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Performance Assessment of a Gas Turbine Power Plant

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Abstract:** This study focuses on the energy-exergy performance evaluation of a 5.67MW rated gas turbine power plant located at Total Exploration and Production Port Harcourt Office complex in Nigeria. Design data were collected from the installation document and temperature readings from the control room. A MATLAB program was written that utilized the data collected and various thermodynamic equations to output various performance parameters. The simulation of ambient air temperature on the performance of the gas turbine power plant was investigated. The results show that that there was an increase of 46.1176kW in the work done by the compressor for every 1o rise in the ambient air temperature, an increase of 33.3888kW in the net power generated per 1o rise in the ambient air temperature, an increase of 28.71089kJ/kW in the heat rate per every 1o rise in ambient air temperature and a decrease of 0.06287% in the thermal efficiency of the plant for every 1o rise in ambient air temperature. The exergy assessment showed that the combustion chamber was the most exergy inefficient component of the gas turbine as it had an exergy efficiency of just 59.168% with an average exergy destruction of 9368.507kW, the turbine section had an exergy efficiency of 72.997% with an average exergy destruction of 3661.844kW and the compressor section of the power plant is the most exergy efficient component as it has an exergy efficiency of 83.409% with an average exergy destruction of 676.107kW. This study showed the shortcomings of the gas turbine power plant and offered recommendations to improve efficiency.
***Keywords:** Energy, Exergy, Plant Efficiency, Net Power, Thermal Efficiency, Work-done*

I. INTRODUCTION

This work considers the performance assessment of the gas turbine power plant at Total Exploration and Production Nigeria Limited situated at Port Harcourt, Nigeria. The study comprises both energy-based and exergy-based performance assessment for a holistic view of the indices that track the performance of the plant. The most prevalent system performance assessment criteria used for power plants are energy-based or first law-based criteria. Energy-based performance assessment are instrumental in the evaluation of a plant and can be interpreted in terms of monetary cost if the monetary values of the output such as net power, and input such as fuel and maintenance costs, are known (Lior & Zhang, 2005).

Second law-based or exergy-based system performance assessment criteria consider the differences between the performance of a real system versus an ideal (reversible) system operating between similar thermodynamic limits. The energy-based or first law-based criteria mainly account for the work done by the plant, but fail to account for the maximum possible available work. Exergy-based assessments take into account the maximum possible available work and can render much better recommendations for system improvement.

The aim of this study is to conduct an energy-exergy performance analysis of the gas turbine power plant at Total Exploration and Production Nigeria Limited, Port Harcourt office complex, Nigeria.

For the sake of achieving the aim of this study, the following objectives would be required:

1) To determine compressor work, turbine work and the thermal efficiency of the power plant.
2) To determine the amount of exergy destroyed in each component of the plant.
3) To compare calculated performance data with collected design data.

II. LITERATURE REVIEW

Ankit et al. (2017) carried out a thermodynamic assessment of an open cycle gas turbine power plant. Mathematical formulations that describe the specific work and efficiency were derived and analyzed. The effects of the thermodynamic operating parameters on the plant such as the ambient temperature, the relative humidity, compressor pressure ratio, turbine inlet temperature, air-fuel ratio, isentropic efficiencies of compressor and turbine, net power output and the heat rate of a gas turbine plant were investigated. A MATLAB code was developed and the performance data generated using that code was utilized to draw various related graphs.

The results show that the compression ratio, ambient temperature, air-fuel ratio and the isentropic efficiencies can have a strong effect on the thermal efficiency. Additionally, the thermal efficiency and power output decreased linearly as the ambient temperature and air-fuel ratio increased. Also, fuel consumption and heat rate increased linearly with increase in both ambient temperature and air-fuel ratio. They also described various technologies targeted at effecting a positive change in the performance of gas power plants.

