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Energy and Exergy Appraisal of a 112.5MW Single Shaft Gas Turbine Power Plant in the Niger Delta, Nigeria

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ABSTRACT
This research presents a thermodynamic appraisal of a 112.5 MW Gas Turbine (GT) Plant through energetic and exergetic analyses utilizing the average operational data for ten (10) years. The components of the GT plant were simplified into control volumes and energy and exergy inflows and outflows, applying the first and second laws of thermodynamics, respectively. The results of the energy analysis showed that a 3.16^{o}C rise in ambient temperature resulted in 10.99% drop in thermal efficiency and 27.92% drop in net power output, indicating a decline in both performance parameters with an increasing ambient temperature. The result of the exergetic analysis revealed that on average, every 1^{o}C increase in ambient temperature resulted in a 0.95% rise in the overall exergetic destruction growth and a 3.22% drop in the overall exergetic efficiency. It was found that a high level of irreversibility in the combustion chamber (CC) is responsible for the low overall exergetic efficiency and the increment in the overall exergy destroyed. By raising the turbine inlet temperature, the high exergy destruction in the CC can be reduced to improve the GT performance. Hence, for performance optimization, measures to reduce the exergetic destruction rate within the combustor should be put in place. The study provides performance-based guides for energy managers and investors for critical decision-making.

KEYWORDS: Energetic Assessment, Exergetic Assessment, Appraisal, Gas Turbine Power Plant, Overall Exergetic Efficiency.

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1. INTRODUCTION
Throughout the history of man, major advances in human development have been accomplished through the improvement in energy production and utilization, accompanied with an increasing trend in the energy consumption (Hart, 2018). The electrical form of energy is one of the key drivers of the development of modern society. Thermal power plants are utilized globally for electricity generation. However, the gas turbine (GT) remains the most widely used power generation technology despite the developments in the utilization of available renewable energy sources (Jianpeng *et al.,* 2021). The gas turbine is a conventional device for converting thermal energy from mechanical energy to electrical energy (Lebele-Alawa & Le-ol, 2015). It is a natural gas power plant that generates electricity by burning natural gas as fuel and operates on the Brayton cycle, which consists of a compressor, a combustion chamber, and a turbine (Le-ol & Aziaka, 2018). Apart from electric power generation, gas turbines are also used variously in other fields such as powering aircraft, ships, trains, and other industrial activities (Jianpeng *et al*., 2021). The choice of the gas turbine power plant among other kinds of power plants is informed by its attractiveness in the power generation field due to its low capital cost to power ratio, high flexibility, high reliability, compactness, easy commissioning and commercial operation, fast starting-accelerating and quick shutdown (Pappas *et al.,* 2012).

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In Nigeria, gas turbine technology is relatively new, but it has become a promising venture which contributes about 80% of the electric power generation to the national grid (Akhator *et al.,* 2019) due to the abundance of natural gas in the Niger Delta region with a proven gas reserve of 206.53 tcf (DPR, 2021). Although most power plants installed in the Niger Delta are designed at 15°C ISO conditions, the environmental conditions showed that the gas turbine plants mostly operate at a temperature range and relative humidity of 28 to 30°C and 80%, respectively (Lebele-Alawa & Jo-Appah, 2015). These deviations drastically affect the performance of the gas turbine installed in the region as it is widely demonstrated in literature that inlet conditions especially ambient temperature, which varies at the different times of the day, greatly impact on gas turbine performance. Hence, there is the need for appraisal of the gas turbine power plants (GTPPs) to evaluate the effect of the variation in GT performance on electricity production.

Conventionally, some of the parameters used to evaluate the merit of an exergy system includes: energy and exergy performance, economic viability and environmental implications (Nkoi *et al.,* 2015). Energy and exergy analyses provide insight into the losses in the various components of the GT by quantity and quality, respectively (Almutairi *et al,* 2016). Energy analysis uses the first law of thermodynamics, applying energy and mass conservation equations to each of the GT components, integrated into a whole system to evaluate performance by quantity. Exergy analysis, on the other hand, accounts for both quantity and quality of energy gained or lost. Exergy admits that energy cannot be created nor destroyed but can degrade in quality (Hart, 2018). It identifies the causes, location, and magnitude (extent) of irreversibility and energy degradation that leads to performance deviation; hence, provides valuable insights for energy policies and economics.

The concept of energy and exergy has severally been applied to energy systems in literature. Osueke *et al.* (2015) used energy and exergy methodologies to analyse the performance of a steam power plant, identifying areas of loss and developing models for more efficient and effective power plant improvements. Adumene *et al.* (2015) conducted an exergy-based analysis of an offshore gas plant and observed that there is a drop in both the thermal and exergy efficiency as the operational load decreases. Adibhatla *et al.* (2014) performed energy and exergy analyses of a thermal power plant under various load conditions. The study revealed that the boiler had the highest rate of exergy destruction in the plant, and there was a significant reduction in the rate of exergy destruction at part-load conditions for the turbine in the case of a sliding pressure operation. Le-ol *et al.* (2018) did a comparative assessment of thermal power systems performance under uncertainty. The result showed that the GT plants perform better at a lower ambient temperature and higher relative humidity with a higher return on investment. Qi and Huang (2022) performed energy and exergy analyses of supercritical/trans-critical CO_2 cycles for a water-infected hydrogen gas turbine. The result showed that the combustion chamber is the component with the maximum exergy loss, accounting for 23.58%.

