Compression Ratio and TIT on Specific Fuel Consumption.

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

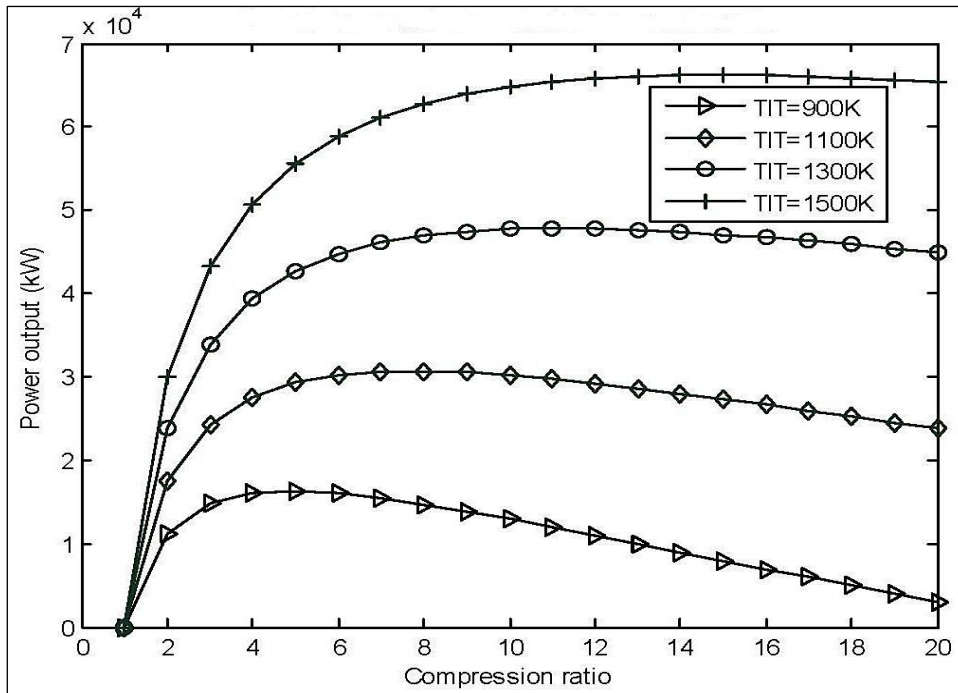


Fig. 5. Effect of Compression Ratio and TIT on Power Output.

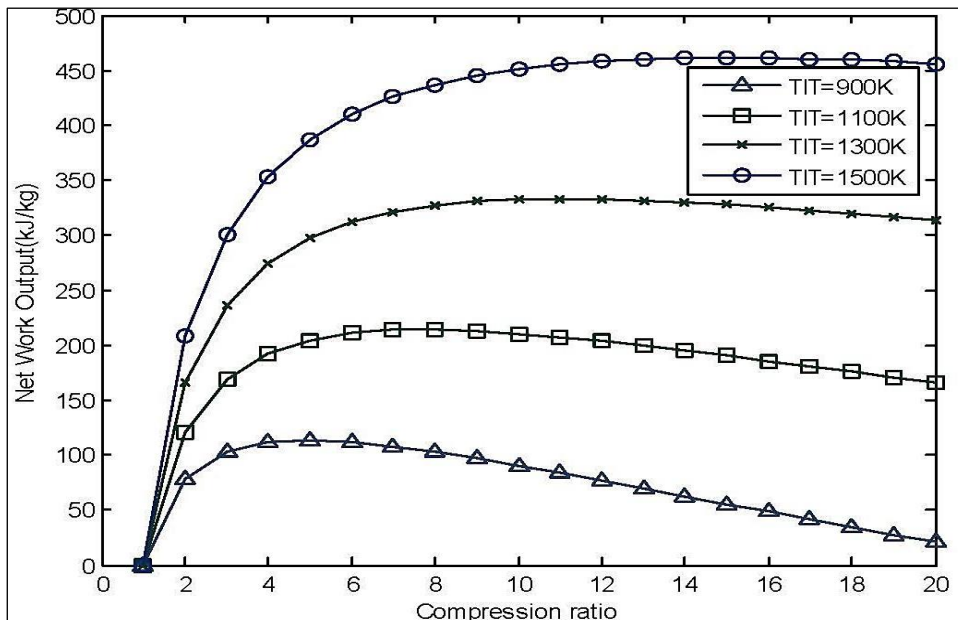


Fig. 6. Effect of Compression Ratio and TIT on Network Output.

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, rise in compression ratio for the same turbine inlet temperature cuts the heat supplied. 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.

