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Performance and Exergoeconomic Analysis of a Gas Turbine Power Plant in Port Harcourt, Nigeria

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ABSTRACT

The performance and exergoeconomic analysis of a simple cycle General Electric (GE) gas turbine power plant located at Trans-Amadi, Port Harcourt, was conducted in this study. The data for the analysis were obtained from the power plant log sheets. The study was to assess the exergy destruction rates and the associated cost of exergy destruction across the plant components. The computation and simulation were done with the MATLAB software using EXCEM, SPECO and ECDD analysis methods. Results of performance analysis revealed that the net electric power output ranges from 9.95MW to 10.64MW as against the 25MW installed capacity per unit and thermal efficiency ranging from 17.82% to 18.68%. Calculation showed that the firing temperature of the combustor is 41% of 2525K adiabatic flame temperature which depends on the fuel heating value and the temperature of the burning gases. This low firing temperature shows that a significant amount of heat loss occurs in the combustor. Exergy analysis also revealed that the combustion chamber suffers a high rate of exergy destruction, and that the plant has an overall exergy efficiency of 10.95%. Exergoeconomic analysis revealed that a total cost of $1199.77 is required to generate electricity per hour of which 25% results from exergy destruction. An average exergoeconomic factor of 42% across the plant components shows that more resources are being used up to compensate for the high rate of exergy destruction. This work therefore revealed that compression work and exergy destruction rate increase with increasing ambient temperature leading to a decrease in electric power output. The high rate of exergy destruction is caused by inefficiency of the plant component material type and the high frictional effect encountered by the turbine shaft during rotation and work transfer.

KEYWORDS: Exergy-cost disposition, Flame Temperature, Levelization, Performance Characteristics of a gas turbine, Sustainability index

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1. INTRODUCTION

The gas turbine is a leading option in the energy supply industries in Nigeria as its contribution is about 86% of the electric power generation to the national grid (Akhator *et al*., (2019). They identified that fossil fuels account for over 90% of the country's gross national income (GNI). This shows that gas turbine technology is a promising venture due to natural gas (NG) availability. However, James *et al*. (2018) revealed that a large volume of this gas reserve is constantly being flared in the Niger Delta particularly due to lack of infrastructure to harness crude oil associated gases. The high focus on exportation by gas vendors made Okere (2015) estimate that only 9% of this gas reserve is made available to the thermal power stations in Nigeria. Electricity generation and distribution need urgent attention since the growing population of the country solely rely on electricity for smooth running of businesses and industrial activities.

The effect of ambient conditions on gas turbine combined cycle power plants studied by González-Díaz *et al.* (2017), Almutairi *et al*. (2018), Ait-Ali (1997) and Bouam *et al.* (2008) showed that low ambient temperature favors net power output and cycle efficiencies on full load. González-Díaz *et al.* (2017) therefore established that ambient temperature, pressure ratio and air mass flow through the system influence gas turbine performance. However, they argued that the relative humidity of ambient air has no significant effect on the overall performance of gas turbine power plant. To improve performance, Aref and Pilidis (2012) and Korobitsyn (1998) found that a modification from simple cycle to combined cycle yields a higher cycle efficiency. In this regard, Lebele-Alawa and Le-ol (2015) revealed that a heat

Loss equivalent to 42.46MW from the existing gas turbine power plant located at Omoku,

Rivers State could be converted to 12.9MW of electric power by incorporating a steam bottoming plant combined cycle and a heat recovery steam generator. By this, the flue gas emission to the environment can be minimized. They obtained an overall efficiency of 48.8%, that is, 84% increase from the simple cycle.

Besides, Nkoi and Isaiah (2016) implemented performance simulation of a simple cycle (baseline), intercooled (IC) and intercooled-recuperated (ICR) three-spool large-scale aero-derivative industrial gas turbine derived from turbofan engine. In doing so, design and off-design point performances of the engine models were established, and they found that the IC and ICR cycles exhibit better thermal efficiency than the simple engine. Similarly, heat rate in combustor is reduced in the advanced cycles than the simple engine. It was, however, worthy of note that for large-scale aero-derivative gas turbines having power rating of 100 MW and above, intercooled cycle would consume less fuel than intercooled-recuperated and simple cycles.

An exergoeconomic analysis conducted by Mousafarash and Ameri (2013) on a gas turbine power plant located in Iran shows that the highest exergy destruction occurs in the combustor which is caused by a high irreversibility rate. They argued that this may be due to high temperature difference, higher fuel exergy and chemical reaction during combustion and that increase in ambient temperature decreases the net power output and cycle exergy efficiency. Similar findings on combustor high irreversibility rate have also been published by Adumene *et al.* (2016), Almutairi *et al.* (2018), Mousafarash and Ameri (2013) and Oyedepo *et al.* (2015) in their works on exergyeconomic analysis. Following this, Oyedepo *et al.* (2015) found that the percentage of exergy destruction in the combustion chamber is extremely high and varies between 86% and 95%. In addition, Adumene *et al.* (2016) revealed that about $234.98 per hour economic waste arises from exergy destruction in the exhaust of the turbine outlet. The literatures reviewed in this work have identified, in general, the parameters which influence gas turbine power plant performance and how exergoeconomics is used to determine the cost of exergy destruction. The performance of the Trans-Amadi power plant has not improved over the years according to literature. This work is therefore intended to investigate the plant performance level and the probable causes of exergy destruction in the various plant components and their associated cost burden. To achieve this, EXCEM, SPECO and ECDD analysis methods are used in the present work based on the following objectives.

- **i.** To obtain the input parameters and variables of the plant such as temperature, pressure, and material cost.
- **ii.** To perform an energy analysis based on steady flow energy equation.
- **iii.** To perform an exergy analysis based on the second law of Thermodynamics.
- **iv.** To determine the levelized cost of the plant components.
- **v.** To conduct an exergoeconomic analysis of the plant based on developed economic and exergy models.

2. MATERIALS AND METHODS

The operational data for the performance of the plant were obtained from the Trans-Amadi 4x25MW General Electric (GE) gas turbine located in Port Harcourt, in the Niger Delta region of Southern Nigeria. The gas turbine power plant is an air breathing electric power plant in which a gas turbine (GT) is the prime mover. It is used for the generation of electricity as shown in Fig. 1. The air filter (AF) helps to remove dust particles from the air before the air is fed into the compressor. 1-2 is the air compression process in the compressor (C), 2-3 is the air-fuel combustion process in the combustion chamber (CC) while 3-4 is the expansion process in the gas turbine (GT). The electric generator (EG) converts the rotary energy of the turbine shaft into electricity. The principle of operation is discussed extensively by Oko (2012) and in the literature.

