Exergy costing analysis and performance evaluation of selected gas turbine power plants

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Abstract: In this study, exergy costing analysis and performance evaluation of selected gas turbine power plants in Nigeria are carried out. The results of exergy analysis confirmed that the combustion chamber is the most exergy destructive component compared to other cycle components. The exergetic efficiency of the plants was found to depend significantly on a change in gas turbine inlet temperature (GTIT). The increase in exergetic efficiency with the increase in turbine inlet temperature is limited by turbine material temperature limit. This was observed from the plant efficiency defect curve. As the turbine inlet temperature increases, the plant efficiency defect decreases to minimum value at certain GTIT (1,200 K), after which it increases with GTIT. This shows degradation in performance of gas turbine plant at high turbine inlet temperature. Exergy costing analysis shows that the combustion chamber has the greatest cost of exergy destruction compared to other components. Increasing the GTIT, both the exergy destruction and the cost of exergy destruction of this component are found to decrease. Also, from exergy costing analysis, the unit cost of electricity produced in the power plants varies from cents 1.99/kWh (N3.16/kWh) to cents 5.65/kWh (N8.98/kWh).

Subjects: Engineering Economics; Mechanical Engineering; Power & Energy

Keywords: exergy analysis; economic analysis; gas turbine; exergy cost; levelized cost; F-rule; P-rule

ABOUT THE AUTHOR
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PUBLIC INTEREST STATEMENT
In the last two decades, electricity-generating plants in Nigeria have been operating below their capacity with available capacity barely surpassing half the installed capacity which is short of international standards of over 95% installed capacity. Due to this low availability, other key performance indicators (capacity factor and load factor) have also been relatively low. This study therefore aims at evaluating the performance of selected gas turbine power plants in Nigeria using first and second laws of thermodynamics combined with economic analysis with a view of providing possible ways of improving the performance, thus meeting the international standards. The results of the study imply that increase in gas turbine efficiency can be achieved by improving the performance of the most inefficient component of the system. Also, the study provided a suitable methodology for relatively quick identification of the key items requiring performance improvement in a gas turbine power plant.
1. Introduction

Energy is the keystone of life and prosperity. The continued development and application of energy are essential to the sustainable advancement of society. With the exergy analysis, it is possible to evaluate the performance of energy conversion processes not only on a thermodynamics basis, but also by including economic and environmental aspects and impacts of the studied processes. This comprehensive approach of the energy resources utilization has, as one of the most important features, the identification of sustainable ways of energy resources utilization (Silvio, 2013).

The exergy analysis of thermal–mechanical conversion plants aims to characterize how the fuel exergy is used and destroyed in the energy conversion processes that take place in these plants. The exergy analysis provides means to evaluate the degradation of energy during a process, the entropy generation, and loss of opportunities to do work and thus offer space for improvement of power plant performance. Combine with economic analysis, this method allows evaluation of costs caused by irreversibility which may include the investment and operating cost of each component (Ibrahim bin, Masrul, Mohd Zamri, & Mobd, 2001).

Exergy analysis is based on the first and second laws of thermodynamics, and combines the principles of conservation of energy and non-conservation of entropy. The essence of exergy analysis is primarily for optimization. If properly done, it reveals where in the plant the largest energy wastage occurs and therefore the need for design improvements (Ofodu & Abam, 2002; Rosen, 2009). Hence, exergy is often treated as a measure of economic value (Ray, Ganguly, & Gupta, 2007).

Exergy-based cost analysis aims at determining the costs of products and irreversibilities (exergy destroyed) generated in energy conversion processes, by applying cost partition criteria which are function of the exergy content of every energy flow that takes place in the studied process. In this approach, one studies the cost formation processes by valuing the products according to its exergy content and the destroyed exergy during the energy conversion processes. This combination of energy analysis with economic concepts is called thermoeconomic analysis when monetary costs are used and exergoeconomic analysis when exergy costs are employed (Silvio, 2013).

The needs to evaluate the cost production process in a thermal power plant can be rationally conducted if the exergy of the product of the plant (i.e. electricity generated) is taken as the value basis. This is an interesting application of exergoeconomics concepts to evaluate and allocate the cost of exergy throughout the energy conversion processes, considering costs related to exergy inputs and investment in equipment (Dincer & Rosen, 2003). Exergy is taken as a rational basis for economic cost allocation between the resources and products involved in thermal power plant processes and for the economic evaluation of their thermodynamic imperfections (Querol, Gonzalez-Regueral, & Perez-Benedito, 2013).

Exergy costing analysis is a tool used not only to evaluate the cost of inefficiencies or the costs of individual process streams (including intermediate and final products), but also to improve overall system efficiency and lower life cycle costs of a thermodynamic system (Seyyedi, Ajam, & Farahat, 2010). A complete exergoeconomic analysis consists of (1) an exergetic analysis, (2) an economic analysis, and (3) an exergoeconomic evaluation.

A number of studies on exergy and exergy costing analyses of thermal power plants have been carried out by several researchers (Ameri, Ahmadi, & Hamidi, 2009; Aras & Balli, 2008; Can, Celik, & Dagtekin, 2009; Gorji-Bandpy & Ebrahimian, 2006; Gorji-Bandpy & Goodarzian, 2011; Igbon & Fakorede, 2014; Marzouk & Hanafib, 2013; Mousafarash & Ahmadi, 2014; Mousafarash & Ameri, 2013; Kaviri, Mohd Jafar, Tholudin, & Barzegar Avval, 2011; Sahoo, 2008; Singh & Kaushik, 2014).

Most of the past studies on exergy and economic analyses of gas turbine power plant performance were based on a single gas turbine unit. In the present work, analyses are performed on 11 gas turbine units at three different stations in Nigeria.
The prime objectives of the study are:

- To evaluate the exergy performance of the selected power plants by overall exergy efficiency.
- To identify the most significant source of exergy destruction in the power plants and the location of occurrence.
- To evaluate effect of gas turbine inlet temperature (GTIT) on exergy efficiency of the selected gas turbine plants.
- To evaluate exergoeconomic performance of the selected power plants by analyzing exergetic cost parameters of each component of the power plants.
- To determine the unit cost of electricity (product) in the selected power plants using exergy costing analysis.

2. System description
Gas turbine power plants in Nigeria operate on simple gas turbine engine consisting mainly of a gas turbine coupled to a rotary type air compressor and a combustion chamber which is placed between the compressor and turbine in the fuel circuit. Auxiliaries, such as cooling fan, water pumps, etc., and the generator itself are also driven by the turbine. Other auxiliaries are starting device, lubrication system, duct system, etc. For ease of analysis, the steady state model of simple gas turbine is presented in Figure 1.

3. Methodology
Based on the idea that exergy represents the only rational basis not only for assessing the inefficiencies of thermal power system, but also for assigning costs to irreversibilities in the system, a methodological approach called exergoeconomic analysis is applied to evaluate performance of the selected gas turbine power plants in Nigeria.