The observable performance of gas turbines, however they are operated, whether as simple cycles or as combined cycles, is limited by the ambient conditions, especially in arid or tropical climates that are predominant in Sub-Saharan African countries such as Nigeria. In a study by Ukwamba et al. (2018) a performance evaluation of a simple gas turbine power plant using a vapor-absorption-chiller was performed. A similar power plant was modeled using IPSEpro software and which used heat wasted from exhaust gases of the base power plant to produce water vapor (steam) in a heat exchanger. Some of the generated steam was effectively used to run a lithium-bromide single-effect vapour absorption chiller which consequently conditioned the compressor air intake. The results obtained showed that the vapour absorption cooling system reduced the inlet temperature to the compressor from 25.7^{o}C to 15.1^{o}C. It was also observed that the mass of air increased by 13.23kg/s, consequently boosting the power output with an increment of 3.5MW and increasing the thermal efficiency by 1.12%. Furthermore, the specific fuel consumption reduced by 0.0079Kg/KWh and the net power plant heat rate reduced by 369KJ/KWh. This change in the heat rate signified that less energy is consumed to generate a net power output, and if retrofitted, the gas turbine power plant would be more efficient than the existing plant. Additionally, greenhouse gas emission saving was analyzed. The result showed a net savings of 276.654kg greenhouse gases (Ukwamba et al., 2018).

Adegboyega and Famoriji (2013) examined basic factors such as plant capacity, plant utilization, load factor, and utilization to evaluate and estimate the key performance indices of a gas turbine central power plant. Data were obtained from the Edjeba gas turbine power plant in Delta State, Nigeria. These were monthly energy production inventory and operational statistics from 2002 to 2012. The plant capacity, plant utilization factor, load factor, and utilization factor were determined from the data. The capacity factor of the Edjeba gas turbine power plant was 20.4% as against the target of 40-65% of ISO standard, the plant utilization factor was 29.14546% as against the target of 50-70% of ISO standard, and the load factor was 81.76% as against 80% of ISO standard, and a utilization factor of 49.1-58.9% as against 85% of ISO standard.

Egware and Obanor (2013) considered the use of exergy analysis in assessing the performance indices of Omotosho Phase I gas-fired power plants. The data utilized in that study were taken from the data logging books. Exergetic analysis and the laws of mass and energy conservation were applied to each component. The results obtained showed that the turbine exergy efficiency was 96.17%, combustor exergy destruction was 54.15% and the overall plant exergy efficiency was 41.83%. In addition, investigation was carried out on the varying effects of ambient temperatures between 21ºC and 33ºC on the gas turbine. It was noticed that as the ambient temperature increased, exergy efficiency decreased. Therefore, to improve system performance, an air intake cooling system was suggested for the plant.

III. MATERIALS AND METHODS

A. Overview

The following methods were employed to implement the objectives of this study:

1) Data were collected directly from the installation documents, parameters that could not be directly measured were arrived at using applicable existing equations.
2) Temperature readings were collected from the gas turbine control room.
3) Steady flow energy equations and the second law of thermodynamics were applied to the different parts of the gas turbine power plant.
4) Exergy equations were also applied.
5) MATLAB R2015a and Microsoft Excel were used to analyze data and calculate certain performance criteria.

B. Materials

The power plant is composed of three gas turbines running on compressed natural gas each having a capacity of 5.67MW. It powers a 33/11KV substation and distribution, supplying all office complexes at Total Exploration and Production Nigeria (TEPNG) Port Harcourt. The plant operates with a compression ratio of 12.2 and a compressor flow rate of 77760kg/h. The design turbine exit temperature is 510^{o}C, exhaust flow is 78385kg/h and fuel consumption rate is 8.3439kg/h.