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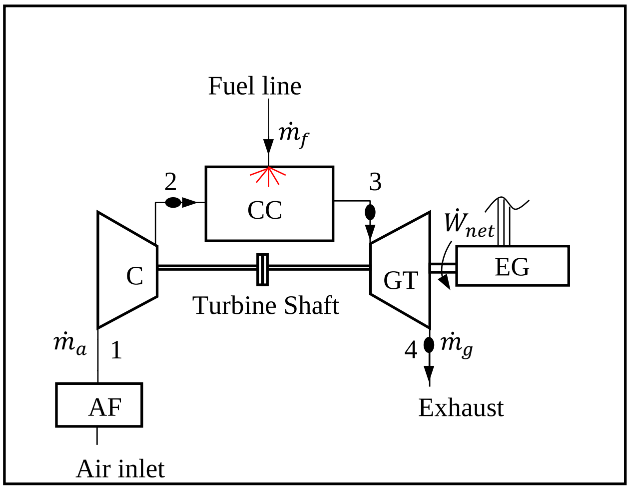

Fig. 1 Schematic diagram of shaft-mounted General Electric gas turbine power plant.

The gas turbine power plant operates on the Brayton power cycle as illustrated in Fig. 2 on T-s diagram. The enclosed area bounded by the cycle shows the magnitude of the system available energy. The actual cycle which is the primary concern of the present work follows the path 1-2-3-4 especially where a pressure drop (ΔP) exists, while the ideal cycle represents processes 1-2′- 3′-4′.

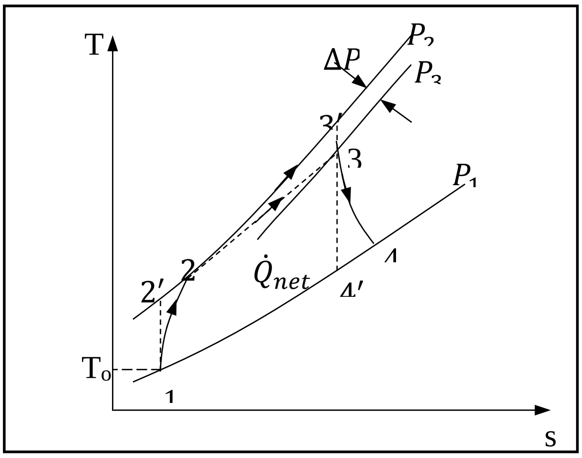

Fig. 2 Brayton power cycle on T-s diagram

Gas turbine performance parameters identified by Gülen (2019) are thus discussed as follows.

2.1 Compressor

Axial flow rotary compressors are found useful in modern gas turbine applications because of the advantage of handling large flows and high-pressure ratios of 10 or more in multistage compressors (Ideriah, 1986). Equation (1) is the steady flow energy equation (SFEE) of the first law of thermodynamics per unit mass flow.

$$Q_s - W = \Delta h + \Delta K_e + \Delta P_e \quad (1)$$

where Q_s(kJ/kg) is the specific heat supplied, W(kJ/kg) is the specific work, Δh is the change in specific enthalpy, ΔK_e is the change in specific kinetic energy and ΔP_e is the change in specific potential energy.

The actual compression work rate, $\dot{W}_{12}$, from (1) with respect to Fig. 2, process 1-2, is given as follows assuming that air behaves as an ideal gas where the heat capacity at constant pressure varies with temperature.

$$\dot{W}_{12} = \dot{m}_a W_{12} \quad (2a)$$
$$W_{12} = [h(T_1) - h(T_2)] \quad (2b)$$

where $\dot{m}_a$ (kg/s) is the mass flow rate of air and W_{12} (kJ/kg) the specific work of the compressor.

2.2 Combustion Chamber

Natural gas comes from dry natural reservoir methane (CH_4) and are used as fuel in gas turbine applications (Rajput, 2013). The balanced stoichiometric chemical equation is given as

$$CH_4 + 2[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 2(3.76N_2) \quad (3)$$

The rate of heat addition in the combustion chamber, process 2-3, is given as:

$$\dot{Q}_s = \dot{m}_g[h(T_3) - h(T_2)] \quad (4)$$

where $\dot{Q}_s$ (MW) is the heat supplied from the air-fuel gas mixture and $\dot{m}_g$ is the mass flow rate of the gas product of combustion, respectively.

2.3 Gas Turbine (GT)

The nozzles convert the high thermal pressure energy of the combustion product into kinetic energy which impinges on the blade causing it to rotate delivering rotational work output which in turn is converted to electricity by directly mounting an electric generator on the rotating shaft

(Oko, 2012). Gas turbine power output as a strong function of air mass flow rate is given as

$$\dot{W}_{34} = \dot{m}_g W_{34} \tag{5a}$$

$$W_{34} = [h(T_3) - h(T_4)] \tag{5b}$$

where W_{34} is the specific work of turbine.
The net power output as a function of the mass flow rate is given as

$$\dot{W}_{net} = \dot{m}_g W_{net} \tag{6}$$

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{7}$$

where W_{net} is the specific net work, and $\dot{m}_f$ is the mass flow rate of fuel.

2.4 Thermal Efficiency and Ideal Air Standard Cycle Efficiency

The plant thermal efficiency is the ratio of the net power output to the total heat supplied due to combustion. It is given as

$$\eta_{th} = \frac{W_{net}}{Q_s} \tag{8}$$

where Q_s is the specific heat supplied.

2.5 Exergy Analysis

Exergy is the maximum work potential which can be extracted from a system as it approaches the equilibrium state of the environment. In every real energy transformation process, exergy is always destroyed as the system proceeds to the dead state. Rao and Parulekar (2007) therefore described exergy as that maximum portion of input thermal energy which can be converted into useful work.

2.5.1 Specific Exergy

The total specific exergy for matter inflow or outflow of a system is the algebraic sum of all the exergy components given as

$$e = e_{ph} + e_{po} + e_{ki} + e_{ch} \tag{9}$$

where e_{ph}, e_{po}, e_{ki}, e_{ch} (kJ/kg) are the physical, potential, kinetic and chemical exergies of the system, respectively. The present work assumed that e_{po}, e_{ki} and e_{ch} are negligible since their magnitudes are exceedingly small compared to the physical exergy. Moran (1989) gave the specific chemical exergy of gaseous hydrocarbon fuel

2.5.2 Physical Exergy

The physical exergy of a steady flow process is expressed as follows when the heat capacity of a gas at constant pressure varies with temperature.

$$e_{ph\_i} = [h^o(T_i) - h^o(T_o)] - T_o\left[s^o(T_i) - s^o(T_o) - R\ ln\left(\frac{P_i}{P_o}\right)\right] \tag{10}$$

where $i = 1,2,3,4$ are the state points of Fig. 1; R (kJ/kgK) is the gas constant; T_o(K) and P_o(N/m^2) are the temperature and pressure corresponding to the state of the environment; and h(kJ/kg) and s(kJ/kgK) are the specific enthalpy and entropy of the gas, respectively.