3.1. Exergy costing analysis
Exergy costing analysis is an effective tool used to evaluate the cost effectiveness of thermal systems, with the intent of evaluating and enhance the system performance from both economic and exergetic (second law of thermodynamics) point of view. The analysis assists in the understanding of the cost value associated with exergy destroyed in a thermal system, and hence provides energy system’s designers and operators with the information, necessary for operating, maintaining, and evaluating the performance of energy systems (Fellah, Mgherbi, & Aboghres, 2010).

In the exergy costing analysis of energy conversion system, four steps proposed by Tsatsaronis (1993) were followed in this study. The first step is exergy analysis. The second step is evaluation of non-exergy related cost (economic analysis) of each of the plant components. This step provides the monetary costs associated with investment, operation, and maintenance. The third step is the estimation of exergetic costs associated with each flow and finally, the fourth step is the exergoeconomic evaluation of each system component.
3.1.1. Exergy analysis

Exergy can be divided into four distinct components. The two important ones are the physical exergy and chemical exergy (Ahmadi, Dincer, & Rosen, 2011; Ameri et al., 2009). In this study, the other two components which are kinetic exergy and potential exergy are assumed to be negligible, as the changes in them are insignificant.

The following steps of exergy analysis itemized by Demirel (2013) are used in this study:

- Define the system boundary of processes to be analyzed.
- Define all the assumptions and the reference temperature, pressure, and composition.
- Determination of the total exergy losses.
- Determination of the thermodynamic efficiency (exergetic efficiency).
- Use exergy loss profiles to identify the regions performing poorly.
- Identification of improvements and modifications to reduce the cost of energy and operation.

Applying the first and the second laws of thermodynamics, the following exergy balance is obtained:

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

The subscripts $i$, $e$, $j$, and $0$ refer to conditions at inlet and exits of control volume boundaries and reference state. $\dot{E}$, $\dot{Q}$, and $\dot{W}$ are the rates of exergy, heat, and work transfer, respectively, $m$ is the mass flow rate. $T$ is the absolute temperature at inlet or exit of control volume.

Equation (1) can be written as:

$$\dot{E}^\text{tot}_i - \dot{E}^\text{tot}_e - \dot{E}_D = 0$$  \hspace{1cm} (2)

where $\dot{E}^\text{tot}_i$, $\dot{E}^\text{tot}_e$, and $\dot{E}_D$ are the total exergy rates at inlet and exit of control volume and rate of exergy destroyed, respectively.

Equation (2) implies that the exergy change of a system during a process is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of irreversibilities.

The exergy-balance equations and the exergy destroyed during each process and for the whole gas turbine plant are written as follows (Abam & Moses, 2011):

### Air compressor

$$\dot{E}^{\text{WAC}} = (\dot{E}^\text{T}_1 - \dot{E}^\text{T}_1^\text{in}) + (\dot{E}^\text{T}_2 - \dot{E}^\text{T}_2^\text{in}) + T_0(\dot{S}_1 - \dot{S}_2)$$  \hspace{1cm} (3a)

$$\dot{E}^{\text{DAC}} = T_0(\dot{S}_2 - \dot{S}_1) = mT_0\left[c_{p1-2}\ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)\right]$$  \hspace{1cm} (3b)

### Combustion chamber

$$\dot{E}^{\text{CHE}} + (\dot{E}^\text{T}_3 + \dot{E}^\text{T}_5 - \dot{E}^\text{T}_3^\text{in}) + (\dot{E}^\text{p}_2 + \dot{E}^\text{p}_5 - \dot{E}^\text{p}_5^\text{in}) + T_0\left(\dot{S}_3 - \dot{S}_2 + \dot{S}_5 + \frac{\dot{Q}_{\text{CC}}}{T_0}\right) = 0$$  \hspace{1cm} (4a)

$$\dot{E}^{\text{DC}} = T_0\left[\dot{S}_3 - \dot{S}_2 + \dot{S}_5 + \frac{\dot{Q}_{\text{CC}}}{T_0}\right] = mT_0\left\{c_{p2-3}\ln\left(\frac{T_3}{T_2}\right) - R\ln\left(\frac{P_3}{P_2}\right)\right\}$$  \hspace{1cm} (4b)

$$+ c_{p5}\ln\left(\frac{T_5}{T_0}\right) - R\ln\left(\frac{P_5}{P_0}\right) + \frac{c_{p2-3}(T_3 - T_2)}{T_{\text{in CC}}^\text{in}}$$
Gas turbine
\[ E_{WGT} = (E_T^3 - E_T^4) + (E_P^3 - E_P^4) + T_0 (S_3 - S_4) \] (5a)
\[ E_{DGT} = mT_0 \left[ c_p (T_3 - T_4) + R (P_4 / P_3) \right] \] (5b)

From Equations (3) to (5), \( \dot{E}_{WAC} \) and \( \dot{E}_{WGT} \) represent the exergy flow rate of the power output from the air compressor and the gas turbine, respectively. \( \dot{E}_{DAMC}, \dot{E}_{DC}, \) and \( \dot{E}_{DG} \) denote the exergy destroyed in the air compressor, combustion chamber, and gas turbine, respectively. \( \dot{E}_T \) is the thermal component of the exergy stream, \( \dot{E}_P \) is the mechanical component of the exergy stream, \( Q_{CHE} \) represents the heat transfer rate between the combustion chamber and the environment; the term \( \dot{E}_{CHE} \) denotes the rate of chemical exergy flow of fuel in the combustion chamber; \( \dot{S} \) is the entropy transfer rate; \( T_{in CC} \) is the temperature of the source from which the heat is transferred to the working fluid, \( T_0 \) and \( P_0 \) are the pressure and temperature, respectively, at standard state; \( m \) is the mass flow rate of the working fluid; \( R \) is the gas constant; \( c_p \) is the specific heat at constant pressure.

For a control volume at steady state, the exergetic efficiency is
\[ \xi = \frac{\dot{E}_P}{\dot{E}_F} = 1 - \frac{\dot{E}_D + \dot{E}_L}{\dot{E}_F} \] (6)
where the rates at which the fuel is supplied and the product is generated are denoted by \( \dot{E}_f \) and \( \dot{E}_p \), respectively. \( \dot{E}_D \) and \( \dot{E}_L \) denote the rates of exergy destruction and exergy loss, respectively. The exergy rate of product, \( \dot{E}_p \), and exergy rate of fuel, \( \dot{E}_f \), for major components of gas turbine power can be determined using the equations presented by Bejan, Tsatsaronis, and Moran (1996).