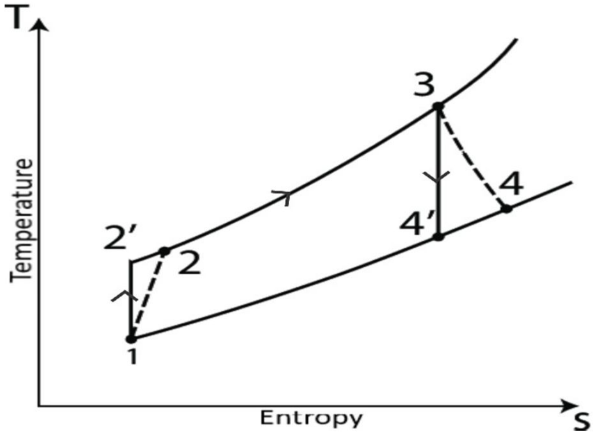

Figure 1: Brayton Cycle T-s chart

Figure 1 represents the Brayton cycle that is a thermodynamic cycle that describes the gas turbine operation. Air at ambient conditions would be drawn into the compressor by a number of rotor and stator stages at state 1 from the surroundings and later returned to the surroundings at state 4 with a temperature greater than the surrounding temperature. In Figure 1, the line process 1-2ˈ represents an isentropic compression, the line process 2ˈ-3 represents a constant pressure heat addition process in the combustor while line 3-4ˈ shows an isentropic expansion in the turbine. After mixing with the environment, the unit mass of the air exhausted from the turbine power plant eventually returns to the ambient condition, as the air sucked into the compressor, so the air flowing throughout the components of the gas turbine is considered to have gone through a thermodynamic cycle. In actual plant operation, however, the compression and expansion processes are not isentropic and the broken lines in Figure 1 indicate the actual processes. Also, there is a usual pressure drop in the combustor of the actual plant.

C. Methods

The methods used in this work include:

1) Compressor Analysis

According to Rahman *et al*. 2011, the compressor efficiency η_c is given by

$$\eta_c = \frac{isentropic\ compression\ work}{actual\ compression\ work} \times 100\% \qquad (1)$$

The work done by the compressor is given as (Lebele-Alawa & Jo-Appah, 2015)

$$W_c = m_a C_{pa}(T_2 - T_1) \qquad (2)$$

where m_a is the mass flow rate of air into the compressor and C_{pa} is the specific heat capacity of air.

The specific heat capacity of any substance at constant pressure may be subject to the prevailing temperature, hence all specific heats used in this study will be calculated using the polynomial form (Cengel & Boles, 2006)

$$C_p = \frac{a+bT+cT^2+dT^3}{M} \qquad (3)$$

where a,b,c,d are constants, T is the prevailing temperature and M is the molar mass.

2) Compressor Exergy Analysis

The chemical equation representing energy exchange for the total mass flow rate of air into the compressor is given as (Rajput, 2007)

$$\{(0.2319O_2 + 0.7681N_2)\}m_a \xrightarrow{energy\ added} \{(0.2319O_2 + 0.7681N_2)\}m_a \quad (4)$$

The total exergy of a system can be separated into four components and is given as (Bejan, 1996)

$$E = E^{PH} + E^{CH} + E^{KN} + E^{PT} \qquad (5)$$

where E^{PH} is the physical exergy of the system, E^{CH} the chemical exergy of the system, E^{KN} the kinetic energy of the system, and E^{PT} is the potential exergy of the system. The levels of potential and kinetic exergies are assumed to be zero and since no chemical reaction or combustion is observed in the turbine and compressor, the chemical exergy of both components is assumed to be zero (Awuladin *et al*., 2016).

The total exergy of the air entering the compressor is given as (Truls, 2009).

$$E_1 = \{0.2319(\Delta h - T_o\Delta s)_{O_2} + 0.7681(\Delta h - T_o\Delta s)_{N_2}\}m_a \quad (6)$$

where Δh is the change in enthalpy from the atmospheric conditions in KJ/kg, T_o is the atmospheric temperature in K and Δs is the change in entropy from the atmospheric conditions in KJ/kgK and is given as (Awaludin *et al.*, 2016)

$$\Delta s = C_p \ln\left(\frac{T}{T_o}\right) - R\ln\left(\frac{P}{P_o}\right) \quad (7)$$

where T and P refer respectively to the prevailing temperature and pressure conditions and P_o is the atmospheric pressure.

The total exergy of the system after compression is also given as

$$E_2 = \{0.2319(\Delta h - T_o\Delta s)_{O_2} + 0.7681(\Delta h - T_o\Delta s)_{N_2}\}m_a \quad (8)$$

(similar to equation 6 but in different condition)

The exergy destroyed by each component of the gas turbine power plant is given as (Awaludin *et al.*, 2016)

$$E_D = E_{in} - E_{out} \quad (9)$$

where E_{in} is the exergy input into the component and E_{out} is the exergy output of the component.