The present study is aimed at performing the thermodynamic performance appraisal of a 112.5MW GT plant through energetic and exergetic analyses. It entails the simplification of the various components of the gas turbine plant (compressor, combustor, and turbine) into control volumes, indicating energy and exergy inflow and outflow streams, utilizing the first and second laws of thermodynamics. The specific objectives of the study were to:

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i. Perform energy analysis of the GT power plant.
ii. Evaluate exergetic performance of the GT.
iii. Evaluate the effect of ambient temperature on the energetic and exergetic performance of the GT.

The key performance indicators utilized for the energy analysis are thermal efficiency, net power output, specific fuel consumption (SFC) and thermal discharge index (TDI); while for exergy analysis, the exergetic destruction value and the second law efficiency are utilized. The results show the applicability of the energetic and exergetic methods for performance prediction for gas turbine projects in the tropical rainforest. The study provides performance-based guides for energy managers and investors in critical decision making.

2. MATERIALS AND METHODS

The research considers the thermodynamic appraisal of gas turbine plants based on energetic and exergetic analytical assessment. Operational data for ten years were used for the performance analysis. This is followed by the gas turbine system description and model formulation utilizing standard thermodynamics equations considering key performance indicators.

2.1 Description of Experimental Engine

The experimental engine utilized for the analysis is the Sapele II gas turbine power plant (SGTPP) located in Sapele, Delta State. It is one of the National Integrated Power Projects (NIPP) initiated by the Federal Government in 2005 to alleviate the power problems in the country.

The plant has four (4) gas turbine generating units of 112.5MW capacity each, totalling 450MW. It is a simple cycle operation plant of model GE FRAME PG9171E with provision for future conversion into a combined cycle. The GT is a single shaft, three bearing heavy duty industrial unit with a 17 – stage axial flow compressor, can-annular type combustor and a 3- stage axial flow turbine.

2.2 Gas Turbine System Description and Theoretical Model Formulation

The gas turbine power plant analyzed in this study is an open cycle GTPP operating on a dry mode, otherwise known as Brayton cycle and consists of an axial flow air compressor in which air with inlet energy and exergy streams from the atmosphere at ambient state 1 is compressed into the combustor at state 2, as shown in the schematic diagram of Figure 1 and the temperature-entropy diagram of Figure 2. These lead to an exergy flow of flue gases at high temperatures and pressure exiting the combustion chamber into the turbine, driving a generator to produce electricity while the flue gas flow continues to the exhaust at state 4.

2.3 Methods for Energy and Exergy Analysis

2.3.1 Energy Performance Indicators

The indicators used to evaluate the energy performance of the gas turbine plants include:

i. Gas Turbine Net Power Output

The gas turbine net power output $\dot{W}_{n,gT}$ is the combination of the actual compressor work $\dot{W}_{aC}$ and actual turbine work $\dot{W}_{aT}$ given by Equation (1)

$$\dot{W}_{n,gT} = \dot{W}_{aC} + \dot{W}_{aT} \quad (1)$$

ii. Thermal Efficiency of the Gas Turbine System

The gas turbine thermal efficiency η_{thgT} can be evaluated as the ratio of gas turbine work $\dot{W}_{n,gT}$ to heat input in the combustion chamber $\dot{Q}_{cc}$ as in Equation (2)

$$\eta_{thgT} = \frac{W_{n,gT}}{Q_{cc}} \quad (2)$$

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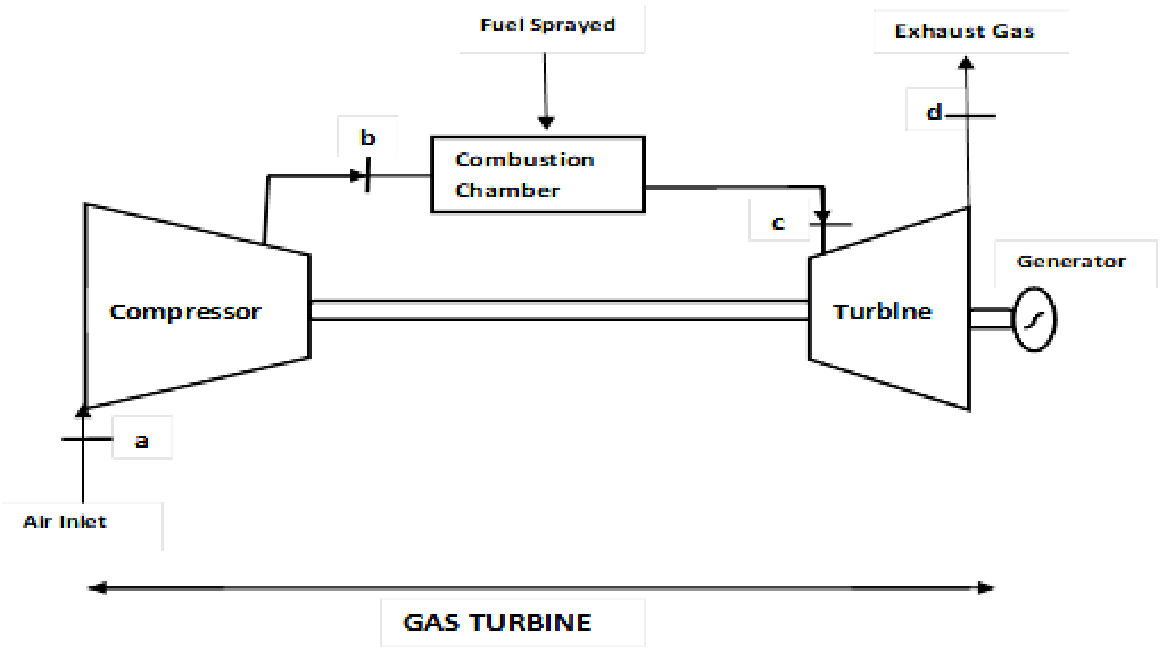