The \( i \)th component efficiency defect denoted by \( \delta_i \) is given by Equation (7) (Abam, Ugot, & Igbong, 2011):
\[ \delta_i = \frac{\sum \Delta \dot{E}_{Di}}{\sum \Delta \dot{E}_{xin}} \] (7)
where \( \sum \Delta \dot{E}_{Di} \) and \( \sum \Delta \dot{E}_{xin} \) are the total rate of exergy destruction in the plant and total rate of exergy flow into the plant, respectively.

The overall exergetic efficiency of the entire plant is given as:
\[ \psi_i = \frac{W_{net}}{E_{x fuel}} \] (8)
where \( W_{net} \) and \( E_{x fuel} \) are the network output of the plant and exergy of fuel (natural gas) flowing into the combustion chamber, respectively.

The amount of exergy loss rate per unit power output as important performance criteria is given as:
\[ \zeta = \frac{E_{D Total}}{W_{net}} \] (9)
where \( \zeta \) is the exergetic performance coefficient and \( E_{D Total} \) denotes the total exergy destroyed in the entire plant.

Exergy destruction rate and efficiency equations for the gas turbine power plant components and for the whole cycle are summarized in Table 1.
3.1.2. Estimation of non-exergy related cost

The economics of gas turbine assess the non-exergy related cost; which is the cost of the various components of the system (Igbong & Fakorede, 2014). This cost comprises the cost associated with the investment, operation, maintenance, and fuel costs of gas turbine power plant (Ahmadi & Dincer, 2011; Siahaya, 2009). These monetary values are used in the cost balances to determine cost flow rates (Bejan et al., 1996).

The annualized (levelized) cost method of Moran (1982) is used to estimate the capital cost of system component in this work.

The amortization cost for a particular component may be written as (Kim, Oh, Kwon, & Kwak, 1998):

\[ \text{PW} = \text{PEC} - (\text{SV}) \times \text{PWF} (i, n), \]

where the salvage value (SV) at the end of the nth year is taken as 10% of the initial investment for component (or purchase equipment cost, PEC). The present worth (PW) of the component may be converted to the annualized cost by using the capital recovery factor CRF \((i, n)\) (Gorji-Bandpy & Goodarzian, 2011; Kim et al., 1998), i.e.

\[ \text{C} \ (\$/year) = \text{PW} \times \text{CRF} (i, n), \]

\[ \text{CRF} (i, n) = i/1 - (1 + i)^{-n}, \]

\[ \text{PWF} = (1 + i)^{-n}, \]

where \(i\) is the interest rate and it is taken to be 17% (Gorji-Bandpy & Goodarzian, 2011), \(n\) is the total operating period of the plant in years and was obtained from the selected plants. PEC is the purchased equipment cost and \(C\) is the annualized cost of the component.

Equations for calculating the PEC for the components of the gas turbine power plant are as follows (Barzegar Avval, Ahmadi, Ghaffarizadeh, & Saidi, 2011; Bejan et al., 1996; Gorji-Bandpy, Goodarzian, & Biglari, 2010):

**Air compressor**

\[ \text{PEC}_{ac} = \left[ \frac{71.1 m_a}{0.9 - \eta_{ac}} \right] \left[ \frac{P_2}{P_1} \right] \ln \left[ \frac{P_2}{P_1} \right] \]

\[ \text{Combustion chamber} \]

\[ \text{PEC}_{cc} = \left[ \frac{46.08 m_a}{0.995 - P_3/P_2} \right] \left[ 1 + \exp \left( 0.018T_3 - 26.4 \right) \right] \]

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**Table 1. The exergy destruction rate and exergy efficiency equations for gas turbine**

| Component          | Exergy destruction rate          | Exergy efficiency          |
|--------------------|----------------------------------|----------------------------|
| Compressor         | \( E_{DC} = E_{in} - E_{out} + W_C \) | \( \epsilon = \frac{E_{out} - E_{in}}{E_{out}} \) |
| Combustion chamber | \( E_{DCC} = E_{in} - E_{out} + E_{fuel} \) | \( \epsilon = \frac{E_{fuel}}{E_{out}} \) |
| Gas turbine        | \( E_{DG} = E_{in} - E_{out} - (W_{out} + W_C) \) | \( \epsilon = \frac{W_{out} + W_C}{E_{out}} \) |
| Total exergy destruction rate \( E_D_{Total} \) | \( E_D_{Total} = \sum E_D = E_{DC} + E_{DCC} + E_{DG} \) |
Gas turbine

\[
\text{PEC}_{gt} = \left[ \frac{479.34 \dot{m}_g}{0.92 - \eta_{st}} \right] \ln \left[ \frac{P_3}{P_4} \right] \left[ 1 + \exp \left( 0.036 T_3 - 54.4 \right) \right]
\]  

(14)

Dividing the levelized cost by annual operating hours, \( N \), we obtain capital cost rate for the \( k \)th component of the plant (Kwon, Kwak, & Oh, 2001):

\[
\hat{Z}_k = \frac{\phi_k \dot{C}_k}{3600 \times N}
\]  

(15)

From Equations (12) to (15), \( \text{PEC}_{ac}, \text{PEC}_{cc}, \) and \( \text{PEC}_{gt} \) represent the purchasing equipment cost for air compressor, combustion chamber, and gas turbine component, respectively; \( \dot{m}_a \) and \( \dot{m}_g \) denote air mass flow rate and mass flow rate of gas product in the plant, respectively; \( \eta_{ac} \) and \( \eta_{st} \) are the isentropic efficiencies of compressor and gas turbine, respectively; \( P_1, P_2, P_3, \) and \( P_4 \) are compressor inlet pressure, compressor outlet pressure, combustor inlet pressure, and gas turbine inlet pressure, respectively; \( T_3 \) and \( T_4 \) represent the combustor inlet temperature and combustor outlet temperature, respectively; \( \phi_k \) is maintenance factor.

The maintenance cost is taken into consideration through the factor \( \phi_k = 1.06 \) for each plant component (Gorji-Bandpy & Goodarzian, 2011; Gorji-Bandpy et al., 2010).

3.1.3. The auxiliary equation rules

According to Lazzaretto and Tsatsaronis (1999), the following two rules for formulating the auxiliary equations are valid, when finding the specific costs of exergy associated with flows is desired. These are:

**F-principle**

The total cost associated with the removal of exergy must be equal to the cost at which the removed exergy was supplied to the same stream in upstream components.

**P-principle**

Each exergy unit is supplied to any stream associated with the product at the same average cost.