The exergetic efficiency of each component is given as (Awaludin *et al.*, 2016)

$$\eta_e = \frac{E_{out}}{E_{in}} \quad (10)$$

3) *Combustor Analysis*

Specific fuel consumption (SFC) is the ratio of fuel used by a machine to the net power the machine produces and it is given as (Lebele-Alawa & Jo-Appah, 2015)

$$SFC = \frac{3600 \times m_f}{W_{net}} \quad (11)$$

where m_f is the mass flow rate of the fuel and W_{net} is the net power output

The heat supplied Q_{add} is given as (Lebele-Alawa & Jo-Appah, 2015)

$$Q_{add} = m_p C_{pa}(T_3 - T_2) \quad (12)$$

where m_p is the mass flow rate of products.

4) *Combustor Exergy Analysis*

The mass flow rate of the fuel in gas turbines is very minute compared to the amount of air flow. Hence, we must balance the equation of combustion that represents the combustion reaction that occurs in the combustor. Equation 13 shows the reactants that are sent into the combustor. The power plant runs on natural gas, hence methane is used for the combustion reaction equations.

$$reactants \rightarrow m_f CH_4 + m_a(0.2319O_2 + 0.7681N_2) \quad (13)$$

The chemical exergy of the fuel entering the combustion chamber is taken into account. An approximation for the chemical exergies of hydrocarbon fuels of the form C_xH_y is given as (Moran *et al.*, 2014)

$$E_f^{CH} \cong m_f \times LHV\left\{1.033 + 0.0169\frac{y}{x} - \frac{0.0698}{x}\right\} \quad (14)$$

where LHV represents the lower heating value of the fuel, y is the number of hydrogen atoms present in the fuel and x is the number of carbon atoms present in the fuel.

Representing all elements as mass fractions of their parent compound/mixture as shown.

$$reactants \rightarrow m_f(0.7487C + 0.2513H_4) + m_a(0.2319O_2 + 0.7681N_2) \quad (15)$$

Using combustion arithmetic, we arrive at the equation for the products of the combustion process as shown.

$$products \rightarrow m_f\{(0.062392CO_2) + (0.2513H_2O)\} + m_a\{(0.0072469O_2) + (0.027432N_2)\} - m_f\left(0.062392 + \frac{0.2513}{2}\right)O_2 \quad (16)$$

Then multiplying through each product by their molar mass to convert to mass basis,

$$products \rightarrow m_f\{(2.745248CO_2) + (4.5234H_2O)\} + m_a\{(0.2319O_2) + 0.7681N_2\} - m_f(6.017344)O_2 \quad (17)$$

The total exergy of the system after the combustion process can then be written as

$$E_3 = 2.745248m_f(\Delta h + T_o\Delta s)_{CO_2} + 4.5234m_f(\Delta h + T_o\Delta s)_{H_2O} + (0.2319m_a - 6.017344m_f)(\Delta h + T_o\Delta s)_{O_2} + 0.7681m_a(\Delta h + T_o\Delta s)_{N_2} + E^{CH} \quad (18)$$

where E_{CH} is the chemical exergy given as (Moran *et al.*, 2014)

$$E^{CH} = \sum_i z_i E_i^{CH} + RT_o \sum_i z_i \ln z_i \quad (19)$$

where i is the ith molecule or compound present in the product, z_i is the mass participation or the coefficient of each molecule or compound, R is the gas constant of each product and T_o is the ambient temperature.

5) Turbine Analysis

The net power output is given as (Rahman *et al.*, 2011)

$$W_{net} = W_t - W_c \quad (20)$$

where W_t is the work done by the turbine.

The gas turbine thermal efficiency is the percentage of the total fuel energy input that appears as the net work output of the cycle and is given as

$$\eta_{th} = \frac{W_{net}}{Q_{add}} \times 100\% \quad (21)$$

The heat rate (HR) is a measure used to determine how efficiently a generator uses heat energy. It is given as (Rahman *et al.*, 2011)

$$HR = \frac{3600}{\eta_{th}} \quad (22)$$

where η_{th} is the thermal efficiency of the power plant.