Figure 1: Schematic Diagram of a Simple Gas Turbine Plant

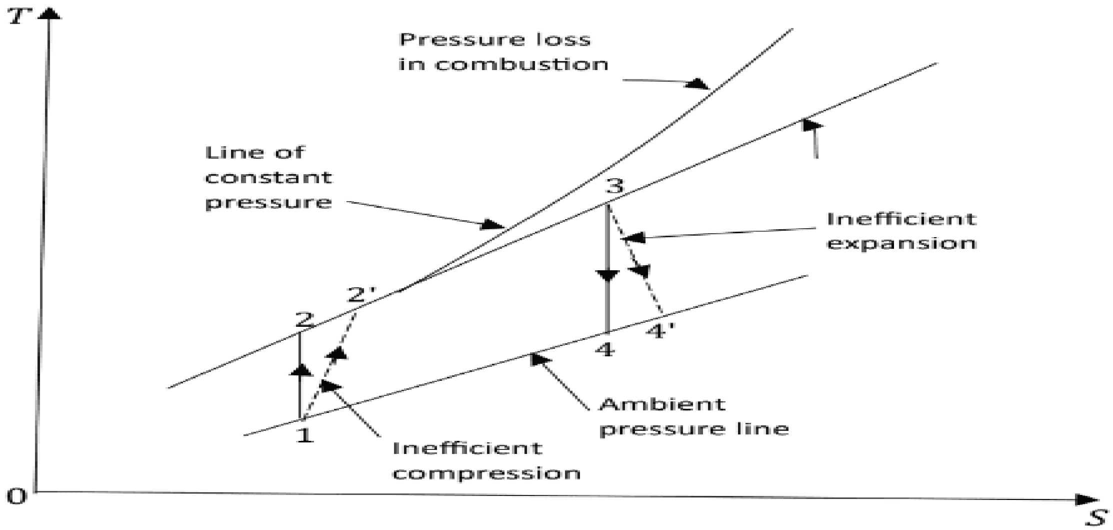

Figure 2: Temperature-Entropy Diagram for a Gas Turbine Cycle

iii. Total Energy Loss in the Gas Turbine System

The total energy loss in the gas turbine system is the summation of the component losses from compressor, combustion chamber and the turbine engine. This is given by Equation (3)

$$\dot{Q}_{TgT,loss} = \dot{Q}_{c,loss} + \dot{Q}_{cc,loss} + \dot{Q}_{gT,loss} \quad (3)$$

where , the energy lost by the compressor due to the difference in input air and compressor-specific inlet temperatures $\dot{Q}_{c,loss}$ is represented as:

$$\dot{Q}_{c_{loss}} = \dot{Q}_0 - \dot{Q}_c \quad (4)$$

The air compressor heat energy input at ambient condition $(\dot{Q}_0)$ and the energy input at

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compressor specific inlet condition ($\dot{Q}_c$) are given respectively by Equations (5) and (6)

$$\dot{Q}_0 = \dot{m}_a c_{pa_c}(T_2 - T_0) \quad (5)$$

$$\dot{Q}_c = \dot{m}_a c_{pa_c}(T_2 - T_1) \quad (6)$$

where, m_a is the mass flow rate of the air via the compressor, T_0, T_1, T_2 (k) are ambient temperature, compressor specific inlet temperature and compressor exit temperature respectively, c_{pa_c} is the specific heat capacity of air given by Equation (6a) as

$$C_{pa_c} = \left\{\frac{1}{T_2 - T_1}\left[A(T_2 - T_1) + B(T_2^2 - T_1^2) + C(T_2^3 - T_1^3) + D(T_2^4 - T_1^4)\right]\right\} \times \left(\frac{1}{M_{\text{air}}}\right) \quad (6a)$$

where M_{air} is the molar mass of air, T_3 as the exit of the combustion chamber, A, B, C and D are constant characteristics of gas obtained from standard thermodynamic tables for selected ideal gas.

The energy loss within the combustion chamber ($\dot{Q}_{cc,loss}$) as utilized by Njoku *et al.* (2018) can be evaluated using Equation (7)

$$\dot{Q}_{cc,loss} = \dot{m}_a c_{pa_{cc}} T_2 + \dot{m}_f LHV = \dot{m}_f c_{pg_{cc}} T_3 \quad (7)$$

where $\dot{m}_f$ is mass flow rate of fuel, T_3 is the gas outlet temperature from the combustion chamber, LHV is the lower heating value, $C_{pg_{cc}}$ and $C_{pa_{cc}}$ are specific heat of air and combustion products respectively which are functions of temperature changes given by Equations (7a) and (7b) as

$$C_{pa_{cc}} = \left\{\frac{1}{T_3 - T_1}\left[A(T_3 - T_1) + B(T_3^2 - T_1^2) + C(T_3^3 - T_1^3) + D(T_3^4 - T_1^4)\right]\right\} \times \left(\frac{1}{M_{\text{air}}}\right) \quad (7a)$$

and

$$C_{pg_{cc}} = C^h_{pf(CO_2, H_2O, N_2O_2)} = \left\{\frac{1}{T_3 - T_{2f}}\left[A(T_3 - T_{2f}) + B(T_3^2 - T_{2f}^2) + C(T_3^3 - T_{2f}^3) + D(T_3^4 - T_{2f}^4)\right]\right\}\left(\frac{m_{f(CO_2, H_2O, N_2O_2)}}{M_{f(CO_2, H_2O, N_2O_2)}}\right) \quad (7b)$$

where $T_{2f} = \frac{T_f + T_2}{2}$ and T_f is the fuel temperature into the combustion chamber, T_2 and T_3 are combustor inlet and outlet temperatures respectively, m_{CO_2}, m_{H_2O}, m_{N_2}, and m_{O_2} are the percentages by mass of the various gaseous products from combustion analysis while M_{CO_2}, M_{H_2O}, M_{N_2}, and M_{O_2} are the molar masses of the combustion products. A, B, C and D are constant characteristics of gas obtained from standard tables for selected ideal gas shown in Cengel and Boles (2006)