2.5.3 Exergy efficiency

The overall rational (exergy) efficiency of the plant is expressed as

$$\psi_o = \frac{Total\ exergy\ in\ products}{Total\ exergy\ inputs} = \frac{\dot{W}_{net}}{\dot{E}_f + \dot{E}_1 + \dot{W}_{12}} \tag{11}$$

where $\dot{E}_f$ is the exergy rate of fuel. The compressor inlet exergy $\dot{E}_1$ is zero since the ambient temperature is the same as the dead state as shown in Fig. 3.

2.6 Economic Model Analysis

Non-exergy related costs comprise the capital investment cost and the operation and maintenance cost (Ahmadi & Dincer, 2011; Siahaya, 2009). The present worth of the plant components or capital investment is given as

$$P_w = PEC - S_v\,(1+i)^{-n} \tag{12}$$

where S_v is the component salvage value over its analysis period (n) taken as ten percentage of the purchased equipment cost (PEC); i (%) is the interest rate. This capital investment cost is levelized over the plant analysis period using a capital recovery factor (Oko *et al.*, 2016).

Levelization is the process of converting a non-uniform series of costs of the capital investment into a uniform series. The PEC ($) of gas turbine power plant components were given by Avval *et al.* (2011) and Moran (1989) as a model equation which depends on the flow and the thermodynamic

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properties of the system. The levelized capital cost rate $\dot{Z}_k$ ($/s.) of the plant k[th] component is given as

$$\dot{Z}_k = \frac{P_w(CRF)\varphi}{3600xN} \quad (13a)$$

$$CRF = \frac{i}{1-(1+i)^{-n}} \quad (13b)$$

where φ is the dimensionless component maintenance or service factor, CRF is the capital recovery factor and N (hr./year) is the annual plant operating hours, assuming 24 hours daily operation (Oyedepo *et al.*, 2015).

2.7 Exergoeconomics and EXCEM Analysis

Exergoeconomics combines exergy and economic model analysis to determine the system potential in terms of maximum available work, the cost of electricity generation and the cost of exergy destruction. EXCEM represents the excitation and the corresponding response necessary to give the system desired effect. Since energy and mass are conserved and do not support or give a qualitative assessment of the resources expended, exergy and cost are therefore considered relevant in this section. This is illustrated in Fig. 3 referred to as Exergy-cost disposition diagram (ECDD). Where $\dot{E}$, $\dot{C}$, $\dot{W}$ and $\dot{I}$ are the exergy rate, cost rate, work rate and the irreversibility (exergy destruction) rate, respectively. It shows the 8 potential exergy streams of the SCGT. The exergy rate of the fuel is designated as $\dot{E}_f = \dot{E}_8$. The exergy stream analysis is based on the Grassman diagram. However, the cost of product is greater than the cost of fuel.

2.7.1 Plant Component Exergy Balance

Dincer and Rosen (2007) in their book on exergy methods explained that the exergy of a system always decreases or remains constant. Exergy balance on component basis with respect to Fig. 3 is given as

$$\dot{E}_a = (\dot{E}_{in} - \dot{E}_{out}) - \dot{E}_{con} \quad (14)$$

where $\dot{E}_a$ is the exergy accumulation rate and $\dot{E}_{con}$ is the exergy consumption rate. Assuming that $\dot{E}_a = 0$, the following equations are deduced on component basis. The simple cycle total irreversibility rate/exergy destruction (I_{T_SC}) is based on the Gouy-Stodola relation for steady flow processes.

$$I_{T\_SC} = I_C + I_{CC} + I_{GT} + I_{EG} = T_o S_g \quad (15)$$

where I is the irreversibility rate in the different plant components, T_o is the dead state temperature and S_g is the entropy generation rate.

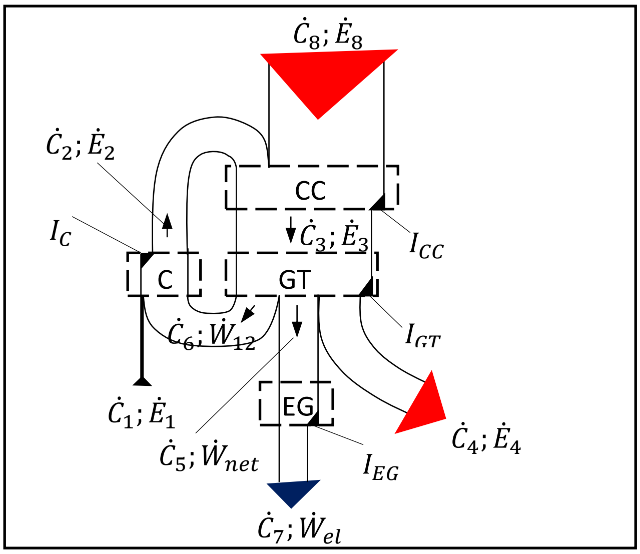

Fig. 3 Simple cycle exergy-cost disposition diagram (ECDD)

2.7.2 Plant Component Cost Balance

Dincer and Rosen (2007) identified that the cost of an exergy stream always increases or remains constant but is never conserved. This is explained by the component cost balance equation based on Fig. 3 as

$$\dot{C}_a = (\dot{C}_{in} - \dot{C}_{out}) + \dot{C}_{gen} \quad (16)$$

where $\dot{C}_a$ is the accumulation cost rate, $\dot{C}_{gen}$ is the total generation cost rate levelized over the life cycle of the system and $\dot{C}_{gen} = \dot{Z}$ +other creation and maintenance costs. The cost rate of exergy $\dot{C}_j$ ($/s) according to Ahmadi and Dincer (2011) across the streams is given as

$$\dot{C}_j = c_j \dot{E}_j \quad (17)$$

where $\dot{Z}$ is the capital cost rate of the plant components, $\dot{C}_j$($/s) is the cost rate of exergy, c_j ($/kJ) is the

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specific cost of exergy and $\dot{E}_j$ (kJ/s) is the exergy rate. j=1, 2, …, 8 streams.

The number of cost equations required for each component is $k + (j - 1)$, where k is the component main balance equation, j is the number of exiting streams, while $j - 1$ forms the number of auxiliary equations to be formed from the P and F principles as defined by Oko *et al.* (2016) and Oyedepo *et al.* (2015). The following cost equations are deduced on component basis from equation (16), assuming that $\dot{C}_a = 0$.

Compressor:

$$\dot{C}_1 + \dot{C}_6 + \dot{Z}_C = \dot{C}_2 \qquad (18)$$

Since the ambient air at compressor inlet is not purchased, its cost is not penalized, that is $\dot{C}_1 = 0$

$$c_2\dot{E}_2 - c_6\dot{W}_{12} = \dot{Z}_C \qquad (19)$$

where $\dot{Z}_C$ is the capital cost rate of the compressor.