6) Turbine Exergy Analysis

The total exergy at the exit of the turbine section is given as

$$E_4 = 2.745248m_f(\Delta h + T_o\Delta s)_{CO_2} + 4.5234m_f(\Delta h + T_o\Delta s)_{H_2O} + (0.2319m_a - 6.017344m_f)(\Delta h + T_o\Delta s)_{O_2} + 0.7681m_a(\Delta h + T_o\Delta s)_{N_2} \quad (23)$$

Table 1: Exergy equilibrium of each component

Component	E_{in}	E_{out}	E_D	η_e
Compressor	E_1	$E_2 - W_c$	$E_1-(E_2- W_c)$	$\frac{E_2 - W_c}{E_1}$
Combustor	$(Q_{add}+E_{fuel}+E_2)$	E_3	$(Q_{add}+E_{fuel}+E_2) - E_3$	$\frac{E_3}{Q_{add} + E_{fuel}}$
Turbine	E_3	(E_4+W_{net})	$E_3 - (E_4+W_{net})$	$\frac{E_4 + W_{net}}{E_3}$

Table 1 shows the expressions for exergy input, exergy output, exergy destroyed and exergy efficiency of each component of the plant.

IV. RESULTS AND DISCUSSION

A. *Energy-Based Assessment*

Table 2 shows the net energy loss or gain of each component of the gas turbine power plant calculated from collected data from the gas turbine power plant control room.

Table 2: Mean Values of Energy-based Assessment

Compressor WorkDone (kW)	Turbine WorkDone (kW)	Net Power Generated (kW)	Heat Added (kW)	Thermal Efficiency (%)	SFC (kg/kW)	Heat Rate (kJ/kW)
1383.544	6436.1306	5052.586	18031.43	28.0274	0.0016	12845.46

Table 3 shows the comparison between the collected data and design values.

Table 3: Comparison of Calculated with Design Parameters

Design Thermal Efficiency (%)	$(\frac{\eta_{th}}{design\ \eta_{th}})$x 100%	Design Net Power Generated (kW)	$(\frac{P_g}{design\ P_g})$x100 %	Design Heat Rate (kJ/kW)	$(\frac{HR}{design\ HR})$x100%
31.5	88.97587	5670	89.42631	11265	112.3837

In Table 3 $\boldsymbol{(\frac{\eta_{th}}{design\,\eta_{th}})x100\%}$ is the ratio of thermal efficiency to the design value, $\boldsymbol{(\frac{P_g}{design\ P_g})x100\%}$ is the ratio of net power generated to the design value, and $\boldsymbol{(\frac{HR}{design\ HR})x100\%}$ is the ratio of heat rate to the design value.

Figures 2 to 6 are straight-line graphs and are graphical representation of the energy performance assessment done on the gas turbine power plant.

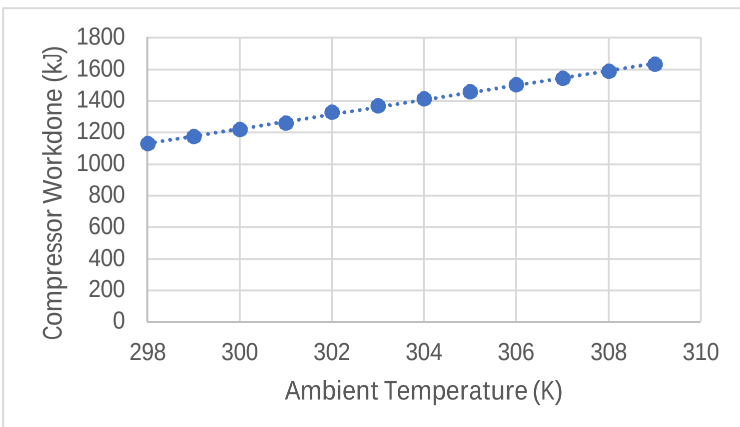

Figure 2: Compressor Work Done vs Ambient Temperature

Figure 2 shows that there is an increment of 46.1176kW in the work done by the compressor per every 1^o rise in the ambient temperature. The direct relationship is governed by the equation

$W_c = 46.1176T_1 - 12612.6894.$ (24)

where W_c is the work-done by the compressor and T_1 is the ambient temperature.

This result agrees with the work of Lebele-Alawa and Jo-Appah (2015) where a one degree increase in ambient temperature was responsible for a 0.3% increase in compressor work.