The energy loss in turbine engine is given by Equation (8) as

$$\dot{Q}_{gT,loss} = \dot{m}_g c_{pg_T} T_4 \quad (8)$$

where T_4 is the turbine exhaust temperature, and $\dot{m}_g$ is the mass flow rate of flue gas, c_{pg_T} is the total specific heat of the gaseous products (carbon dioxide, water vapour, nitrogen and oxygen) from the turbine engine which can be obtained utilizing Equations (7b) however, utilizing turbine entry temperature (TET), T_3 and turbine exhaust temperature T_4 as the temperature difference.

iii. **Specific Fuel Consumption (SFC)**

The cycle thermal efficiency of a gas turbine is visible in practical terms when it is expressed as a function of fuel flow (Nkoi *et al.,* 2015). The SFC is the ratio of the fuel required to the power output, which is also a measure of the engine efficiency.

The SFC is determined by Equation (9)

$$SFC = \frac{3600\dot{m}_f}{\dot{w}_{n,gT}}\ (kg/kWh) \quad (9)$$

iv. Thermal Discharge Index (TDI)

This is the total amount of thermal energy discharged to the environment per unit generated electrical energy by the power plant (Lebele-Alawa and Asuo, 2011). The thermal discharge index depends strongly on the plant's thermal efficiency. A lower value of the thermal discharge index indicates a more efficient power plant and vice versa. Hence, a lower value of TDI is always desired. It is evaluated by.

$$T_{di} = \frac{1}{\eta_{thgT}} - 1 \quad (10)$$

2.3.2 Exergetic Performance Indicators

Some of the performance indicators used to evaluate the exergy performance of the thermal system include:

i. Exergetic destruction (E_{xD})

The consequence of the non-conservation of exergy in any real process results to a "destruction" term in an exergy balance (E_{xD}) which becomes zero only for a reversible process. Exergy destruction is therefore the potential work lost resulting from system irreversibilities (Oko & Njoku, 2017).

The application of the first and second laws of thermodynamics results in the general form of exergy balance for a control volume and is written according to Ahmadi & Dincer (2011) as

$$\frac{dEx_{cv}}{dt} = \dot{E}_{xCV} = \dot{E}_{x_{\dot{Q}}} - \dot{E}_{x_{\dot{W}}} + \textstyle\sum_i(\dot{m}e_x) - \sum_e(\dot{m}e_x) - E_{xD} \quad (11)$$

where $\dot{E}_{x_Q}$ = exergy rate of heat transfer, $\dot{E}_{x_W}$ = exergy rate of work transfer, $\dot{E}_{xD}$ = lost work due to internal irreversibility

For a control volume at steady state $E_{xCV} = 0$, hence, Equation (11) can be written as

$$\sum\left[\dot{E}_{x_{\dot{Q}}} + \textstyle\sum_i(\dot{m}e_i)\right] = \sum\left[\sum_e(\dot{m}e_x) + \dot{E}_{x_{\dot{W}}}\right] + \dot{E}_{xD} \quad (12)$$

where the subscript i and e represent the inlet and exit flow of the control volume, $\dot{E}_{xD}$ is the exergy destruction rate, $\dot{m}e_x$ is the total exergy rate, $\dot{m}$ is the mass flow rate.

$$e_x = e_{x_{th}} + e_{x_{ch}} \quad (13)$$

where, e_x is the specific exergy of the control volume which is expressed as a combination of the thermo-mechanical (physical) $\dot{E}_{th}$ and chemical exergies $e_{x_{ch}}$ and,

$$\dot{E}_x = \dot{E}_{x_{th}} + \dot{E}_{x_{ch}} \quad (14)$$

where $E_x = \dot{m}e_x$ is the total exergy rate.

The exergy rate of heat and work transfer respectively can be expressed according to Ameri *et al.* (2009) as

$$\dot{E}_{x_{\dot{Q}}} = \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_i \quad (15)$$

and

$$\dot{E}_{x_{\dot{W}}} = \dot{W}_{cv} \quad (16)$$

where $\dot{E}_{x_{\dot{Q}}}$ and $\dot{E}_{x_{\dot{W}}}$ are representation of the exergy rate of heat and work transfer, respectively, across the control surface, $T_i\ (k)$ is the temperature of the ith portion of the control volume where heat transfer Q_i. occurs, $T_0\ (k)$ is the absolute temperature (at dead state or ambient conditions) and $\dot{W}_{cv}$ is the Actual work rate of the control volume.