Combustion chamber:

The cost rate $\dot{C}_f$ (\$/s) associated with fuel (methane) according to Valero *et al.* (1994) is given as:

$$\dot{C}_f = c_f\dot{m}_f LHV \qquad (20)$$

$$-c_2\dot{E}_2 + c_3\dot{E}_3 = \dot{Z}_{CC} + \dot{C}_8 \qquad (21)$$

where $c_f = c_8$ (\$/kJ) is the specific cost of fuel and $\dot{C}_f = \dot{C}_8$. $\dot{C}_8$ is known as the exergy content of the fuel. LHV is the lower heating value of the fuel and $\dot{Z}_{CC}$ is the capital cost rate of the combustion chamber.

Gas turbine:

$$-c_3\dot{E}_3 + c_4\dot{E}_4 + c_5\dot{W}_{net} + c_6\dot{W}_{12} = \dot{Z}_{GT} \qquad (22)$$

where $\dot{Z}_{GT}$ is the capital cost rate of the gas turbine. The auxiliary equation based on the fuel principle can be written as

$$c_3 - c_4 = 0 \qquad (23)$$

The auxiliary equation based on the product principle can be written as

$$c_5 - c_6 = 0 \qquad (24)$$

Electric generator:

$$-c_5\dot{W}_{net} + c_7\dot{W}_{el} = \dot{Z}_{EG} \qquad (25)$$

where $\dot{Z}_{EG}$ is the capital cost rate of the electric generator.

Rewriting the system of linear equations (19) through (25) in matrix form

$$\begin{bmatrix} \dot{E}_2 & 0 & 0 & 0 & -\dot{W}_{12} & 0 \\ -\dot{E}_2 & \dot{E}_3 & 0 & 0 & 0 & 0 \\ 0 & -\dot{E}_3 & \dot{E}_4 & \dot{W}_{net} & \dot{W}_{12} & 0 \\ 0 & 1 & -1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & -1 & 0 \\ 0 & 0 & 0 & -1 & 0 & \dot{W}_{el} \end{bmatrix} \begin{Bmatrix} c_2 \\ c_3 \\ c_4 \\ c_5 \\ c_6 \\ c_7 \end{Bmatrix} = \begin{Bmatrix} \dot{Z}_C \\ \dot{Z}_{CC} + \dot{C}_8 \\ \dot{Z}_{GT} \\ 0 \\ 0 \\ \dot{Z}_{EG} \end{Bmatrix} \qquad (26)$$

where c_1 and c_8 are known, and the variable specific costs: c_2, c_3, c_4, c_5, c_6 and c_7 are determined using MATLAB software. The total cost of exergy destruction is summed as follows with respect to Fig. 3.

$$\dot{C}_{D\_Total} = \dot{C}_{D\_C} + \dot{C}_{D\_CC} + \dot{C}_{D\_GT} + \dot{C}_{D\_EG} \qquad (27)$$

where $\dot{C}$ is the cost rate of exergy destruction of the different plant components.

2.8 Combustor Adiabatic Flame Temperature

This is the maximum theoretical temperature of the combustor exit gas for a stable flame propagation such that $T_3 < T_{AF}$ for real processes Fig. 4.

Neglecting work transfer, heat transfer and the kinetic and potential energies, equation 1 becomes

$$h_p = h_R \qquad (28)$$

where h_p and h_R are the enthalpies of product and reactant, respectively. Fig. 4 shows the maximum point of the adiabatic flame temperature of the air-fuel mixture. The reactant species are air and fuel at different temperatures mixed in the combustor and burnt rapidly raising the temperature of the mixture to the temperature (T_{AF}) of the gas product. Adiabatic flame temperature is a strong function of the temperature of the burning gases and the heating value of the fuel. Therefore, it is used as a design criterion for assessing air-firing combustion chambers (Liu & Gupta, 2011). By the method of enthalpy of formation, equation (28) becomes

$$\sum_p n_p(\overline{\Delta h}) = \sum_R n_R\left(\overline{h_f^0}\right) + \sum_R n_R(\overline{\Delta h}) - \sum_p n_p\left(\overline{h_f^0}\right) \qquad (29)$$

where n_p, and n_R are the number of moles of product and reactant species. $\overline{\Delta h}$ and $\overline{h_f^0}$ are the enthalpy of combustion and the enthalpy of formation on mole basis.

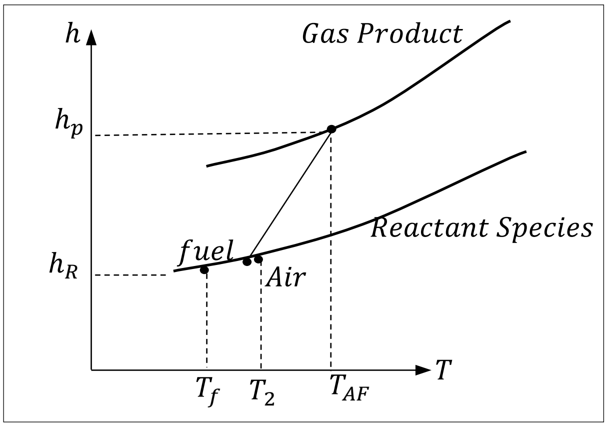

Fig. 4 Enthalpy of combustion vs. temperature

Expanding the RHS of equation (29) with respect to equation (3).

$$\sum f_T = \left(\overline{h_f^0}\right)_{CH_4} + \left[2(\overline{\Delta h})_{O_2} + 7.52(\overline{\Delta h})_{N_2}\right] - \left[\left(\overline{h_f^0}\right)_{CO_2} + 2\left(\overline{h_f^0}\right)_{H_2O}\right] \qquad (30)$$

where $\sum f_T$ (kJ/kmol) is the species sum at their respective temperatures. The enthalpy changes $(\overline{\Delta h})_{O_2}$ and $(\overline{\Delta h})_{N_2}$ (kJ/kmol) are read at compressor exit temperature (T_2) and the standard conditions. Expanding the LHS of equation (29) with respect to equation (3)

$$(\overline{\Delta h})_{CO_2} + 2(\overline{\Delta h})_{H_2O} + 7.52(\overline{\Delta h})_{N_2} = \sum f_T \qquad (31)$$

where the enthalpy change $(\overline{\Delta h})$ and the enthalpy of formation $\left(\overline{h_f^0}\right)$ of the various gases are read from the JANAF thermochemical tables and the result of $\sum f_T$ is compared with those of Table 1, such that $\sum f_i < \sum f_T < \sum f_j$ and, thereafter, linear interpolation is made to determine the adiabatic flame temperature corresponding to it. The linear interpolation technique for determining the adiabatic flame temperature is given by the relation:

$$T_{AF} = T_i + \frac{(\sum f_T - \sum f_i)}{(\sum f_j - \sum f_i)}(T_j - T_i) \qquad (32)$$

where $\sum f_i$ and $\sum f_j$ (kJ/kmol) are the sums of the species on the LHS of equation (31) with initial and final guessed temperatures. The enthalpy of formation of diatomic gases at standard conditions of 25°C and 1atmosphere is zero.