Figure 3 represents the relationship between the net power generated and ambient temperature. This relationship is governed by the straight-line equation

$$P_g = 33.3888T_1 - 5080.4997 \qquad (25)$$

where P_g is the net power generated by the plant and T_1 is the ambient temperature.

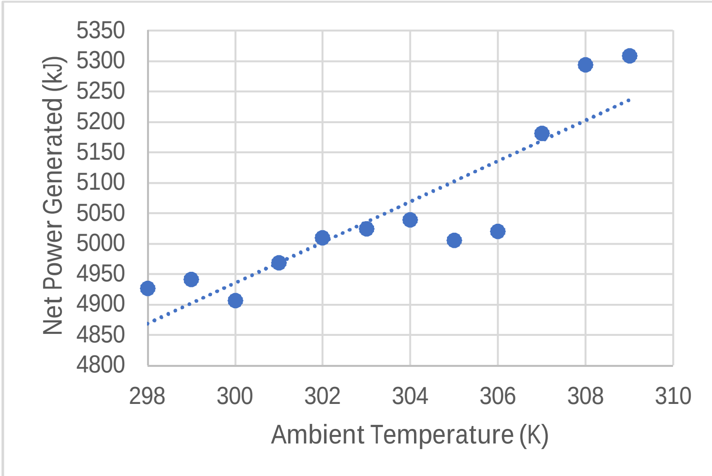

Figure 3: Net Power Generated vs Ambient Temperature

It shows that there is an increase of 33.3888kW in the net power generated for every 1^{o} rise in the ambient temperature.

Figure 4 shows a similar increase in the heat rate of the gas turbine power plant. The relationship between the heat rate and ambient temperature is governed by the straight-line equation

$$HR = 28.7077T_1 + 4132.21021 \qquad (26)$$

where HR is the power plant heat rate and T_1 is the ambient temperature.

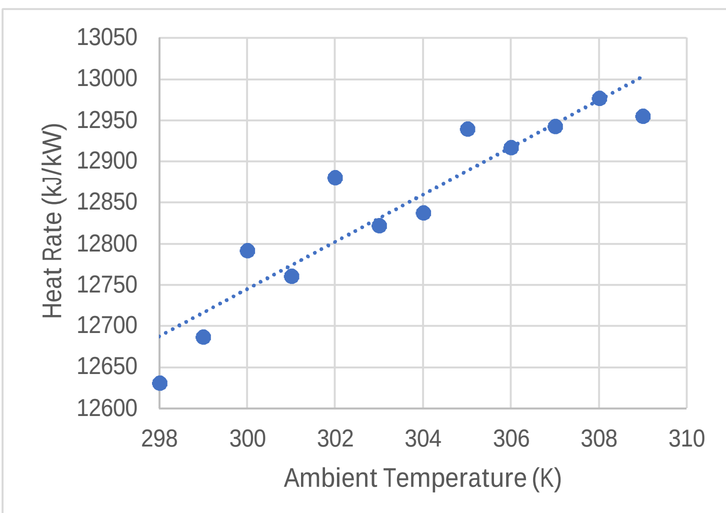

Figure 4: Heat Rate vs Ambient Temperature

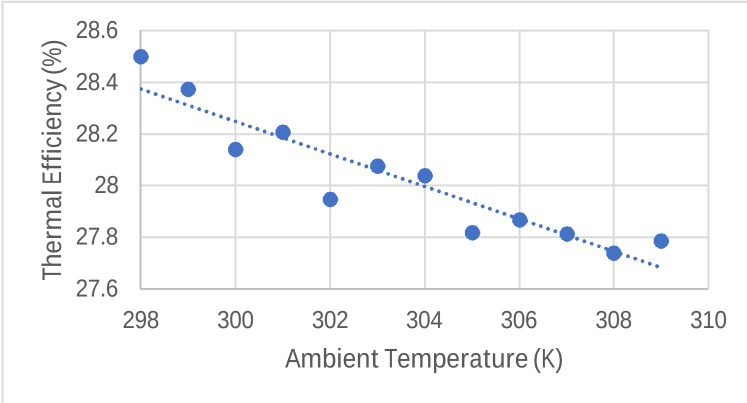

Figure 5: Thermal Efficiency vs Ambient Temperature

Figure 5 shows the relationship between the thermal efficiency of the plant and the ambient temperature. The relationship is governed by the equation

$\eta_{th} = -0.062867T_1 + 47.110269.$ (27)

where η_{th} is the power plant thermal efficiency and T_1 is the ambient temperature.