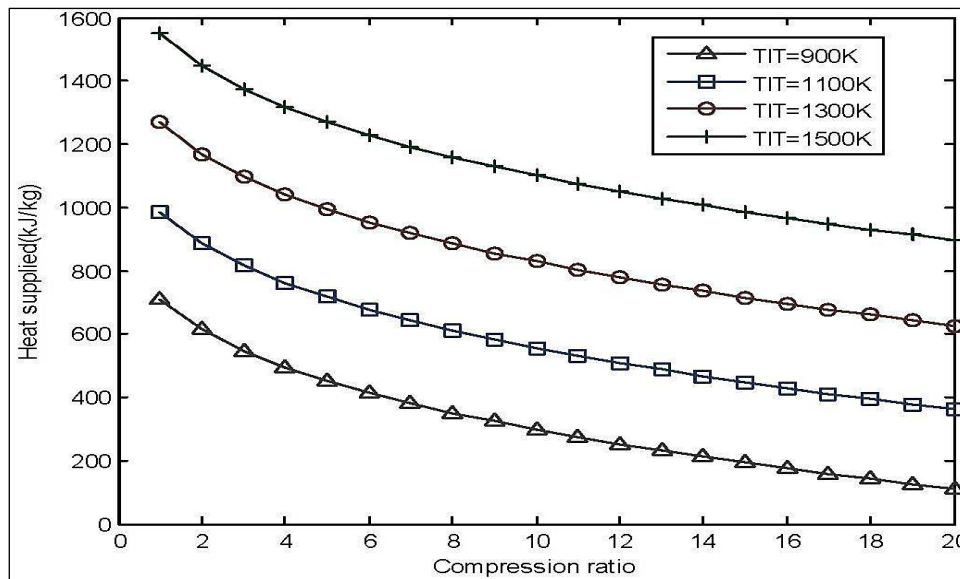


Fig. 7. Effect of Compression Ratio and TIT on Heat Supplied.

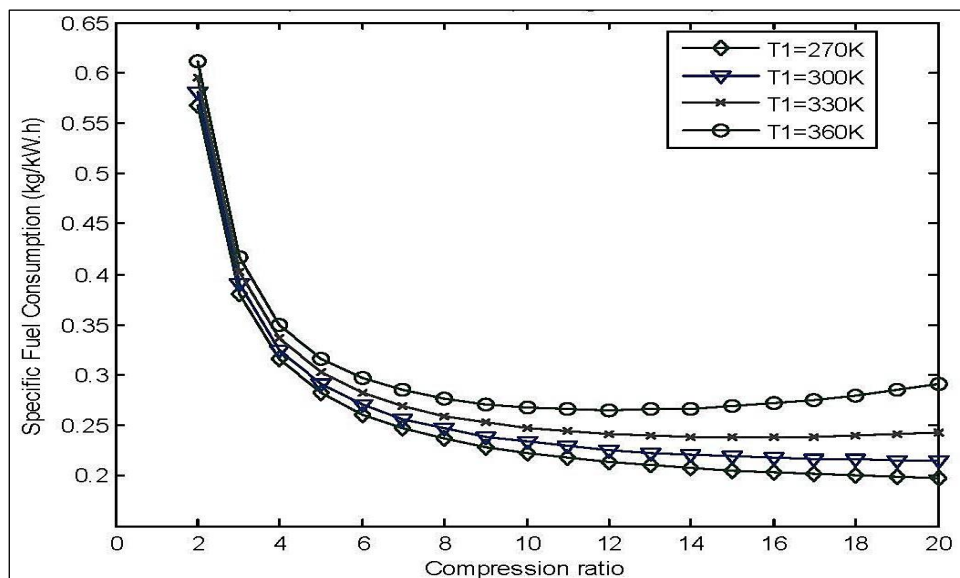


Fig. 8. Effect of Ambient Temperature and Compression Ratio on Specific Fuel Consumption.

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, [35] thereby reducing the network 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.

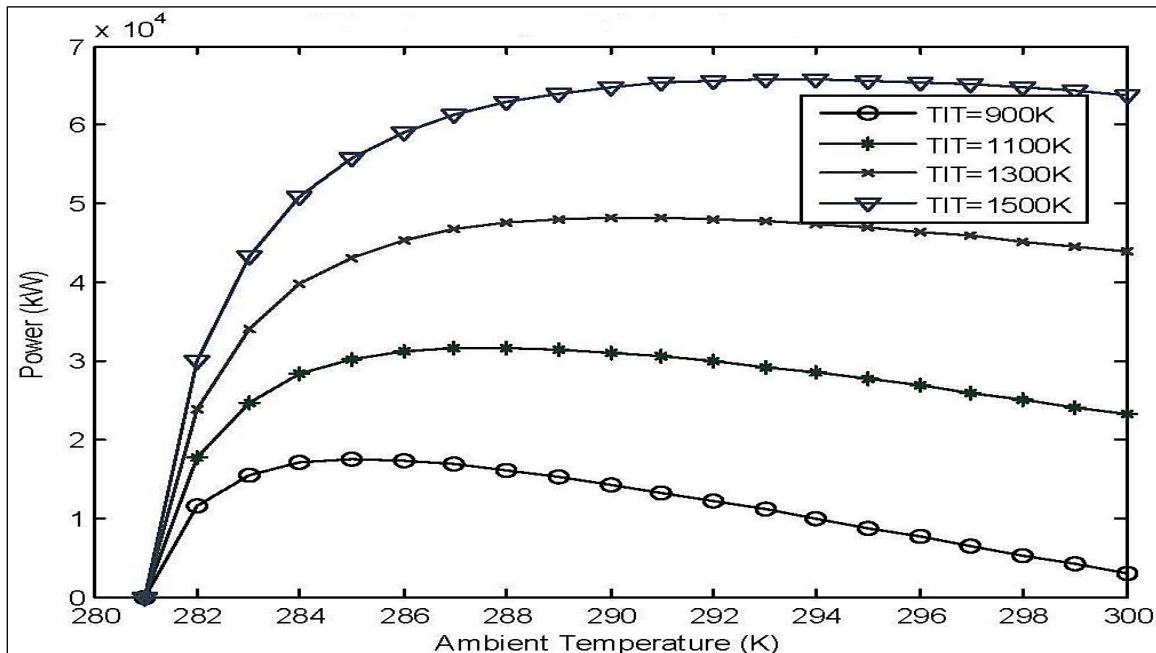


Fig. 9. Effect of Ambient Temperature and TIT on Power Output.