Therefore, Equation (12) can be re-written as

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$$\dot{E}_{xD} = I_s = \sum_i \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_i + \sum_i \dot{m}e_{x_i} - (\dot{w}_{c_v} + \sum_e \dot{m}e_{x_e})$$
$$= \sum \dot{E}_{x_i} - \sum \dot{E}_{x_e} \quad (17)$$

where I_s is the system irreversibility.

ii. Exergetic Efficiency

The second law efficiency also known as the exergetic efficiency of a thermal power plant is a measure of the ratio of the exergy output from the system $\dot{E}_{x_{out}}$ to the exergy input into the system $\dot{E}_{x_{in}}$ (Midilli & Dincer, 2009). It is expressed as

$$\eta_{ex} = \frac{E_{x_{out}}}{E_{x_{in}}} \times 100\% \quad (18)$$

It is therefore a performance indicator that characterizes the degree of reversibility of the processes of a system. For an entire power plant, as cited in Njoku *et al*. (2020), the overall exergetic efficiency of the power plant which is a measure of the ratio of the shaft work output to the available exergy can be written as

$$\eta_{exPP} = \frac{\dot{W}_{n,gT}}{\Delta\dot{G}_0} = \frac{\dot{W}_{n,gT}}{\dot{E}_{x_f}} = 1 - \frac{\dot{E}_{xD}}{\dot{E}_{x_f}} \quad (19)$$

and

$$\Delta\dot{G}_0 = \dot{m}_f \, \phi \, LHV \quad (20)$$

where η_{exPP} is the overall exergetic efficiency of the entire power; plant, $\dot{W}_{n,gT}$ is gas turbine work and it is given as ϕ = 1.06 for natural gas (methane Ch_4) (Nag, 2001) and ΔG_0 is the exergy rate of fuel, ϕ is the ratio of fuel chemical exergy to the lower heating valve (LHV).

3. RESULTS AND DISCUSSION

The SGTPP with an installed capacity of 112.5MW was analyzed using operational data from the period of 10 years (2012 – 2021). This section discusses the findings, graphical interpretations and predictions from the analyses made.

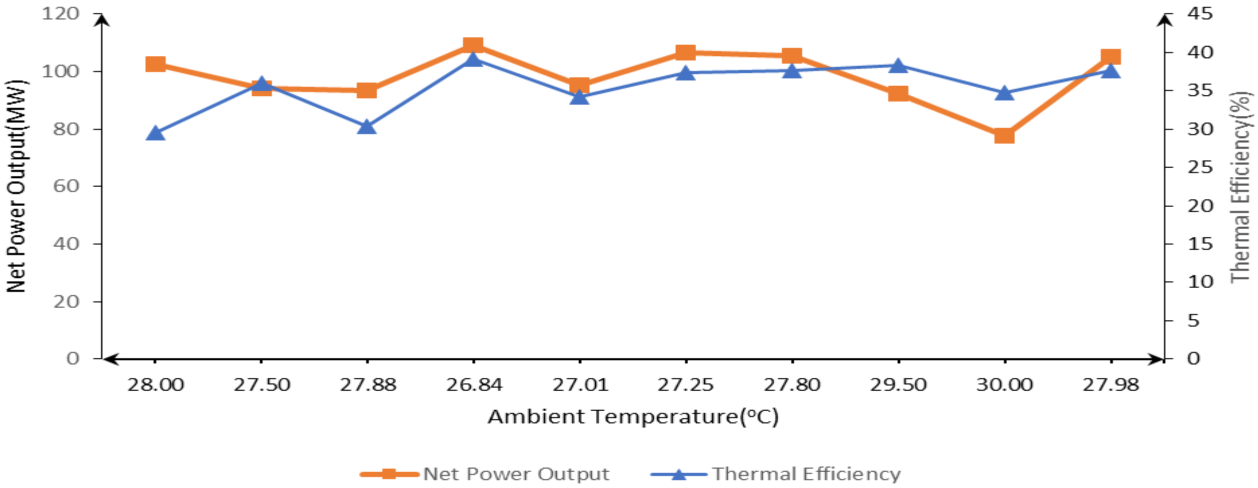

Figure 3: Effect of Ambient Temperature on Thermal Efficiency and Net Power Output

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Figure 3 shows the variation of thermal efficiency and net power output with ambient temperature. The result showed that at the lowest ambient temperature of 26.84°C, the thermal efficiency is maximum having a value of 39.14% with a corresponding maximum net power output of 109.05MW. However, the ambient temperature is maximum (30.00°C) when both thermal efficiency and net power out are minimal with values of 34.82% and 77.64MW, respectively. It is further observed that a 3.16°C rise in ambient temperature resulted in a 10.99% drop in thermal efficiency and 27.92% drop in net power output. Higher thermal efficiency means that more of the energy from the fuel is converted to useful work, hence a decrease in fuel consumption resulting to an increase in the net power output. On the other hand, lower thermal efficiency implies that less of the energy from the fuel is being converted into useful work and therefore a low net power output.