2.9 Total Cost Rate of Electricity Generation

The total cost rate of electricity generation of the plant is the sum of all expenditures that must be paid to produce electricity (Oyedepo *et al*., 2015). This is given as:

$$\dot{C}_{Tot} = \dot{C}_f + \sum \dot{Z}_k + \sum \dot{C}_{D,k} + \dot{C}_{env} \qquad (33)$$

where $\dot{Z}_k$ is the component cost rate, $\dot{C}_{D,k}$ is the component cost rate of exergy destruction and $\dot{C}_{env}$ is the cost rate of environmental impact remediation, respectively. $\dot{C}_{env}$ is neglected as environmental damage is not within the scope of the present work.

2.10 Exergoeconomic Performance Parameters

The criteria for assessing the exergoeconomic performance of thermal power plants is given by Oyedepo *et al*., (2015). They established that

efficient fuel consumption of the plant components could be better described based on depletion number ($D_{p,k}$) and sustainability index of the fuel resource. In the present work, these concepts are narrowed to the plant components.

$$D_{p,k} = \frac{\dot{E}_{D,k}}{\dot{E}_{in}} \qquad (34)$$

The sustainability index ($S_{I,k}$) of the fuel resource at the component level, according to

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Altayib (2011), is the inverse of the depletion number given as

$$S_{I,k} = \frac{1}{D_{p,k}} \quad (35)$$

The component level rational exergy efficiency, from equation 11, is given as

$$\psi_k = \frac{\dot{E}_{in} - \dot{E}_{D,k}}{\dot{E}_{in}} = 1 - \frac{\dot{E}_{D,k}}{\dot{E}_{in}}$$
$$\psi_k = 1 - D_{p,k} \quad (36)$$

where $\dot{E}_{in}$ and $\dot{E}_{D,k}$ are the exergy input and exergy destruction of the k^{th} component, respectively. The exergoeconomic factor, f_k, is given by Giuma *et al.* (2010) and Oko *et al.* (2016). They argued that if the exergoeconomic factor of a component is high, the fixed capital cost must be reduced. However, an extremely low value means a high rate of exergy destruction in that component which means that additional capital must be invested to reduce exergy destruction.

2.11 Model Flow Chart

The model flow chart of the procedure implemented in MATLAB computing environment is shown in Fig. 5. This computational procedure is executed, and the results presented in section 3.

3. RESULTS AND DISCUSSION

The results obtained shows that the thermal efficiency of the SCGT ranges from 17.82 % to 18.68%. The work ratio ranges from 19.86% to 21.36% as against the ideal(maximum) work ratio which ranges from 49.14% to 50.05% while the specific fuel consumption(sfc) was found to be 0.4207 kg/kWh on the average. The maximum theoretical flame temperature T_{AF} of the combustor is computed as 2525K which depends on the fuel heating value and the temperature of the burning gases, as shown in Table 1. Calculations showed that the firing temperature of the combustor is 41% of 2525K adiabatic flame temperature. This firing temperature is clearly seen to be far below the adiabatic flame temperature. This shows that a significant amount of heat loss occurs in the combustor. Fig. 6 shows how monthly weather conditions significantly influence electric power generation through the year.

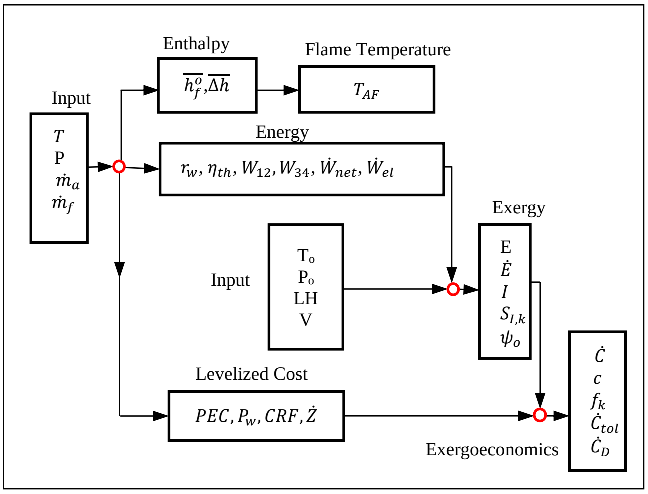

Fig. 5 Computational model flow

Table 1 Property Table for Adiabatic Flame Temperature

Product Species (f)	Initial Guessed Temperature T(K)	Adiabatic Flame Temperature T_{AF} (K)	Final Guessed Temperature T(K)
	T_i = 2200	T_{AF} = 2525	T_j = 2600
$(\Delta h)_{CO_2}$	103460	-	127388
$2(\Delta h)_{H_2O}$	2(83236)	-	2(105220)
$7.52(\Delta h)_{N_2}$	7.52(63316)	-	7.52(78185)
	$\sum f_i$ = 746068.32	$\sum f_T$ = 891946.46	$\sum f_j$ = 925779.20

T_{AF} is calculated from equation 32.

The maximum net electric power output of 10.64MW and thermal efficiency of 18.68% of the plant are recorded in the month of August which has the least ambient temperature of 25.2℃. This shows that low ambient temperature favors gas turbine power plant. Fig. 7 shows that an increase in ambient temperature decreases power output for fixed combustor pressure. This finding is observed to be like that of Mousafarash & Ameri (2013). The coefficient of determination, r^2, shows that a strong linear relationship exists between the electric power output and ambient temperature. The average computed electric power output of the SCGT over the twelve-months analysis period is 10.27MW.