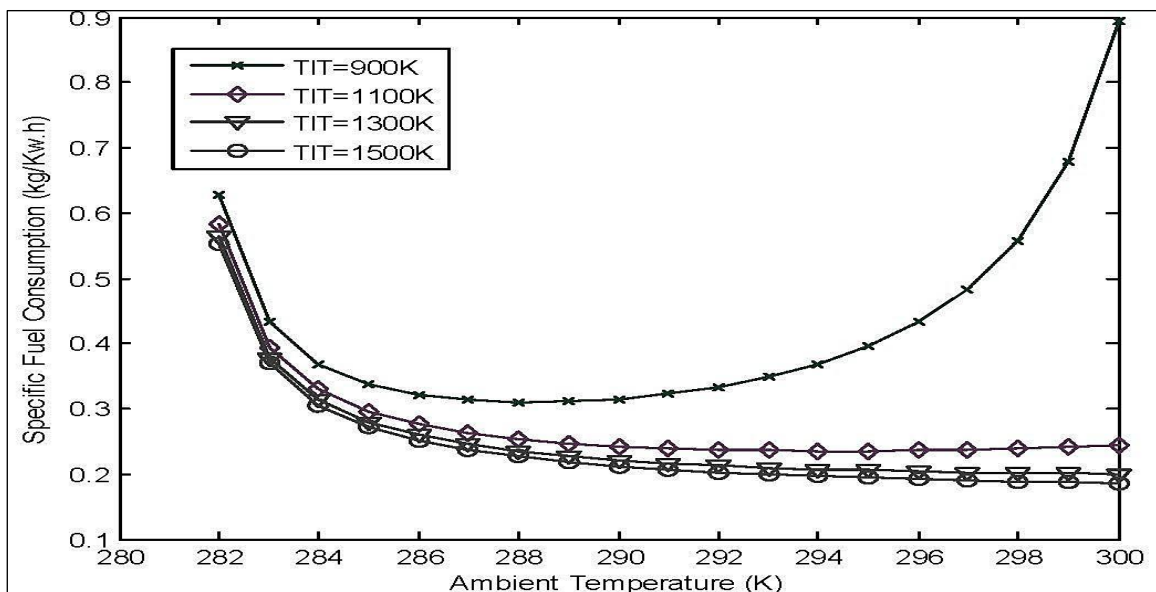


Fig. 10. Effect of Ambient Temperature and Turbine Inlet Temperature on Specific Fuel Consumption.

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 T_1 . The increase in power output due to turbine inlet temperature is as a result of the network 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 direct to a

maximum air density and a lesser 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.

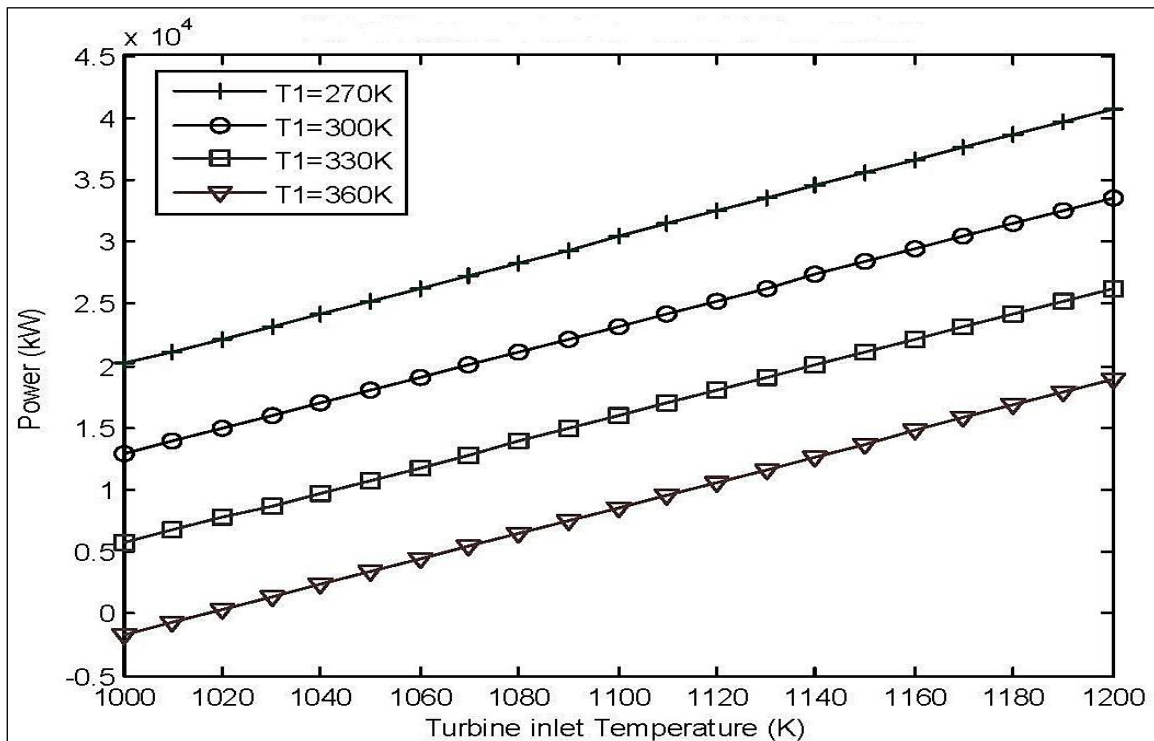


Fig. 11. Effect of Turbine Inlet Temperature and Ambient Temperature on Power Output.