3.1.4. Estimation of gas turbine exergetic costs

Exergy costing is usually applied at the plant component (Tsatsaronis & Winhold, 1984). In order to perform exergy costing calculations, gas turbine components (Figure 1) must be combined into suitable control volumes on which exergetic cost balance equation was then applied on an individual basis. The component in each control volume (CV) with their input and output streams are given as follows:

- **CV 1**: Air compressor (AC)—Input streams: 1, 6
  Output stream: 2
- **CV 2**: Combustion chamber (CC)—Input streams: 2, 5
  Output stream: 3
- **CV 3**: Gas turbine (GT)—Input stream: 3
  Output streams: 4, 6, 7

For a component that receives heat transfer and generates power, cost balance equation may be written as follows (Ameri et al., 2009; Barzegar Avval et al., 2011; Gorji-Bandpy & Goodarzian, 2011; Gorji-Bandpy et al., 2010):

\[
\sum (c_e E_e)_k + c_{w,k} W_k = c_{q,k} E_{q,k} + \sum (c_e E_e)_k + \hat{Z}_k
\]  

(16)
\( \dot{C}_j = c_j \dot{E}_j \)  

(17)

The cost–balance equations for all the components of the system construct a set of nonlinear algebraic equations, which was solved for \( \dot{C}_j \) and \( \dot{C}_j \).

The formulations of cost balance for each component and the required auxiliary equations are as follows:

**Air compressor**

\[ \dot{C}_2 = \dot{C}_1 + \dot{C}_6 + \dot{Z}_{ac} \]  

where subscript 6 denotes the power input to the compressor.

**Combustion chamber**

\[ \dot{C}_3 = \dot{C}_2 + \dot{C}_5 + \dot{Z}_{cc} \]  

(19)

**Gas turbine**

\[ \dot{C}_4 + \dot{C}_6 + \dot{C}_7 = \dot{C}_3 + \dot{Z}_{gt} \]  

(20)

The auxiliary equation for gas turbine is given as:

\[ \frac{\dot{C}_3}{\dot{E}_3} = \frac{\dot{C}_4}{\dot{E}_4} \]  

(21)

Additional auxiliary equation is formulated assuming the same unit cost of exergy for the net power exported from the system and power input to the compressor:

\[ \frac{\dot{C}_6}{\dot{W}_{AC}} = \frac{\dot{C}_7}{\dot{W}_n} \]  

(22)

From Equations (16) to (22), \( Z_k \) is capital cost rate of unit \( k \); \( c_i \) and \( c_e \) represent cost per exergy unit at inlet and exit of component \( k \); \( \dot{E}_i \) and \( \dot{E}_e \) are exergy flow rates at inlet and exit of component \( k \), respectively; \( \dot{C}_j \) is monetary flow rate of material stream \( j \); \( \dot{Z}_{ac} \), \( \dot{Z}_{cc} \) and \( \dot{Z}_{gt} \) denote capital cost rates of air compressor, combustion chamber, and gas turbine, respectively; \( \dot{W}_n \) and \( \dot{W}_{AC} \) represent network output and compressor work input, respectively.

The cost rate associated with fuel (methane) is obtained from (Valero et al., 1994):

\[ \dot{C}_f = c_f \dot{m}_f \times \text{LHV} \]  

(23)

where the fuel cost per energy unit (on an LHV basis) is \( c_f = 0.004 \ $/\text{MJ} \) (Valero et al., 1994), \( \dot{m}_f \) is the mass flow rate of fuel and LHV is the lower heating value of fuel.

A zero unit cost is assumed for air entering the air compressor, i.e.

\[ \dot{C}_1 = 0 \]  

(24)

In order to estimate the cost of exergy destruction in each component of the plant, the cost–balance equations were solved for each component. In application of the cost–balance equation (Equation 17), there is usually more than one inlet and outlet streams for some components. In this case, the numbers of unknown cost parameters are higher than the number of cost–balance equations for that component. Auxiliary exergoeconomic equations (Equations 22 and 23) are developed to solve this problem. Implementing Equation (17) for each component together with the auxiliary equations...
forms a system of linear equations as follows (Ahmadi, Barzegar, Ghaffarizadeh, & Saidi, 2010; Ahmadi, Dincer, et al., 2011; Ameri, Ahmadi, & Khanmohammadi, 2008):

\[ \begin{bmatrix} E_k \end{bmatrix} \times [c_k] = [Z_k] \]  \hspace{1cm} \text{(25)}

where \([E_k]\), \([c_k]\), and \([Z_k]\) are the matrix of exergy rate which were obtained in exergy analysis, exergetic cost vector (to be evaluated) and the vector of \(Z_k\) factors (obtained in economic analysis), respectively.

The above set of equations was solved using MATLAB to obtain the cost rate of each line in Figure 1.

### 3.1.5. Exergoeconomic variables for gas turbine components evaluation

In exergoeconomic evaluation of thermal systems, certain quantities play an important role. These are the average cost of fuel \((c_{F,k})\), average unit cost of product \((c_{P,k})\), the cost rate of exergy destruction \((\dot{C}_{D,k})\), relative cost difference \(r_k\) and exergoeconomic factor \(f_k\) (Ahmadi, Ameri, & Hamidi, 2009; Gorji-Bandpy et al., 2010).

Then the average costs per unit of fuel exergy \((c_{F,k})\) and product exergy \((c_{P,k})\) are calculated from (Fellah et al., 2010):

\[ c_{F,k} = \frac{C_{F,k}}{E_{F,k}} \]  \hspace{1cm} \text{(26)}

\[ c_{P,k} = \frac{C_{P,k}}{E_{P,k}} \]  \hspace{1cm} \text{(27)}

The cost rate associated with exergy destruction is estimated as:

\[ \dot{C}_{D,k} = c_{F,k}E_{D,k} \]  \hspace{1cm} \text{(28)}

Relative cost difference \(r_k\) is given as (Moran & Tsatsaronis, 2000):

\[ r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} = \frac{1 - \epsilon_k}{\epsilon_k} + \frac{Z_k}{c_{F,k}E_{P,k}} \]  \hspace{1cm} \text{(29)}

One indicator of exergoeconomic performance is the exergoeconomic factor, \(f_k\). The exergoeconomic factor is defined as (Fellah et al., 2010; Gorji-Bandpy & Goodarzian, 2011):

\[ f_k = \frac{Z_k}{Z_k + \dot{C}_{D,k}} \]  \hspace{1cm} \text{(30)}

### 4. Results and discussion

The average operating data for the selected gas turbine power plants for the period of six years (2005–2010) are presented in Table 2.

#### 4.1. Results of exergy analysis

The exergy flow rates at the inlet and outlet of each component of the plants were evaluated based on the values of measured properties such as pressure, temperature, and mass flow rates at various states. These quantities were used as input data to the computer program (MATLAB) written to perform the simulation of the performance of the components of the gas turbine power plant and the overall plant.
Table 3 presents results of the net exergy flow rates crossing the boundary of each component of the plants, exergy destruction, exergy defect, exergetic performance coefficient, and exergy efficiency of each component of the plants. The two most important performance criteria, exergy efficiency and exergetic performance coefficient ($\xi$), vary from 18.22 to 32.84% and 1.45 to 2.44%, respectively, for the considered plants. Since the condition of good performance is derived from a higher overall exergetic efficiency but a lower exergetic performance coefficient for any thermal system; hence, it can be inferred that AF2, AF1, and DEL4 gas turbine plants have good performance.