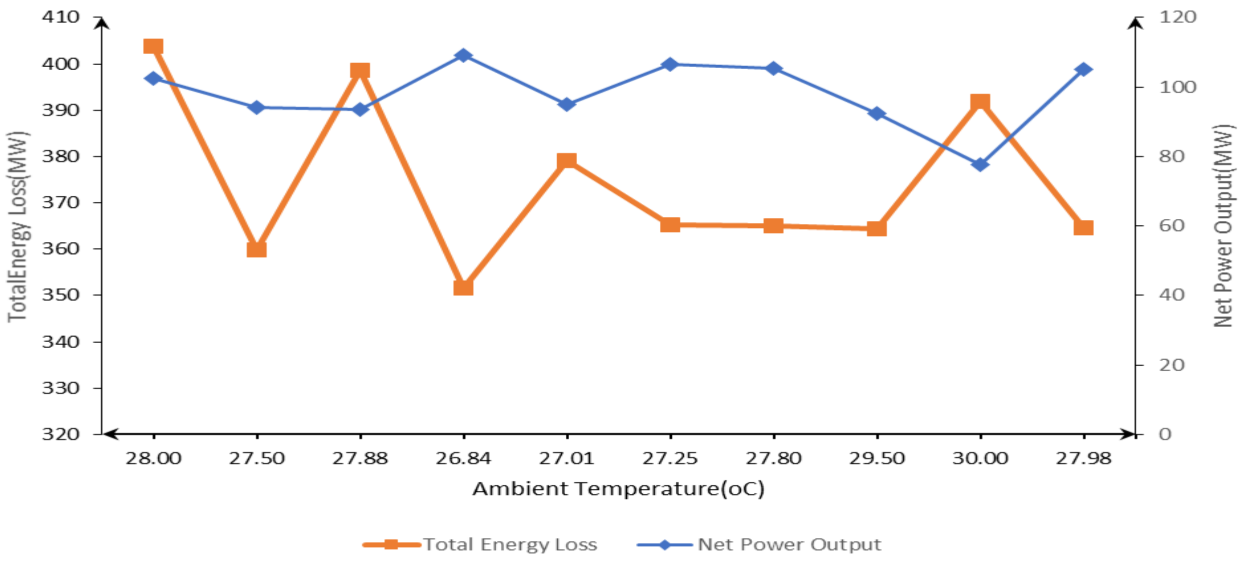

Figure 4: Variation of Total Energy Loss with Ambient Temperature and Net Power Output

Figure 4 shows the effect of energy loss in the compressor section against the ambient temperature and the variation of energy losses with net power outputs. It is observed that at high energy losses, less net power is generated. This is because the energy loss in the compressor subsystem increases with increase in ambient temperature, and at high ambient temperature, the compressor does more work. Hence, the net work rate of the entire gas turbine system drops. This is like the results of Njoku *et al*. (2018).

Figure 5 shows the evolution of the thermal discharge index across the ten (10) years of study. It is observed that between 1.77MW and 3.15MW are annually discharged from the gas turbine plant to the environment per unit MW of electrical energy generated. This indicates an average annual discharge of 2.11MW to the environment from the SGTPP. Also, it is observed that the thermal discharge index (TDI) has its lowest value of 1.77MW at the point of highest thermal efficiency value of 39.1%. This result shows that the TDI is strongly dependent on the plant's thermal efficiency. Hence, to improve the efficiency of the plant, the TDI should be as low as possible (Oyedepo *et al*., 2015).

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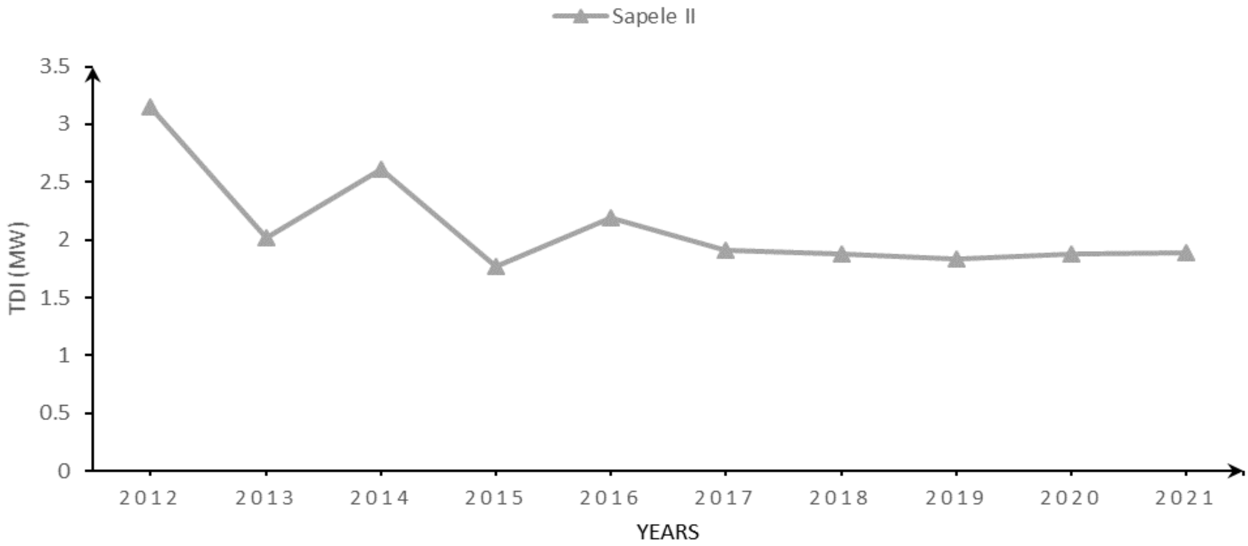

Figure 5: Evolution of Thermal Discharge Index across Year of Stud

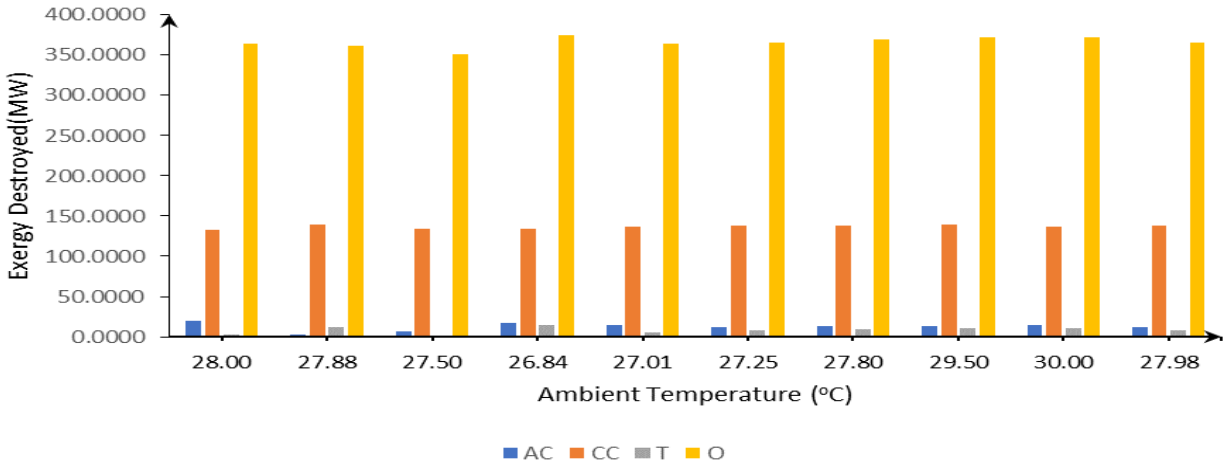

Figure 6: Variation of GT Components Exergetic Destruction with Ambient Temperature