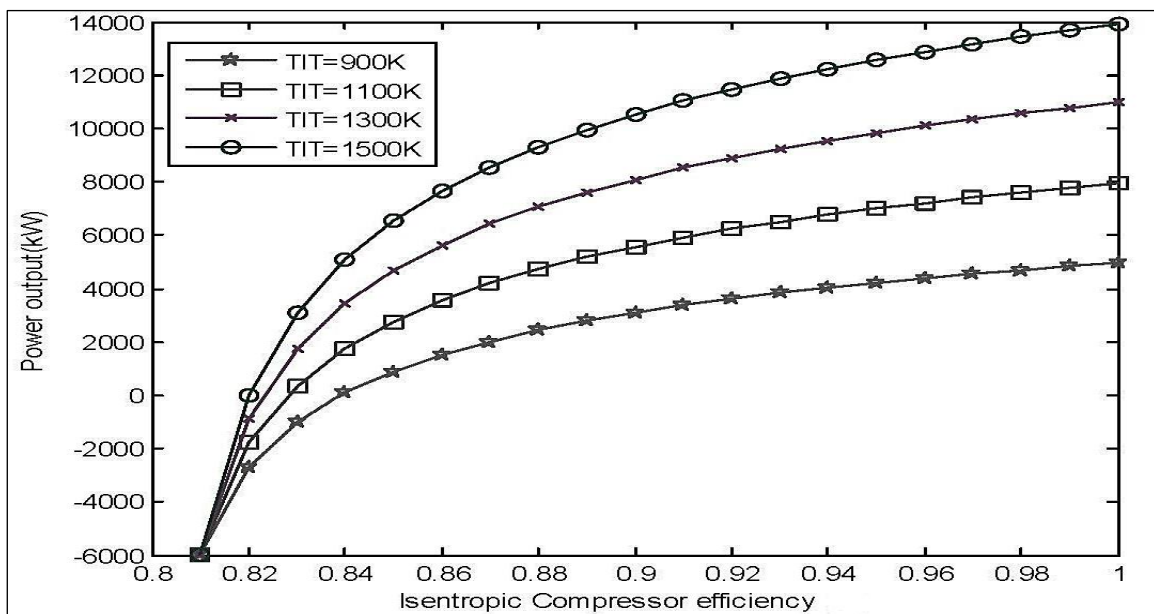


Fig. 12. Effect of Turbine Inlet Temperature and Isentropic Compressor Efficiency on Power Output.

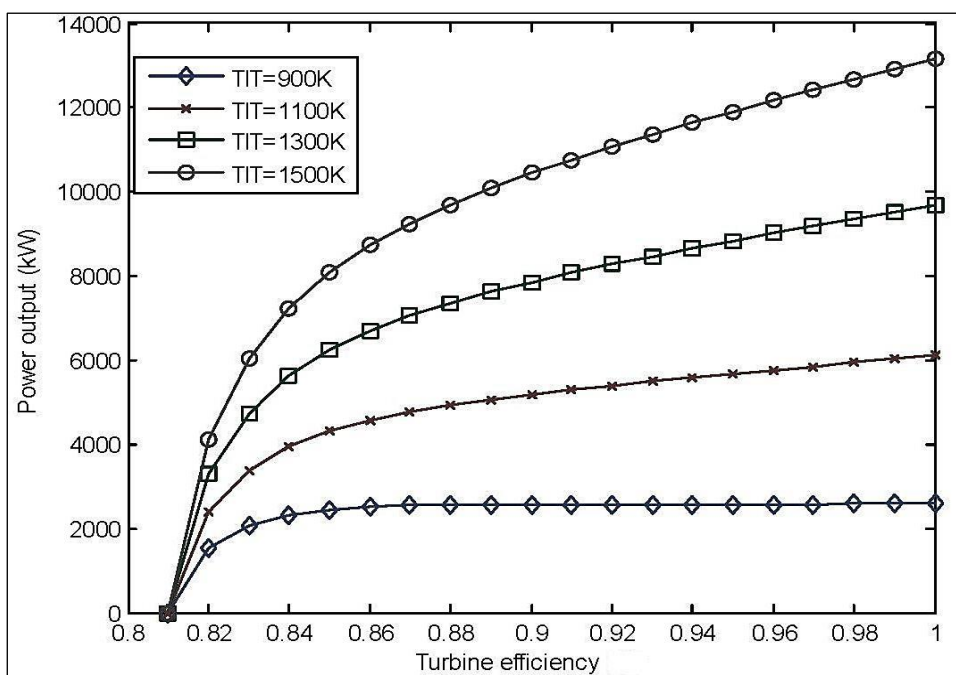


Fig. 13. Effect of Turbine Inlet Temperature and Isentropic Turbine Efficiency on Power Output.

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 amount of increase in power output is more important at higher TIT and higher isentropic compressor and turbine efficiencies.