The resultant straight-line equation shown in Figure 5 reveals that there is a drop of 0.062867% in the thermal efficiency of the plant per degree rise in ambient temperature.

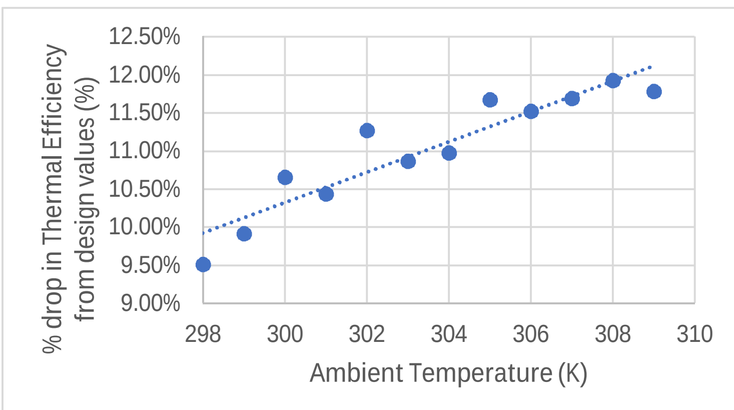

Figure 6: Drop in Thermal Efficiency vs Ambient Temperature

Figure 6 shows that there is an increment of 0.1996% in the drop of thermal efficiency from design values for every 1° rise in ambient temperature. The relationship between the deviation of thermal efficiency from design parameters and ambient temperature is governed by the equation

$\eta_{th}^{\Delta} = 0.19958T_1 - 49.5537521$ (28)

where η_{th}^{Δ} is the deviation of thermal efficiency from design parameters and T_1 is the ambient temperature.

This result agrees with the work of Lebele-Alawa and Jo-Appah (2015) where a one degree increase in ambient temperature was responsible for a 1.49% decrease in thermal efficiency.

B. Exergy-Based Assessment

Table 4 shows the mean exergy values throughout the gas turbine power plant. Exergy is increased through the compression process then hitting peak values after the combustion stage before passing through the turbine stage that reduces the exergy values.

Table 4: Mean Exergy Values at Different Power Plant Stages

E1 (kW)	E2 (kW)	E3 (kW)	E4 (kW)
4074.943	4782.380	13564.619	4850.188

where
E1 = Exergy Entering Compressor (kW)
E2 = Exergy Entering Combustor (kW)
E3 = Exergy Entering Turbine (kW)
E4 = Exergy Exiting in Exhaust Gas (kW)

Table 5: Mean Exergy Destruction and Efficiency Values

Exergy Destroyed in Compressor (kW)	Exergy Destroyed in Combustor (kW)	Exergy Destroyed in Turbine (kW)	Compressor Exergy Efficiency %	Combustor Exergy Efficiency %	Turbine Exergy Efficiency %	Plant Exergy Efficiency %
676.107	9368.507	3661.844	83.409	59.168	72.997	47.428

Table 5 shows mean values for exergy destroyed by the different plant components. It is seen that the compressor destroys the least amount of exergy when compared to the other components, hence it has the highest exergy efficiency followed by the turbine section and then the combustor.

V. CONCLUSION

The first objective of this work which focused on finding the real compressor work done, turbine work done as well as the thermal efficiency of the power plant can be adjudged to be achieved as shown in Table 2. The second objective of this study which is the determination of the amount of exergy created or destroyed by each segment of the gas turbine power plant has also been achieved as it can be concluded that the compressor was the most exergy efficient component, followed by the turbine, and then the combustor as shown in Table 5. Furthermore, the third objective which focused on comparing the calculated performance parameters and the design values has been achieved as both calculated thermal efficiency and net power generated fell short of the design values while the calculated heat rate exceeded the design value as shown in Table 3.
It was recommended that

1) Effort should be made to condition the air that is fed into the compressor. This would have the effect of reducing stress on the compressor, a much colder and denser air would be available for the combustion process and the plant thermal efficiency would be improved.
2) The amount of exergy being wasted as exhaust should be harnessed for other purposes such as regeneration or for a combined cycle system.
3) Research should be made to make the combustion process more efficient as it is the least efficient process in the gas turbine power plant.