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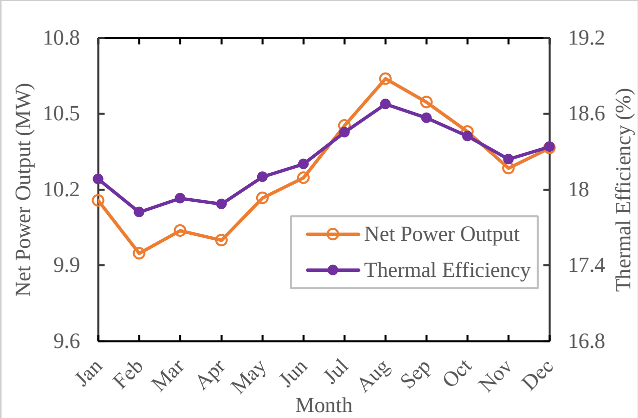

Fig. 6 Monthly power variation

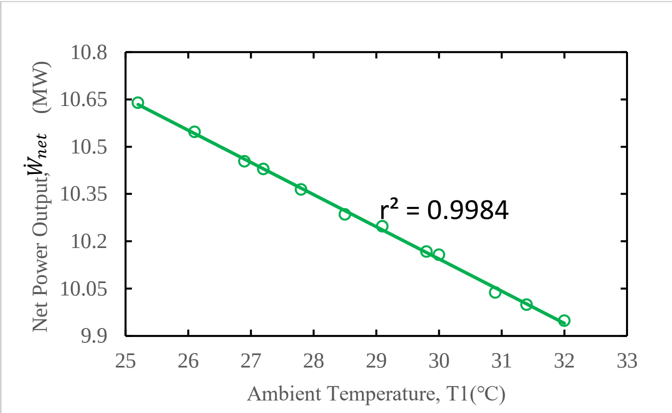

Fig. 7 Power variation with ambient temperature

The decrease in power output is an indication of increased system irreversibility or exergy loss which is due to the high rate of heat interaction in the combustor and the increasing power requirement of the air compressor as ambient temperature increases. Fig.8 shows that gas turbine gross power is highly favored as turbine inlet temperature (TIT) increases just as found by Bouam *et al*. (2008).

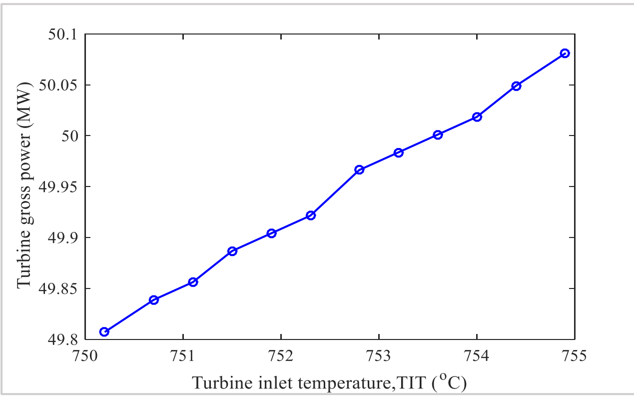

Fig. 8

Turbine gross power vs. TIT

Increase in ambient temperature also increases the compressor work and the fuel consumption rate in the combustor as shown in Fig. 9. This is because, the density of air and the mass flow through the turbine decrease simultaneously.

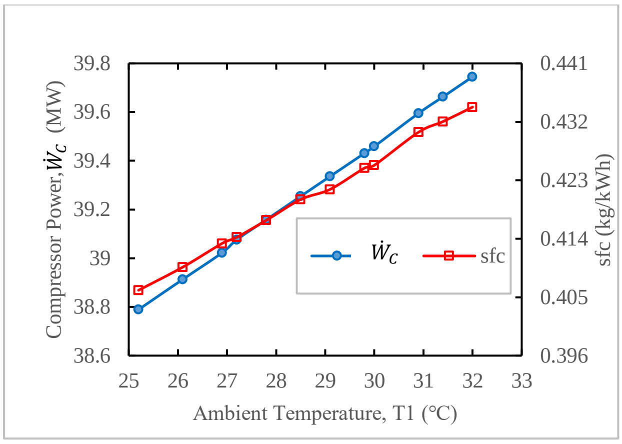

Fig. 9 Effect of ambient temperature on compressor power and sfc

These effects consequently lower the turbine electric power output as illustrated earlier in Fig. 7.

3.1 Result of Exergy and Exergoeconomic Analysis

Exergoeconomic analysis revealed that a total cost of $1199.77 is required to generate electricity per hour of which 25% results from exergy destruction. The total capital investment of the SCGT is $207 per kilowatt. Result from simulation also revealed that the combustion chamber is the component with the highest irreversibility rate (exergy destruction). This is also like the findings of Adumene *et al.* (2016), Almutairi *et al.* (2018) and

Oyedepo *et al.* (2015). The exergoeconomic performance parameters based on the plant components are given in terms of the component exergy efficiency, sustainability index and exergoeconomic factor as presented in Table 2. The higher the magnitude of the plant component

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sustainability index, the more sustainable the fuel resource. Thus, the gas turbine component fuel resource of the simple cycle power plant is most sustainable; that is, fuel stream is efficiently utilized; compared to other components of the plant. However, the exergoeconomic factor, f_k, of the electric generator is extremely low indicating that there exists an extremely high rate of exergy destruction in the component. Thus, optimization of the plant general components is necessary for full performance output of the plant.

Table 2 Exergoeconomic performance parameters

Component	$\dot{C}_{D,k}$ (\$/s)	$\dot{E}_{D,k}$ (MW)	ψ_k (%)	f_k (%)	$D_{p,k}$ (-)	$S_{I,k}$ (-)
C	0.0064	3.67	91.0	50.0	0.09	11.1
CC	0.0066	21.35	76.4	52.3	0.236	4.2
GT	0.005	1.86	97.3	54.8	0.027	37.0
EG	0.065	1.03	90.0	9.4	0.1000	10.0

The overall irreversibility rate or exergy loss of the plant varies linearly with the ambient temperature as shown in Fig. 10.

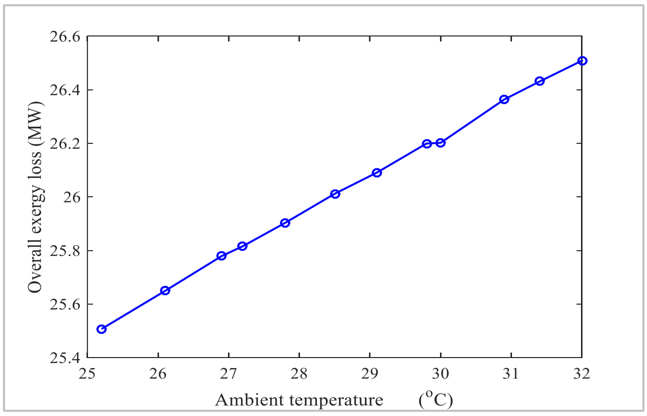

Fig. 10 Exergy loss vs. ambient temperature

This shows that high ambient temperature increases exergy destruction rate and thus does not favor gas turbine performance. Similarly, in Fig. 11 the levelized cost of electricity generation is observed to increase with the rate of exergy destruction (irreversibility rate).