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

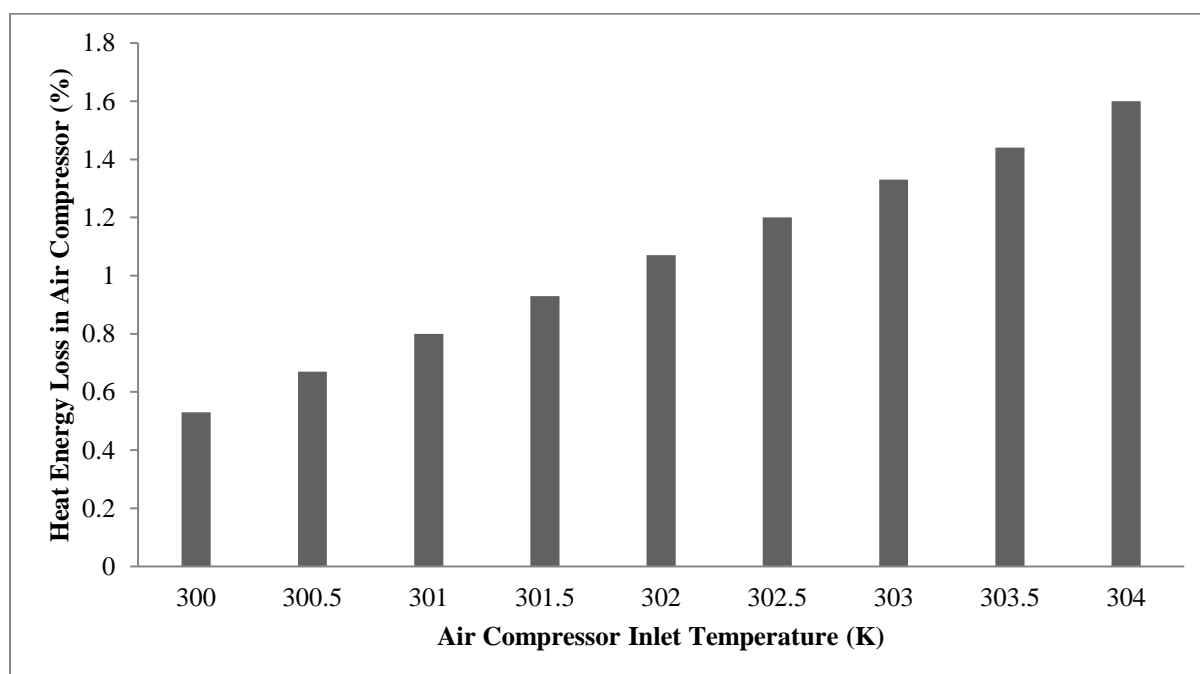
used to explain efficiency from the exergetic point of view. These two indicators (1st law efficiency and 2nd law efficiency) have wide range of application at system and component level. [36, 37] A complete analysis of the thermodynamic performance of a process generally requires the use of both energy and exergy analyses.

Energy Analysis

The average operating data of the selected gas turbine power plant for the period of six years (2005–2010) 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

**Fig. 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 network 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 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 the increase in air mass flow rate, which 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.

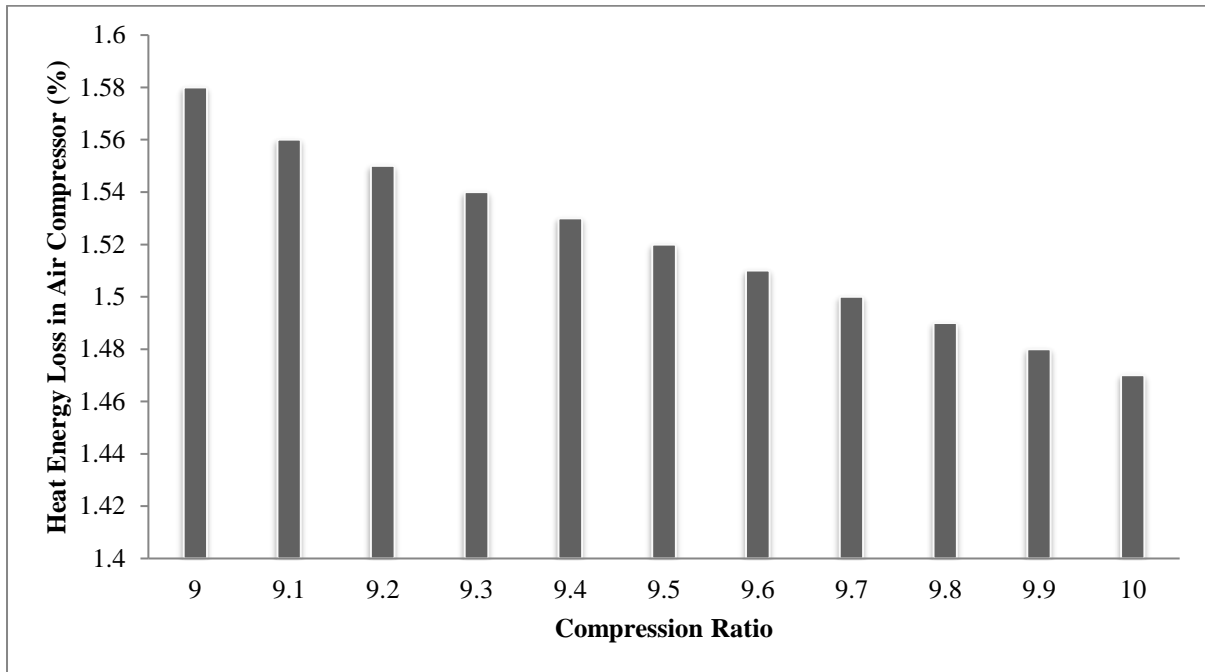


Fig. 15. Heat Energy Loss in Air Compressor (%) Against Compression Ratio.