It is observed from Figure 6 that during the increase of ambient temperature from 26.84°C (299.84K) to 30.00°C (303K), the exergetic destruction decreased slightly in the compressor from 16.67MW to 13.86MW, and from 15.01MW to 11.11MW in the turbine subsystem. However, an increment was observed in the combustion chamber from 134.70MW to 137.34MW resulting in a decrease in the overall exergy destroyed from

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374.75MW to 372.00MW. Hence, the result of the comparative analysis of the components shows that the highest exergy destroyed is recorded in the combustion chamber due to a high level of irreversibility.

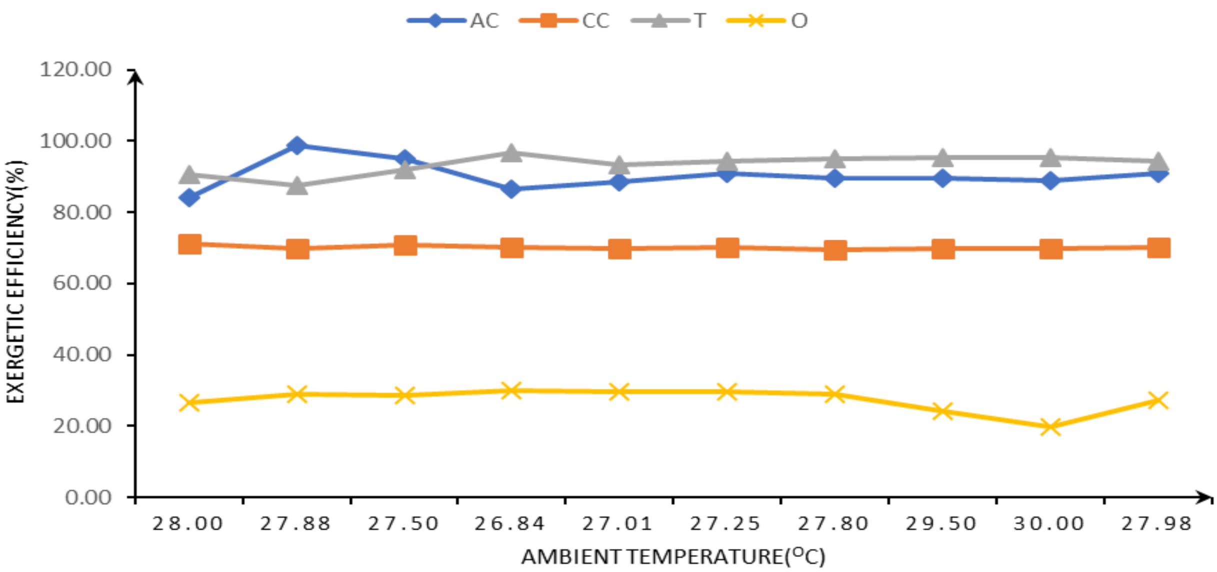

Figure 7: Variation of GT Components Exergetic Efficiency with Ambient Temperature

Figure 7 indicating the effect of ambient temperature on the exergetic efficiency of the GT components, shows that the second law efficiency increases from 86.61% to 88.74% for the compressor with a decrease from 70.20% to 69.72% observed in the combustion chamber. The turbine exergetic efficiency also dropped from 96.78% to 95.49% within the same range of temperature resulting in a decrease in the overall exergetic efficiency from 29.94% to 19.75%. The result further showed that the exergetic efficiency of the compressor is higher than that of the combustion chamber because of the high level of irreversibility in the combustion chamber.

Furthermore, the overall exergetic destruction of the SGTPP at the highest observed ambient temperature of 30.00°C (303K) is 372.00MW at an efficiency of 19.75%, while the overall exergetic destruction at the minimum ambient temperature of 26.84°C (299.84K) is 374.75MW at an efficiency of 29.94% in 2015. This implies that a change in temperature has a direct impact on the exergy performance of the gas turbine because as the air temperature rises, exergy destruction grows and exergy efficiency declines.

4. CONCLUSION

In this study, the energy and exergy viability of the Sapele II power plant was carried out. Utilizing the net power output, thermal efficiency, percentage energy loss, thermal discharge index and specific fuel consumption as the key energy performance indicators, the results of the energetic analysis showed that the average maximum net power output obtained was 103.57MW which represents 92.1% of the installed capacity. The corresponding thermal efficiency was 37.6%, with 5.30KW of electricity produced per kg of fuel consumed, and about 2.00MW of thermal energy being annually discharged to the environment per MW of electricity generated. The mean energy loss indicates an overall average of 374MW for the plant, with the highest proportion of energy losses recorded in the turbine component followed by the combustor and the least in the air compressor. It was observed from the exergy analysis that, on the average, every 1°C increase in ambient temperature resulted in a 0.95% rise in the overall exergetic destruction and a 3.22% decrease in the overall exergetic efficiency of the SGTPP. These indicate that the high exergy destruction in the combustion chamber is responsible for the low overall exergetic efficiency. This can be reduced by increasing the turbine inlet temperature.