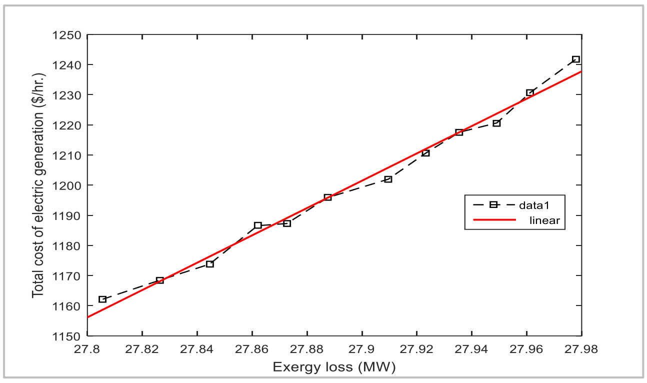

Fig. 11 Cost rate of electricity generation vs. exergy loss

This shows that the cost of electricity generation can be amply reduced if the rate of exergy destruction is minimized. The thermal and exergy efficiencies of the SCGT are also observed to decrease with increasing ambient temperature as shown in Fig. 12. This implies that the performance of the plant can be significantly improved if the compressor inlet air is cooled before suction. This also agrees with the findings of Cyrus and Mee (2000) who argued that evaporative cooling of compressor inlet air is desirable in hot ambient conditions if performance output is to improve.

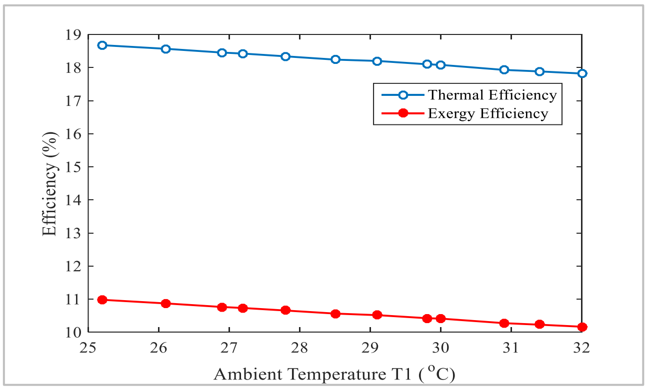

Fig. 12 Effect of ambient temperature on plant efficiency

It is also observed that thermal efficiency is greater than exergy efficiency as energy analysis is a measure of quantity which does not account for losses in the plant components. The rational exergy efficiency varies negatively with temperature just as argued by Almutairi *et al*. (2018) that low ambient temperature favors exergetic efficiency on full load.

4. CONCLUSION

This research was conducted to analyze the overall performance of a 4x25MW simple cycle General Electric (GE) gas turbine power plant located at Trans-Amadi, Port Harcourt, in conjunction with exergoeconomic analysis. The work was to assess the exergy destruction rates and the associated cost of exergy destruction across the plant components. To achieve the first objective of this research, the data for the analysis were gathered from the gas turbine power plant log sheets and the manufacturer's manual over a period of 12 months, 2018. The computation and simulation were done with the MATLAB software on the generated thermodynamic and exergoeconomic model equations using EXCEM, SPECO and ECDD analysis methods.

The second objective was to determine the performance parameters of the power plant. The results obtained from energy analysis revealed that only 9.95MW to 10.64MW of electrical energy and thermal efficiency of 17.82% to 18.68% are obtainable for the twelve-month data points. This shows that the electric power output of the plant has decreased by 59% compared to the 25MW installed capacity per unit. The plant specific fuel consumption obtained is 0.42kg/kWh. The work ratio developed by the turbine was found in the range 19.86% to 21.36% as against the ideal work ratio of 49.57%. An adiabatic flame temperature calculation was also done to show the amount of heat loss in the combustor. The result showed that the firing temperature of the power plant is only 41% of 2525K adiabatic flame temperature. This also reveals that a significant amount of heat loss occurs in the combustor as the firing temperature is extremely low. Objectives one and two are therefore seen to have been achieved as results agree closely with those in the literature.

The third objective was set out to determine the exergetic characteristics of the electric power plant. The key result of exergy analysis therefore revealed that the overall exergy efficiency of the plant is 10.95% and that the combustion chamber suffers a high rate of exergy destruction due to high irreversibility rate and heat loss which may be due to the high temperature difference of the burning gases and material grade. With these findings, objective three is also seen to have been achieved.

Objectives four and five were set out to determine the levelized cost of electricity and the exergoeconomic characteristics. Results revealed that the total cost of capital investment, total levelized cost of exergy destruction and the total levelized cost of electricity generation of the SCGT are \$207 per kilowatt, \$299.45 per hour and \$1199.77 per hour, respectively. The exergoeconomic factor was found to be 42% average across the plant components which also revealed that the plant components suffer high rate of exergy destruction. This research therefore revealed that compression work and exergy destruction rate increased with increasing ambient temperature which consequently leads to decrease in electric power output. The high rate of exergy destruction may also be attributed to plant component material type and the high frictional effect encountered by the turbine shaft during rotation and work transfer. Therefore, the aim of this study titled Performance and Exergoeconomic Analysis of a Gas Turbine Power Plant in Port Harcourt, could be said to have been achieved.

REFERENCES

Adumene, S., Le-ol, A.K., & Lebele-Alawa, B.T. (2016). Performance and economic analysis of gas turbine subsystems for power generation in the Niger Delta. *International Journal of Engineering and Technologies*, *9*(12), 29-41.

Ahmadi, P., & Dincer., I. (2011). Thermodynamic analysis and thermoeconomic optimization of a dual pressure combined cycle power plant with a supplementary firing unit. *Energy Conversion and Management*, *52*(5), 2296–2308.

Ait-Ali, A. (1997). Optimum power boosting of gas turbine cycle with compressor inlet air refrigeration for gas turbine. *Journal of Engineering Gas Turbines Power*, *119*(1), 124-130.

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.

Almutairi, A., Alhajeri, H., Alenezi, A., Zedan, M., & Al-asousi, M. (2018). Exergoeconomic analysis for a two-shaft industrial gas turbine engine. Mechanical Power and Refrigeration Technology Department, *Lodz University of Technology*, *22*(2), 379–395.

Altayib, K. (2011). Energy, exergy and exergoeconomic analysis of gas-based systems (M.Sc. thesis). University of Ontario Institute of Technology.

Aref, P., & Pilidis, P. (2012). Development of framework for thermos-economic optimization of simple and combined gas turbine cycles (PhD thesis). Canfield University, School of Engineering.

Avval, H.B., Ahmadi, A., Ghaffarizadeh, A., & Saidi, M.H. (2011). Thermoeconomic environmental multiobjective optimization of a gas turbine power plant with preheater using evolutionary algorithm. *International Journal. of Energy Research*, *35*(5), 389–403.

Bouam, A., Aissani, S., & Kadi, F. (2008). Gas turbine performance improvement using steam injection in the combustion chamber under Sahara conditions. *Oil and Gas, Science and Technology Journal*, *63*(2), 251-261.

Cyrus, B., & Mee, T. (2000). Inlet forging of gas turbine engine's part B, practical considerations, control, and O&M aspect. *ASME turbo expo proceedings*. May 8-11, Munich.