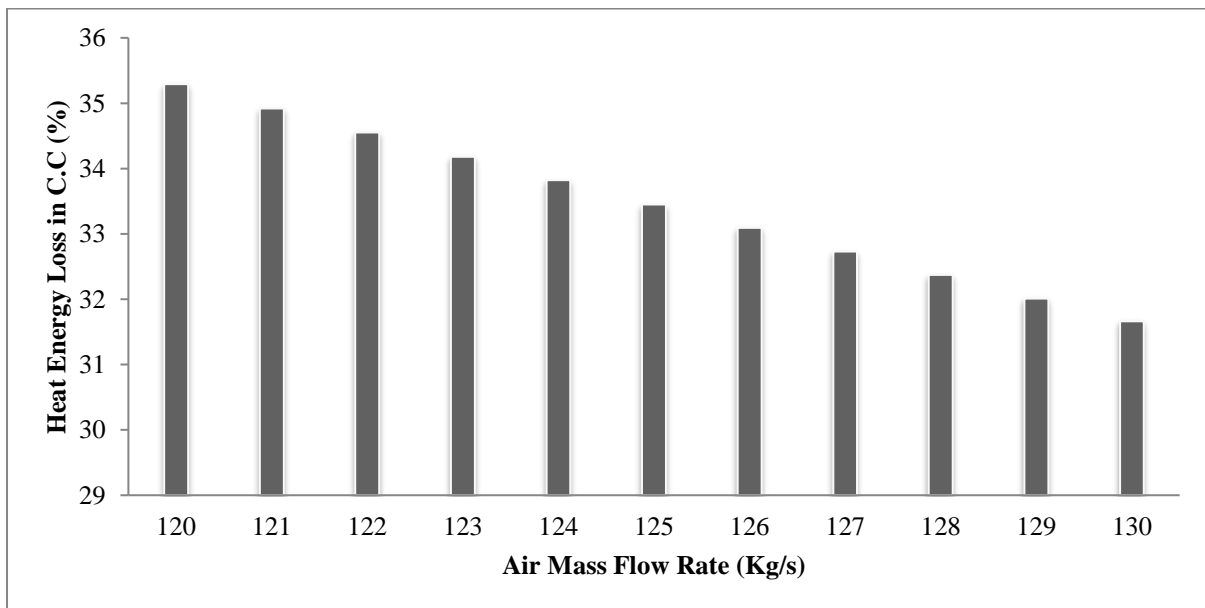


Fig. 16. Heat Energy Loss in Combustion Chamber (%) Against Air Mass Flow Rate (kg/s).

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 differs from power plant to power plant based on either technical deficits 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 confusing as quantifying actual losses in the different system control volumes might not be exactly 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 associated with those of exergetic analysis allowing an improved understanding by quantifying the effect of irreversibility occurring in the plant and the locations.

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 make the simulation of the performance of the parts

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 chief source of thermodynamic inefficiency in the plant considered and this is due to the irreversibility related 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.

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

By comparing data in Tables 3 and 4, total plant loss 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 rise in exergetic efficiency with rise in turbine inlet temperature is restricted by turbine material temperature limit. This can be seen from the plant competence 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. [32]

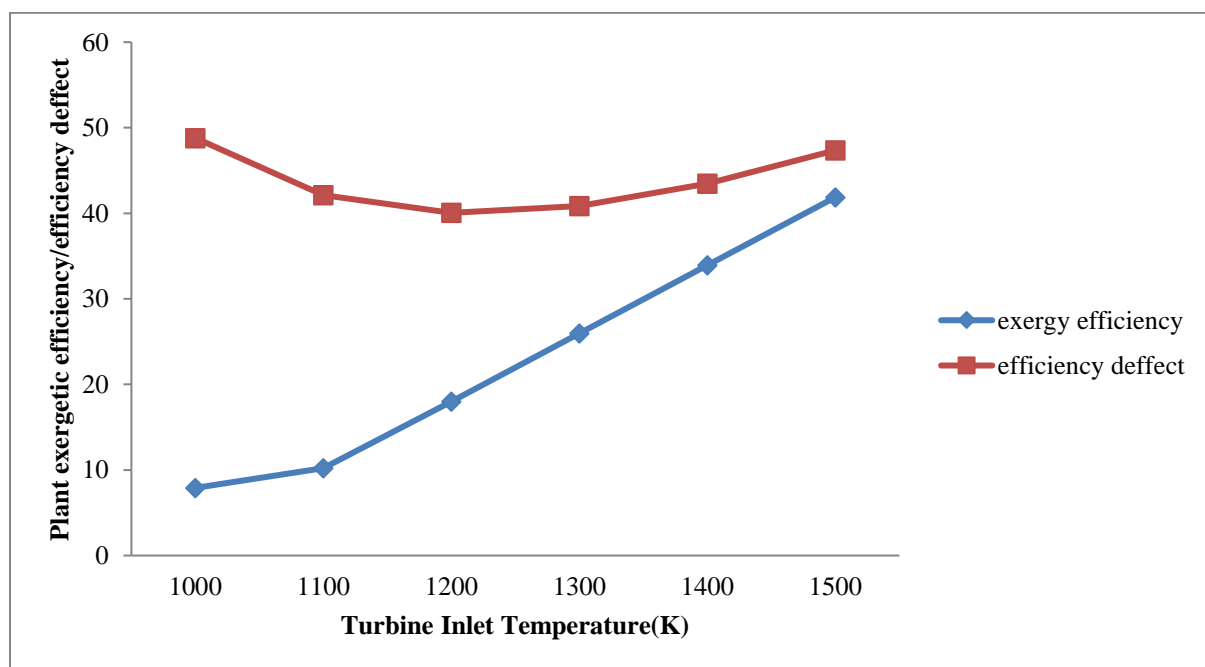


Fig. 17. Variation in Plant Exergetic Efficiency and Efficiency Defect With Turbine Inlet Temperature.