The total exergy destruction rates vary from 59.42 to 234.49 MW, AF4 has the highest value and DEL2 has the least value. The total efficiency defects and overall exergetic efficiency vary from 38.64 to 69.32% and 15.66 to 30.72%, respectively. The efficiency defects are higher for AF4 (69.32%) and AF3 (62.94%) than other units.

The exergy analysis results also show that the highest percentage exergy destruction occurs in the combustion chamber (CC) and followed by the air compressor in some plants in the range of 82.61–91.29% and 4.10–8.16%, respectively. Hence, the combustion chamber is the major source of thermodynamic inefficiency in the plants considered due to the irreversibility associated with combustion and the large temperature difference between the air entering the combustion chamber and the flame temperature. These immense losses basically mean that a large amount of energy present in the fuel, with great capacity to generate useful work, is being wasted. The variations in performance of the plants may be ascribed to poor maintenance procedures, faulty components, and discrepancies in operating data.
To illustrate the effect of operating parameters on the second law efficiency of the components of the gas turbine, the AES1 (PB204) plant is considered as a typical case. The simulation of the performance of plant and components was done by varying the air inlet temperature from 290 to 320 K; and the turbine inlet temperature from 1,000 to 1,400 K, respectively. Figure 2 compares the second law efficiencies of the air compressor, combustion chamber, gas turbine, and the overall plant when the ambient temperature increases. The exergy efficiency of the turbine component and the overall exergetic efficiency of plant decreased with increased ambient temperature, whereas the exergy efficiencies of the compressor and turbine increased with increased ambient temperature.

### Table 3. Result of exergy analysis

| Exergy performance indicator                             | PB204 (AES1) | PB209 (AES2) | PB210 (AES3) | GT17 (AF1) | GT18 (AF2) | GT19 (AF3) | GT20 (AF4) | GT9 (DEL1) | GT10 (DEL2) | GT18 (DEL3) | GT20 (DEL4) |
|----------------------------------------------------------|--------------|--------------|--------------|------------|------------|------------|------------|------------|------------|------------|------------|
| Installed rated power (MW)                               | 33.5         | 33.5         | 33.5         | 75.0       | 75.0       | 138.0      | 138.0      | 25.0       | 25.0       | 100.0      | 100.0      |
| Fuel exergy flow rate (MW)                               | 220.53       | 235.23       | 237.68       | 327.96     | 363.28     | 459.15     | 449.06     | 274.85     | 276.78     | 441.20     | 440.24     |
| Exergy destruction rate of AC (MW)                       | 4.69         | 4.98         | 5.64         | 8.62       | 8.09       | 13.14      | 14.80      | 3.14       | 3.63       | 13.36      | 12.48      |
| Exergy destruction of CC (MW)                            | 56.55        | 56.58        | 55.35        | 139.42     | 159.84     | 176.78     | 180.83     | 62.52      | 61.76      | 171.84     | 173.33     |
| Exergy destruction rate of turbine (MW)                  | 0.29         | 0.52         | 0.23         | 5.99       | 1.47       | 9.39       | 14.50      | 0.39       | 0.14       | 0.70       | 1.80       |
| Total exergy destruction rate (MW)                       | 61.54        | 62.09        | 61.23        | 154.02     | 169.40     | 199.31     | 210.13     | 66.04      | 65.53      | 185.91     | 187.61     |
| Exergy destruction of AC (%)                             | 7.62         | 8.03         | 9.21         | 5.59       | 4.78       | 6.60       | 7.04       | 4.75       | 5.54       | 7.19       | 6.65       |
| Exergy destruction of CC (%)                             | 91.90        | 91.13        | 90.39        | 90.51      | 94.36      | 88.70      | 86.05      | 94.67      | 94.25      | 92.43      | 92.39      |
| Exergy destruction rate of turbine (%)                   | 0.48         | 0.84         | 0.41         | 3.89       | 0.87       | 4.71       | 6.90       | 0.58       | 0.21       | 0.38       | 0.96       |
| Efficiency defect of AC (%)                              | 14.01        | 14.83        | 16.83        | 9.15       | 8.43       | 12.46      | 14.03      | 7.79       | 9.05       | 12.52      | 11.69      |
| Efficiency defect of CC (%)                              | 66.11        | 66.26        | 64.63        | 58.31      | 58.05      | 58.35      | 56.20      | 73.45      | 72.43      | 56.11      | 56.97      |
| Efficiency defect of turbine (%)                         | 0.38         | 0.68         | 0.32         | 3.05       | 0.68       | 4.08       | 6.50       | 0.42       | 0.15       | 0.29       | 0.74       |
| Total efficiency defect (%)                              | 80.50        | 81.77        | 81.78        | 70.51      | 67.16      | 74.89      | 76.73      | 81.66      | 81.63      | 68.92      | 69.40      |
| Exergy efficiency of AC (%)                              | 85.99        | 85.17        | 83.17        | 90.85      | 91.57      | 87.54      | 85.97      | 95.21      | 90.95      | 87.48      | 88.31      |
| Exergy efficiency of CC (%)                              | 74.36        | 75.95        | 76.71        | 57.49      | 56.00      | 61.50      | 59.73      | 77.25      | 77.69      | 61.05      | 60.63      |
| Exergy efficiency of turbine (%)                         | 99.62        | 99.32        | 99.67        | 96.86      | 99.32      | 95.75      | 93.09      | 99.57      | 99.85      | 99.71      | 99.25      |
| Overall exergetic efficiency (%)                         | 19.50        | 18.23        | 18.22        | 29.49      | 32.84      | 25.11      | 23.27      | 18.34      | 18.37      | 31.08      | 30.60      |
| Exergetic performance coefficient (ξ)                    | 1.43         | 1.45         | 1.46         | 1.59       | 1.32       | 1.73       | 2.01       | 1.47       | 1.59       | 1.36       | 1.39       |

### Exergy improvement potential

| Air compressor (MW)                                      | 3.94         | 4.14         | 4.71         | 7.83       | 6.88       | 8.59       | 14.95      | 2.89       | 3.30       | 11.69      | 11.02      |
|---------------------------------------------------------|--------------|--------------|--------------|------------|------------|------------|------------|------------|------------|------------|------------|
| Combustion chamber (MW)                                  | 38.48        | 40.33        | 43.82        | 69.49      | 68.52      | 85.81      | 88.86      | 30.47      | 30.21      | 78.48      | 78.67      |
| Turbine (MW)                                             | 0.015        | 0.023        | 0.13         | 0.19       | 0.22       | 1.05       | 2.50       | 0.15       | 0.17       | 1.00       | 0.38       |
| Entire plant (MW)                                        | 119.44       | 147.15       | 159.28       | 98.30      | 89.99      | 124.18     | 59.99      | 54.04      | 56.46      | 159.88     | 143.28     |
The overall exergetic efficiency decreased from 18.53 to 17.26% for ambient temperature range of 290–320 K. It was found that a 5 K rise in ambient temperature resulted in a 1.03% decrease in the overall exergetic efficiency of the plant. The reason for the low overall exergetic efficiency is due to large exergy destruction in the combustion chamber (Kotas, 1995).