Dincer, I., & Rosen, M.A. (2007). *Exergy, energy, environment, and sustainable development* (1st ed.) [e-book]. Retrieved from https://www.elsevier.com/books/exergy/dincer/978-0-08-044529-8

Giuma, M.F., Fathi, A.M, & Saleh, M.A. (2010). Exergoeconomic analysis for unit Gt14 of south Tripoli gas turbine power plant. *Jordan Journal of Mechanical and Industrial Engineering*, *4*(4), 507-516.

González-Díaz, A., Alcaráz-Calderón, A.M., González-Díaz, M.O, Méndez-Aranda, A., Lucquiaud, M., & González-Santaló, J.M. (2017). Effect of the ambient conditions on gas turbine combined cycle power plants with post-combustion CO_2 capture. *Energy*, *134*(9), 221-233.

Gülen, S.C. (2019). *Gas turbines for electric power generation.* Cambridge University Press. Doi:10.1017/9781108241625

Ideriah, F.J.K. (1986). *An introduction to fluid machinery* [Unpublished manuscript]. Department of Mechanical Engineering, Rivers State University of Science and Technology, Port Harcourt:

James, R.V., Hart, H.I., & Nkoi, B. (2018). Thermodynamic and environmental assessment of gas flaring in the Niger delta region of Nigeria. *Journal of Power and Energy Engineering, 6(12)*, 39-56.

Korobitsyn, M.A. (1998). New and advanced energy conversion technologies: Analysis of cogeneration, combined and integrated cycles (PhD thesis). Laboratory of Thermal Engineering of the University of Twente and Netherlands Energy Research Foundation ECN.

Lebele-Alawa, B.T., & Le-ol, A.K. (2015). Improved design of a 25MW gas turbine plant using combined cycle application. *Journal of Power and Energy Engineering*, 3(8), 1-14.

Liu, Y., & Gupta, R. (2011). *Oxy-fuel combustion for power generation and carbon dioxide (CO_2) capture*. (1st ed.). Cambridge: Woodhead Publishing limited.

Moran, M.J. (1989). *Availability analysis: A guide to efficient energy use* (2nd ed.). New York: ASME Press.

Mousafarash, A., & Ameri, M. (2013). Exergy and exergo-economic based analysis of a gas turbine power generation system. *Journal of Power Technologies*, *93*(1), 44-51.

Nkoi, B. & Isaiah, T. E. (2016). Advanced Cycles Large-scale Aero-derivative gas turbine: Performance Comparison. *Journal of Power and EnergyEngineering*.4(5),7-19. http://dx.doi.org/10.4236/jpee.2016.45002.

Okere, R. (2015, July 28). Nigeria's untapped gas reserves and declining crude oil fortunes. *Guardian*. Retrieved February 5, 2020, from https://guardian.ng/energy/nigerias-untapped-gas-reserves-and-declining-crude-oil-fortunes/

Oko, C.O.C. (2012). *Engineering thermodynamics: An algorithmic approach* (2nd ed.). Rivers State, Port Harcourt: University of Port Harcourt Press.

Oko, C.O.C., Deebom, M.M., & Diemuodeke, E.O. (2016). Exergoeconomic analysis of cascaded organic power plant for the Port Harcourt climatic zone, Nigeria. *Cogent Engineering*, *3*(9), 1-23.

Oyedepo, S.O, Fagbenle, R.O., Adefila, S.S., & Alam, M. (2015). Thermoeconomic and thermoenvironomic modeling and analysis of selected gas turbine power plants in Nigeria. *Energy Science Engineering*, *3*(5), 423–442.

Rajput, R.K. (2013). *A textbook of internal combustion engines* (2nd ed.). New Deilhi-110002: Laxmi Publication (P) Ltd.

Rao, S., & Parulekar, B.B. (2007). *Energy technology: Nonconventional, renewable & conventional* (5th ed.). Delhi-6: Romesh Chander Khanna, New Nirman Printing Press.

Siahaya, Y. (2009). Thermoeconomic analysis and optimization of gas turbine power plant. *In Proceedings of the International Conference on Fluid and Thermal Energy Conversion FTEC*, Tongyeong, South Korea, December 7–10.

Valero, A., Lozano, M.A., Serra, L., Tsatsaronis, G., Pisa, J., Frangopoulos, C., & von-Spakovsky, M.R.V (1994). CGAM problem: definition and conventional solution. *Energy 19* (3), 279–286.

NOMENCLATURE

Abbreviations

AF	Air filter
AFT	Adiabatic flame temperature
C	Compressor
CC	Combustion chamber
ECDD	Exergy cost disposition diagram
EG	Electric generator
EXCEM	Exergy, cost, energy, and mass
GT	Gas turbine
SCGT	Simple cycle gas turbine
SPECO	Specific cost of exergy

Symbols	**Description**
h_f^o	Enthalpy of formation (kJ/kmol)
$\dot{Q}_s$	Heat rate (MW)
$\dot{W}_{net}$	Work rate (MW)
$\dot{m}_a$	Mass flow rate of air (kg/s)
$\dot{m}_f$	Mass flow rate of fuel (kg/s)
$\dot{m}_g$	Mass flow rate of gas (kg/s)
$\dot{C}$	Exergy cost rate (US$/s)
$\dot{C}_D$	Cost of exergy destruction (US$/s)
$D_{p,k}$	Depletion number of the kth component (-)
$\dot{E}$	Exergy rate (kJ/s)
P_o	Dead state pressure (bar)
P_w	Present worth (US$)
$S_{I,k}$	Sustainability index of the kth component (-)
S_g	Entropy generation (kW/K)
S_v	Plant component salvage value (US$)
T_{AF}	Adiabatic flame temperature of the combustor (K)
T_o	Dead state temperature (K)
$\dot{Z}$	Levelized component capital investment (US$/s)
c_f	Specific cost of fuel (US$/MJ)
f_k	Exergoeconomic factor of the kth component (-)
r_P	Pressure ratio (-)
r_{wi}	Ideal work ratio (-)
w_{12}	Compressor specific work (kJ/kg)
w_{34}	Gas turbine specific work (kJ/kg)
$\dot{W}_{el}$	Electric power output (MW)
γ_f	Fuel exergy grade function (-)
η_{th}	Thermal efficiency (%)
ψ_o	Overall rational (exergy) efficiency (%)
c	Specific cost of exergy (US$/kJ)
e	Specific exergy (kJ/kg)
LHV	Lower heating (calorific) value (kJ/kg)
N	Operating hours, hr.
n	Analysis period (months)
P	Pressure (N/m^2)
PEC	Purchased equipment cost (US$)
I	Irreversibility rate (kW)
φ	Plant component maintenance or service factor (-)

Subscripts

in	input
k	Plant component
out	output
Tot	total