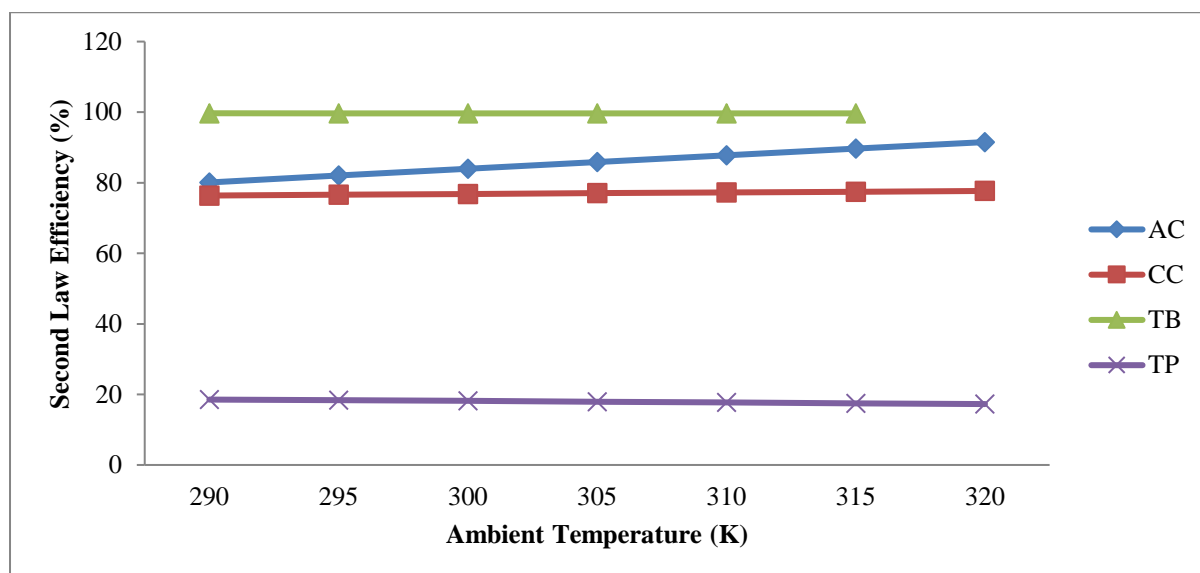


Fig. 18. Variation in Second-Law Efficiency With Ambient Temperature. AC, Second Law Efficiency of Compressor; CC, Second Law Efficiency of Combustion Chamber; TB, Second Law Efficiency of Turbine; TP, Second Law Efficiency of Entire Plant.

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 characteristics considered in this study in gas turbine engine during simulation are inlet temperature of the air compressor, the mass flow degree of fuel and turbine inlet temperature. The results of thermodynamic properties of the cycle from the modelling 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 huge amount of energy present in the fuel, with great volume to generate useful work, is being wasted. Moreover, the results show that an rise in the turbine inlet temperature (TIT) directs 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 flaws 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 incredible 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 lets the turbine operator to achieve meaningfully greater temperature rises 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.^[38]

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 innovative cycle would recover its performance significantly. Among many proven technologies is 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, refining the performance of the compressor will have a proportionately greater influence 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 fight corrosion, which can be a noteworthy source of performance degradation, particularly if a turbine is situated near saltwater. As AES Barge gas turbine plant is located on lagoon, compressor coating technology would improve the plant performance significantly.

ACKNOWLEDGEMENTS

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NOMENCLATURES

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

REFERENCES

1. Reddy V.S., Kaushik S.C., Panwar N.L. Review on power generation scenario of India, *Renew Sustain Energy Rev.* 2013; 18: 43–8p.
2. Ghazikhani M., Khazaee I., Abdekhodaie E. Exergy analysis of gas turbine with air bottoming cycle, *Energy.* 2014; 72: 599–607p.
3. Kreith F., Goswami Y.D. Eds. *The CRC Handbook of Mechanical Engineering.* 2nd Edn., CRC Press; 2005.
4. Chaker M., Meher-Homji C.B., Mee T., *et al.* Inlet fogging of gas turbine engines detailed climatic analysis of gas turbine evaporation cooling potential in the USA, *J Eng Gas Turbine Power.* 2003; 125: 300–9p.
5. Jaber Q.M., Jaber J.O., Khawaldah M.A. Assessment of power augmentation from gas turbine power plants using different inlet air cooling systems, *Jordan J Mech Ind Eng.* 2007; 1(1): 7–15p.
6. Oyedepo S.O., Kilanko O. Thermodynamic analysis of a gas turbine power plant modeled with an evaporative cooler, *Int J Thermodyn (IJoT).* 2014; 17(1): 14–20p.
7. Erdem H.A., Sevilgen H.S. Case study: effect of ambient temperature on the electricity production and fuel consumption of simple cycle gas turbine, *Appl Therm Eng.* 2006; 26: 230–6p.
8. Mahmoudi S.M., Zare V., Ranjbar F., *et al.* Energy and exergy analysis of simple and regenerative gas turbines inlet air cooling using absorption refrigeration, *J Appl Sci.* 2009; 9(13): 2399–407p.
9. Guinee J.B. (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; 2001.