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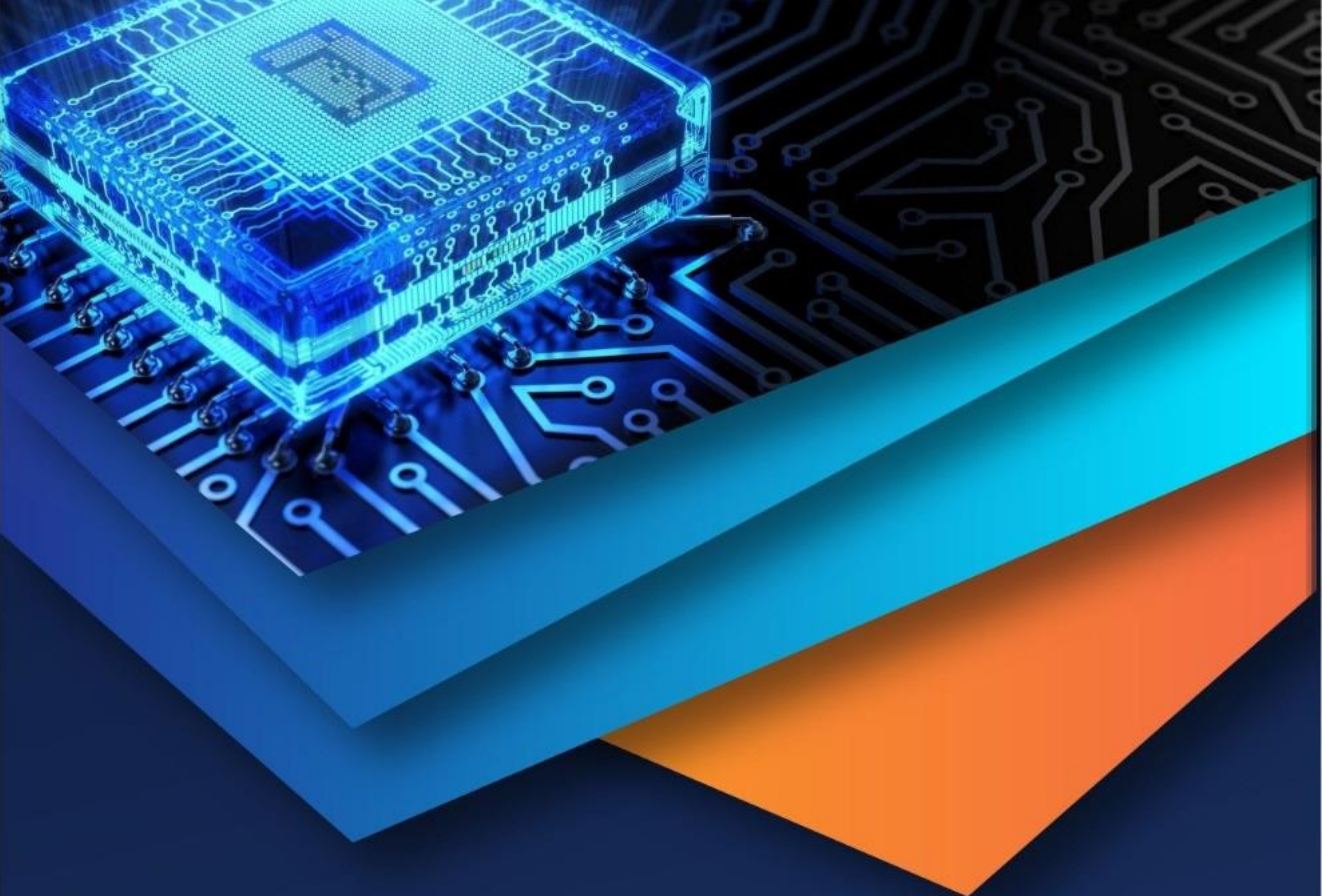

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