The exergetic efficiency (or second law efficiency) of the plant was also found to depend significantly on a change in turbine inlet temperature. Figure 3 shows that the second law efficiency of the plant increases steadily as the turbine inlet temperature increases. The increase in exergetic efficiency with the increase in turbine inlet temperature is limited by turbine material temperature limit. This can be seen from the plant efficiency defect curve. As the turbine inlet temperature increases, the plant efficiency defect decreases to minimum value at certain TIT (1,200 K), after which it increases with TIT. This shows degradation in performance of gas turbine plant at high turbine inlet temperature.

4.2. Results of exergy costing analysis
Solving the linear system of Equations (14–20), the cost rates of the unknown streams of the system are obtained. Tables 4–6 show the results of levelized cost rates and average costs per unit of exergy at various state points in the AES Barges gas turbine system, Afam gas turbine system, and Delta gas
turbine system, respectively. For these systems, the unit cost of electricity computed from the energy costing method for the selected plants varies from $5.23/GJ (cents 1.88/kWh) (N2.99/kWh) to $15.68/GJ (cents 5.65/kWh) (N8.98/kWh) (see Tables 4–6).

The exergoeconomic parameters for each of the components of the plants considered in this study for their actual operating conditions are summarized in Tables 7–9. The parameters include average costs per unit of fuel exergy $C_F$ and product exergy $C_P$, rate of exergy destruction $\dot{E}_D$, cost rate of exergy destruction $\dot{C}_D$, investment, and O &M costs rate $\dot{Z}$ and exergoeconomic factor $f$. In analytical terms, the components with the highest value of $\dot{Z}_k + \dot{C}_D$ are considered the most significant components from an exergoeconomic perspective (Gorji-Bandpy et al., 2010). This provides a means of determining the level of priority a component should be given with respect to the improving of the system.

Table 4. Levelized cost rates and average costs per unit of exergy at various state points in the AES Barges gas turbine system

| State points | AES1 (PB204) | AES2 (PB209) | AES3 (PB210) |
|--------------|--------------|--------------|--------------|
| $\dot{C}$ ($/h$) | $C$ ($$/GJ$) | $c$ ($$/kWh$$) | $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) | $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) |
| 1 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 |
| 2 | 1,356.70 | 11.67 | 0.042011 | 2,001.24 | 15.03 | 0.054108 | 2,098.72 | 15.67 | 0.056422 |
| 3 | 4,001.51 | 10.69 | 0.038468 | 5,167.1 | 13.65 | 0.049153 | 5,517.8 | 13.65 | 0.049128 |
| 4 | 1,117.72 | 10.69 | 0.038468 | 1,459.99 | 13.65 | 0.049153 | 1,825.40 | 13.65 | 0.049128 |
| 5 | 3,080.69 | 3.08 | 0.011090 | 3,491.45 | 3.49 | 0.012569 | 3,628.37 | 3.63 | 0.013062 |
| 6 | 1,536.58 | 10.73 | 0.038617 | 2,001.10 | 13.75 | 0.049489 | 2,098.59 | 14.07 | 0.050655 |

Table 5. Levelized cost rates and average costs per unit of exergy at various state points in the Afam gas turbine system

| State points | AF1 (GT17) | AF2 (GT18) | AF3 (GT19) | AF4 (GT20) |
|--------------|------------|------------|------------|------------|
| $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) | $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) | $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) |
| 1 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 |
| 2 | 3,807.70 | 10.44 | 0.037594 | 3,504.63 | 9.54 | 0.034339 | 5,562.00 | 11.73 | 0.042246 | 7,475.78 | 14.95 | 0.053813 |
| 3 | 9,254.61 | 9.62 | 0.034631 | 8,784.92 | 8.86 | 0.031909 | 14,019.99 | 10.59 | 0.038114 | 16,571.89 | 12.42 | 0.044695 |
| 4 | 2,436.02 | 9.62 | 0.034631 | 2,346.86 | 8.86 | 0.031909 | 4,311.84 | 10.59 | 0.038114 | 5,683.35 | 12.42 | 0.044695 |
| 5 | 4,581.56 | 4.58 | 0.016494 | 4,511.08 | 4.51 | 0.016240 | 5,709.33 | 5.71 | 0.020554 | 5,920.79 | 5.92 | 0.021315 |
| 6 | 3,011.59 | 9.92 | 0.035711 | 2,934.32 | 9.16 | 0.032977 | 4,147.30 | 11.29 | 0.040645 | 3,414.10 | 13.73 | 0.049442 |

Table 6. Levelized cost rates and average costs per unit of exergy at various state points in the Delta gas turbine system

| State points | DEL1 (GT9) | DEL2 (GT10) | DEL3 (GT18) | DEL4 (GT20) |
|--------------|------------|------------|------------|------------|
| $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) | $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) | $\dot{C}$ ($/h$) | $C$ ($$/GJ$$) | $c$ ($$/kWh$$) |
| 1 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 |
| 2 | 835.48 | 5.43 | 0.019552 | 898.54 | 5.80 | 0.020875 | 7,151.43 | 16.86 | 0.060689 | 6,588.44 | 15.63 | 0.056263 |
| 3 | 1,953.22 | 5.01 | 0.018036 | 2,030.82 | 5.28 | 0.019014 | 14,976.74 | 14.63 | 0.052653 | 14,378.32 | 14.02 | 0.050462 |
| 4 | 494.52 | 5.01 | 0.018036 | 514.49 | 5.28 | 0.019014 | 4,145.25 | 14.63 | 0.052653 | 3,621.36 | 14.02 | 0.050462 |
| 5 | 2,074.70 | 2.07 | 0.007469 | 2,088.17 | 2.09 | 0.007517 | 5,489.87 | 5.49 | 0.019764 | 5,476.39 | 5.48 | 0.019715 |
| 6 | 835.31 | 5.23 | 0.018818 | 898.32 | 5.53 | 0.019900 | 7,150.92 | 15.68 | 0.056460 | 6,587.93 | 14.62 | 0.052639 |
| 7 | 623.50 | 5.23 | 0.018818 | 618.15 | 5.53 | 0.019900 | 3,680.89 | 15.68 | 0.056460 | 4,169.36 | 14.62 | 0.052639 |
For all the plants considered, the combustion chamber and air compressor have the highest value of the sum $\dot{Z} + \dot{C}_D$ and are, therefore, the most important components from the exergoeconomic viewpoint. The low value of exergoeconomic factor, $f$, associated with the combustion chamber suggests that the cost rate of exergy destruction is the dominate factor influencing the component. Hence, it is implied that the component efficiency is improved by increasing the capital investment. This can be achieved by increasing GTIT. The maximum GTIT is limited by the metallurgical considerations (Altayib, 2011; Gorji-Bandpy & Goodarzian, 2011).