10. Sadrameli S.M., Goswami D.Y. Optimum operating conditions for a combined power and cooling thermodynamic cycle, *Appl Energy*. 2007; 84: 254–65p.
11. Kakaras E., Doukelis A., Karellas S. Compressor intake-air cooling in gas turbine plants, *Energy*. 2004; 29: 2347–58p.
12. Kamal N.A., Zuhair A.M. Enhancing gas turbine output through inlet air cooling, *Sudan Eng Soc J*. 2006; 52(46): 1–8p.
13. Ameri M., Ahmadi P., Khanmohammadi S. Exergy analysis of a 420MW combined cycle power plant, *Int J Energy Res*. 2007. DOI: 10.1002/er.1351.
14. Hall A.D., Stover J.C., Breisch R.B. Gas turbine inlet-air chilling at a cogeneration facility, *ASHRE Trans*. 1994; 100(Part 1).
15. Bassily A.M. Effects of evaporative inlet and after cooling on the recuperated gas turbine cycle, *Appl Therm Eng*. 2001; 21(18): 1875–90p.
16. Gareta R., Romeo L.M., Gil A. Economic optimization of gas turbine inlet air – cooling systems in combined cycle applications, Center for Power Plant Efficiency Research, *ECOS*. 2005; 1: 409–15p.
17. Lior N., Zhang N. Energy, exergy, and second law performance criteria, *Energy*. 2007; 32: 281–96p.
18. Bilgen E. Exergetic and engineering analyses of gas turbine based cogeneration systems, *Energy*. 2000; 25: 1215–29p.
19. Ray T.K., Ganguly R., Gupta A. 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, 1–8p.
20. Khaldi F., Adouane B. Energy and exergy analysis of a gas turbine power plant in Algeria, *Int J Exergy*. 2011; 399–413p.
21. Chen Q., Han W., Zheng J., et al. 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, *Appl Therm Eng*. 2014; 66: 590–602p.
22. Barzegar-Avval H., Ahmadi P., Ghaffarizadeh A.R., et al. ‘Thermo-economic-environmental multi-objective optimization of a gas turbine power plant with preheater using evolutionary algorithm, *Int J Energy Res*. 2011; 35: 389–403p.
23. Kurt H., Recebli Z., Gredik E. Performance analysis of open cycle gas turbines, *Int J Energy Res*. 2009; 33(2): 285–94p.
24. Tahouni N., Jabbari B., Panjeshahi M.H. Optimal design of a cogeneration system in a kraft process using genetic algorithm, *Chem Eng Trans*. 2012; 29: 19–24p.
25. Adrian I., Dorin L. thermodynamic analysis of gas turbine powered cogeneration systems, *J Sci Ind Res*. 2010; 69: 548–53p.
26. Mozafari A., Ahmadi A., Ehyaei M.A. Optimisation of micro gas turbine by exergy, economic and environmental (3E) analysis, *Int J Exergy*. 2010; 7(1): 1–19p.
27. Alcides C.N. Assessment of novel power generation systems for biomass industry, *PhD Thesis*, Cranfield University, 1999.
28. Oh S.D., Pang H., Kim S., et al. Exergy analysis for a gas-turbine cogeneration system, *J Eng Gas Turbine Power*. 1996; 118: 782–91p.
29. Ebadi M.J., Gorji – Bandpy M. Exergetic analysis of gas turbine plants, *Int J Exergy*. 2005; 2: 31–9p.
30. Abam F.I., Ugot I.U., Igbong D.I. Thermodynamic assessment of grid – based gas turbine power plants in Nigeria’, *J Emerg Trends Eng Appl Sci*. 2011; 2(6): 1026–33p.
31. Bejan A., Tsatsaronis G., Moran M. *Thermal Design and Optimization*.

- New York: John Wiley & Sons, Inc.; 1996.
32. Kotas T.J. *The Exergy Method of Thermal Plant Analysis*. Malabar, Florida: Krieger Publishing Company; 1995.
33. Eastop T.D., McConkey A. *Applied Thermodynamic for Engineering Technologists*. 5th Edn., India: Pearson Educational Ltd.; 2009, 290p.
34. Rajput R.K. *Engineering Thermodynamics (S.I Units)*. Golden House, Daryaganj, New Delhi: Laxmi Publications (P) LTD 113; 2007.
35. Nag P.K. *Power Plant Engineering*. New Delhi: Tata McGraw-Hill Publishing Company Limited; 2008.
36. Mirandola A., Stoppato A., Tonon S. An integrated approaches to the assessment of energy conversion plants, *Int J Appl Thermodyn*. 2000; 3(3): 111–9p.
37. Oyedepo S.O. Thermodynamic performance analysis of selected gas turbine power plants in Nigeria, *Ph.D Thesis*, Covenant University, Ota, Nigeria, 2014.
38. FTT (Florida Turbine Technologies. Inc.). *Turbine Efficiency Improvements for Existing Power Plants*. 2009, 1–8p.