The results obtained from this study show the applicability of the energetic and exergetic methods for gas turbine performance prediction. The study provides performance-based guides for energy managers and investors in critical decision making.

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NOMENCLATURE

Symbols	Meaning	Unit
E_{xD}	Exergy destruction	MW
$\dot{E}_{xQ}$	Exergy rate of heat transfer	MW
E_{xT}	Total Exergy of a thermal system	MW
$\dot{E}_{x_{th}}$	Thermo-Mechanical Exergy	MW
$\dot{E}^{ch}_{x_{mix}}$	Chemical exergy for a mixture	MW
E_{xD}	Exergy destruction	MW
$\dot{E}_{xQ}$	Exergy rate of heat transfer	MW
$\dot{E}_{x_W}$	Exergy rate of work transfer	MW
$E_{x_{Ip}}$	Exergetic improvement potential	MW
$\dot{E}^{o}_{x}$	Exergy output rate	MW
$\dot{E}^{T}_{xD}$	Total exergy destroyed	MW
η_{cc}	Combustion Efficiency	
η_{ex}	Exergetic efficiency	
η^{T}_{ex}	Turbine exergetic efficiency	
SGTPP	Sapele II Gas Turbine Power Plant	
CC	Combustion Chamber	
AC	Air Compressor	
GT	Gas Turbine	
EG	Electric Generator	

REFERENCES

Adibhatla, S. & Kaushik, S. C. (2014). Energy and Exergy Analysis of a Super Critical Thermal Power Plant at Various Load Conditions under Constant and Pure Sliding Pressure Operation. *Applied Thermal Engineering*, 73(1), 51 – 65.

Adumene, S., Le-ol, A. K. & Amadi, R. K. C. (2015). Modelling Compressor's Initial Operating Conditions Effect on Turbine Performance in the Tropical Rainforest. *American Journal of Engineering Research*, 4(2), 80-88

Akhator, P. E., Obanor, A. I. & Sadjere, E. G. (2019). Electricity Situation and Potential Development in Nigeria Using Off-Grid Green Energy Solutions. *Journal of Applied Sciences and Environmental Management*, 23(3), 527-537.

Ameri, M., Ahmadi, P. & Hamidi, A. (2009). Energy, Exergy and Exergo-economic Analysis of a Steam Power Plant: A Case Study. *International Journal of Energy Resources*, 33(5), 499-512.

Cengel, Y.A. & Boles, M.A. (2006). *Thermodynamics: An Engineering Approach*, Fifth Edition. McGraw-Hill, New York.

Department of Petroleum Resources (DPR). (2021). *Nigerian Gas Reserves*. Retrieved October 15, 2022, from https://www.dpr.gov.ng/gas-reserves/.

Hart, H. I. (2018). Sustainable Electricity Generation in Nigeria: of Enthalpy, Entropy, Exergy and All That, Not Politics. *An Inaugural Lecture*, Rivers State University, Port Harcourt.

Jianpeng, L., Ziti, L. & Ruoxuan, Y. (2021). Current Status and Prospects of Gas Turbine Technology Application, *Journal of Physics: Conference Series,* 21(1), 120 - 129.

Kanoglu, M., Ibrahim, D. &. Rosen, A. (2007). Understanding Energy and Exergy Efficiencies for Improved Energy Management in Power Plants, *Energy Policy* 35 (7), 3967–3978.

Lebele-Alawa, B. T. & Le-ol, A. K. (2015). Improved Design of a 25 MW Gas Turbine Plant using Combined Cycle Application. *Journal of Power and Energy Engineering*, 3(8), 1-14

Lebele-Alawa, B. T. & Jo-Appah. V. (2015). Thermodynamic Performance Analysis of a Gas Turbine in an Equatorial Rain Forest Environment. *Journal of Power and Energy Engineering*. 3(1), 11-23.

Midilli, A. & Dincer, I. (2009). Development of Some Exergetic Parameters for PEM Fuel Cells for Measuring Environmental Impact and Sustainability. *International Journal Hydrogen Energy*, 34 (9) 3858-3872

Nag, P. K. (2001). *Power Plant Engineering*, Second Edition. McGraw Hill, New Delhi.

Njoku, I. H., Oko, C. O. C. & Ofodu, J. C. (2018). Performance Analysis of a Combined Cycle Power Plant with Simultaneous Cooling of Inlet Air Streams to the Compressor and Condenser. *International Journal of Advanced Thermofluid Research*, 4(1), 2-24

Njoku, I. N., Oko, C. O. C., Ofodu, J. C. & Diemuodeke O. E. (2020). Optimal Thermal Power Plant Selection for a Tropical Region using Multi-Criteria Decision Analysis. *Applied Thermal Engineering*, 179(5), 1-28.

Nkoi, B., Pilidis, P. & Nikolaidis, T. (2015). Techno - Economic Assessment of Large - Scale Aero - Derivative Industrial Gas Turbines Combined - Heat - and - Power. *International Journal of Engineering and Technology*, *5*(4), 225–240.

Oko, C. O. C. & Njoku, I. H. (2017). Performance Analysis of an Integrated Gas-Steam and Organic Fluid Cycle Thermal Power Plant. *Energy,* 122, 431-443

Osueke, C. O., Onokwai, A. O. & Adeoye, A.O. (2015). Energy and Exergy Analysis of a 75MW Steam Power Plant in Sapele (Nigeria). *International Journal of Innovative Research in Advanced Engineering,* 2(6), 2349-2363.

Qi, Y. & Huang, D. (2022). Energy and Exergy Analysis of Supercritical/Trans-critical CO_2 Cycles for Water Injected Hydrogen Gas Turbine, *Energy*, 260(C), 1-16.