Considering air compressor which has the second highest value of the sum $\dot{Z} + \dot{C}_D$ (except for units AF3, AF4 and DEL3), the relatively large value of the factor $f$ suggests that the capital investment and O and M costs dominate. According to Equation (15) of the cost model, the capital investment costs of the air compressor depend on pressure ratio ($P_2/P_1$) and compressor isentropic efficiency $\eta_{sc}$. Reduction of the $\dot{Z}$ value associated with the air compressor may be achieved by

| Table 7. Exergoeconomic parameters of the gas turbine components for AES barges station |
|---------------------------------------------------------------|
| **Exergoeconomic parameters** | **AES1 (PB204)** | **AES2 (PB209)** | **AES3 (PB210)** |
| C_F ($)/GJ | AC | CC | GT | AC | CC | GT | AC | CC | GT |
| 10.73 | 4.96 | 10.69 | 13.75 | 5.28 | 13.65 | 14.07 | 5.32 | 13.65 |
| C_P ($)/GJ | 11.67 | 10.69 | 10.72 | 15.03 | 13.67 | 13.72 | 15.67 | 13.65 | 13.93 |
| E_p (MW) | 3.21 | 53.83 | 0.29 | 3.45 | 51.82 | 0.51 | 4.23 | 59.92 | 2.26 |
| C_F ($)/h | 124.01 | 2,761.63 | 11.06 | 170.72 | 3,493.79 | 25.04 | 214.41 | 3,575.46 | 111.18 |
| C_p + Z ($)/h | 126.22 | 12.56 | 87.19 | 45.98 | 0.41 | 80.01 | 36.53 | 0.36 | 44.05 |
| f (%) | 50.44 | 4.5 | 88.74 | 54.98 | 0.4 | 80.01 | 36.53 | 0.36 | 44.05 |

| Table 8. Exergoeconomic parameters of the gas turbine components for Afam gas turbine station |
|---------------------------------------------------------------|
| **Exergoeconomic parameters** | **AF1 (GT17)** | **AF2 (GT18)** | **AF3 (GT19)** | **AF4 (GT20)** |
| C_F ($)/GJ | AC | CC | GT | AC | CC | GT | AC | CC | GT | AC | CC | GT |
| 9.92 | 5.49 | 7.62 | 9.16 | 5.27 | 10.86 | 11.29 | 5.76 | 10.59 | 12.13 | 6.58 | 12.42 |
| C_P ($)/GJ | 10.44 | 9.62 | 9.84 | 9.84 | 8.86 | 9.08 | 11.73 | 10.59 | 11.06 | 14.95 | 12.42 | 13.25 |
| E_p (MW) | 3.53 | 136.13 | 6.06 | 141.13 | 8.52 | 11.06 | 5.17 | 197.12 | 15.85 | 12.27 | 188.59 | 23.38 |
| C_F ($)/h | 190.33 | 3,193.95 | 205.97 | 138.43 | 2,849.94 | 208.21 | 210.09 | 3,638.03 | 604.19 | 606.50 | 4,615.55 | 1,044.89 |
| C_p + Z ($)/h | 403.02 | 41.58 | 291.66 | 520.31 | 51.38 | 394.97 | 674.10 | 68.66 | 482.37 | 776.95 | 57.18 | 556.31 |
| f (%) | 67.92 | 1.29 | 58.61 | 78.99 | 1.77 | 63.68 | 76.24 | 1.85 | 44.39 | 56.16 | 1.69 | 34.74 |

| Table 9. Exergoeconomic parameters of the gas turbine components for Delta gas turbine station |
|---------------------------------------------------------------|
| **Exergoeconomic parameters** | **DEL1 (GT9)** | **DEL2 (GT10)** | **DEL3 (GT18)** | **DEL4 (GT20)** |
| C_F ($)/GJ | AC | CC | GT | AC | CC | GT | AC | CC | GT | AC | CC | GT |
| 5.23 | 4.20 | 5.01 | 5.53 | 4.28 | 5.28 | 15.68 | 6.83 | 14.63 | 14.62 | 6.54 | 14.02 |
| C_P ($)/GJ | 5.43 | 5.01 | 5.17 | 5.80 | 5.28 | 5.46 | 16.86 | 14.61 | 15.38 | 15.63 | 14.02 | 14.46 |
| E_p (MW) | 1.66 | 52.62 | 3.35 | 2.19 | 50.71 | 3.55 | 8.82 | 130.36 | 13.86 | 8.05 | 131.82 | 8.12 |
| C_F ($)/h | 31.18 | 1,272.00 | 60.51 | 41.74 | 1,339.48 | 67.43 | 497.89 | 5,624.82 | 729.92 | 423.84 | 5,353.55 | 44.60 |
| C_p + Z ($)/h | 168.06 | 13.99 | 104.96 | 216.99 | 18.19 | 136.40 | 508.98 | 43.57 | 324.53 | 510.68 | 43.61 | 325.00 |
| f (%) | 84.35 | 1.09 | 63.43 | 83.87 | 1.34 | 66.92 | 50.55 | 0.77 | 30.78 | 54.65 | 0.81 | 42.23 |
reducing the pressure ratio ($P_2/P_1$) and/or the isentropic efficiency $\eta_{sc}$ (Bejan et al., 1996). Moreover, Tables 7–9 show that the exergy destruction and investment cost are almost equal for air compressor when compared with other components. This implies that the systems performance may be improved by increasing the investment cost of this component.

The results of the exergy costing analysis of the plants investigated show that the combustion chamber (CC) exhibits the greatest exergy destruction cost. The next highest source of exergy destruction cost is the air compressor. In comparing the results of exergy and exergy costing analyses, similar trends are revealed. Increasing GTIT effectively decreases the cost associated with exergy destruction. Further comparisons between related results are consistent with those reported by Ahmadi, Rosen, and Dincer (2011), and confirm that the most significant parameter in the plant is GTIT. The finding establishes the concept that the exergy loss in the combustion chamber is associated with the large temperature difference between the flame and the working fluid. Reducing this temperature difference reduces the exergy loss. Furthermore, cooling compressor inlet air allows the compression of more air per cycle, effectively increasing the gas turbine capacity.

To illustrate the effect of GTIT on the exergy destruction cost of combustion chamber of the selected plants, AES1 (PB204) plant is considered as sample. The simulation was done by varying the GTIT from 950 to 1,500 K. Figure 4 shows the effect of variation in GTIT on combustion chamber exergy destruction cost. This figure shows that, like the exergy analysis results, the cost of exergy destruction for the combustion chamber decreases with an increase in the GTIT. This is due to the fact that the cost of exergy destruction is proportional to the exergy destruction. Hence, an increase in the GTIT can decrease the cost of exergy destruction. Furthermore, from Figure 3, an increase in the TIT of about 200 K can lead to a reduction of about 29% in the cost of exergy destruction. Therefore, TIT is the best option to improve cycle losses.

5. Conclusions and recommendations
In the present study, exergy costing analysis and performance assessment from thermodynamics point of view were performed for 11 selected gas turbine power plants in Nigeria.

The results from the exergy analysis show that the combustion chamber is the most significant exergy destructor in the selected power plants, which is due to the chemical reaction and the large temperature difference between the burners and working fluid. Moreover, the results show that an increase in GTIT leads to an increase in gas turbine exergy efficiency due to a rise in the output power of the turbine and a decrease in the combustion chamber losses.
The results from the exergy costing analysis, in common with those from the exergy analysis, show that the combustion chamber has the greatest cost of exergy destruction compared to other components. In addition, by increasing GTIT, the gas turbine cost of exergy destruction can be decreased. The finding solidifies the concept that the exergy loss in the combustion chamber is associated with the large temperature difference between the flame and the working fluid. Reducing this temperature difference reduces the exergy loss. Furthermore, cooling compressor inlet air allows the compression of more air per cycle, effectively increasing the gas turbine capacity. The results of this study revealed that an increase in the GTIT of about 200 K can lead to a reduction of about 29% in the cost of exergy destruction. Therefore, GTIT is the best option to improve cycle losses. From exergy costing analysis, the unit cost of electricity produced in the selected power plants varies from cents 1.99/kWh (N3.16/kWh) to cents 5.65/kWh (N8.98/kWh).

5.1. Recommendations to improve performance of the selected power plants
To improve thermodynamic effectiveness of the selected gas turbine power plants, it is necessary to deal directly with inefficiencies related to exergy destruction and exergy loss in the systems. The primary contributors to exergy destruction are chemical reaction, heat transfer, mixing, and friction, including unrestrained expansions of gases and liquids. To deal with them effectively, the principal sources of inefficiency not only should be understood qualitatively, but also be determined quantitatively, at least approximately. Design changes to improve effectiveness must be done judiciously.

Based on the results of this research, the following possible technologies to improve performance of the selected gas turbine power plants are hereby recommended:

• The results of this study revealed that the combustion chamber has the largest irreversibility and cost of exergy destruction. This large exergy loss can be reduced in the selected power plants by addition of spray water and preheating of the reactants in the combustion chamber.
• Heat recovery from hot exhaust gases can be used to augment the performance of the gas turbine plant. Combined cycle is a common way to recover thermal energy from the exhaust gases; it is suitable for these plants as they operate as the base load plants (Oyedepo, Fagbenle, Adefila, & Adavbiele, 2014).
• Though gas turbine engines have the advantage of fast startup, they suffer from low power output and thermal efficiency at high ambient temperatures. GT power plants operating in Nigeria are simple GTs; there is a tremendous derating factor due to higher ambient temperatures. The average efficiency of GT plants in the Nigerian energy utility sector over the past two decades was in the range 27–30% (Abam, Ugot, & Igbong, 2012). Therefore, retrofitting GT power plants in Nigeria with advanced cycle would improve their performance significantly. Among many proven technologies are inlet air cooling, intercooling, regeneration, reheating, steam injection gas turbine etc. Air inlet cooling system (evaporative cooling, inlet fogging or inlet chilling method) is a useful option for increasing power output of the selected power plants. This helps to increase the density of the inlet air to the compressor.
• The compressor airfoils of older turbines tend to be rougher than a newer model simply because of longer exposure to the environment. In addition, the compressor of older models consumes a larger fraction of the power produced by the turbine section. Therefore, improving the performance of the compressor will have a proportionately greater impact on total engine performance. Application of Coatings to gas turbine compressor blades (the “cold end” of the machine) would improve the selected gas turbine engines performance. Compressor blade coatings provide smoother, more aerodynamic surfaces, which increase compressor efficiency. In addition, smoother surfaces tend to resist fouling because there are fewer “nooks and crannies” where dirt particles can attach. Coatings are designed to resist corrosion, which can be a significant source of performance degradation, particularly if a turbine is located near saltwater. As AES Barge gas turbine plant is located on lagoon, compressor coating technology would improve the plant performance significantly.
Another option for improving the selected gas turbine plants performance is to apply ceramic coatings to internal components. Thermal barrier coatings (TBCs) are applied to hot section parts in advanced gas turbines. As some of the selected gas turbines are over 25 years in operation, TBCs can be applied to the hot sections of the selected gas turbines. The TBCs provide an insulating barrier between the hot combustion gases and the metal parts. TBCs will provide longer parts life at the same firing temperature, or will allow the user to increase firing temperature while maintaining the original design life of the hot section.

Nomenclature

Symbols

- $\dot{E}$: exergy rate (kW)
- $\dot{E}_L$: exergy loss rate
- $\dot{E}_D$: exergy destruction rate
- ExIP: exergetic improvement potential
- $h$: specific enthalpy (kJ/kg)
- $I$: irreversibility
- $ke$: kinetic energy (kJ)
- $m$: mass (kg)
- $\dot{m}$: mass flow (kg/s)
- $p$: pressure (bar)
- $P$: power output (kW)
- $pe$: potential energy (kJ)
- $\dot{Q}$: heat (W)
- $r_p$: pressure compression ratio
- $R$: gas constant (kJ/mol K)
- $\dot{S}$: entropy rate
- $T$: temperature, either (K) or (°C)
- $T_{pz}$: primary zone combustion temperature
- $W_c$: compressor work (W)
- $W_t$: turbine work (W)
- $W$: work (W)
- $y_D$: exergy destruction rate ratio

Greek symbols

- $\eta_c$: isentropic efficiency of compressor
- $\eta_T$: isentropic efficiency of turbine
- $\eta_{th}$: thermal efficiency
- $\varepsilon$: exergetic efficiency
- $\varnothing$: rational efficiency
- $\delta$: component efficiency defect
- $\psi$: overall exergetic efficiency
- $\xi$: exergetic performance coefficient

Subscripts

- $i$: inlet
- $e$: exit or outlet
- $p$: pressure
Superscripts

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

Abbreviation

LHV	lower heating value
TET
turbine exit temperature (K)
TDI	thermal discharge index
GTIT
gas turbine inlet temperature (K)

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Source: Oyedepo (2014).

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