Performance evaluation of selected gas turbine power plants in Nigeria using energy and exergy methods

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Abstract

This study presents thermodynamic analysis of the design and performance of eleven selected gas turbine power plants using the first and second laws of thermodynamics concepts. Energy and exergy analyses were conducted using operating data collected from the power plants to determine the energy loss and exergy destruction of each major component of the gas turbine plant. Energy analysis showed that the combustion chamber and the turbine are the components having the highest proportion of energy loss in the plants. Energy loss in combustion chamber and turbine varied from 33.31 to 39.95% and 30.83 to 35.24% respectively. The exergy analysis revealed that the combustion chamber is the most exergy destructive component compared to other cycle components. Exergy destruction in the combustion chamber varied from 86.05 to 94.67%. Combustion chamber has the highest exergy improvement potential which range from 30.21 to 88.86 MW. Also, its exergy efficiency is lower than that of other components studied, which is due to the high temperature difference between working fluid and burner temperature. Increasing gas turbine inlet temperature (GTIT), the exergy destruction of this component can be reduced.

Key words: Energy, Exergy, Exergy efficiency, Exergy destruction, Gas turbine

1. Introduction

The importance of developing thermal systems that effectively use energy resources such as natural gas is apparent. Based on this fact, assessment of energy system performance becomes paramount. There are many ways to assess energy system performance. The most common energy system performance assessment criteria are energy-based and exergy-based analyses (Lior and Zhang 2007).

Energy analysis is traditionally used in industries to carry out performance comparisons and optimizations. The conventional methods of energy analysis are based on the first law of thermodynamics, which is concerned with the conservation of energy (Ray et al., 2007). Exergy analysis is based on the second law of thermodynamics, and generally allows process inefficiencies to be better pinpointed. Exergy is often treated as a measure of economic value.
Energy and exergy analyses are important to explain how energy flows interact with each other and how the energy content of resources is exploited. The energetic efficiency (1st law efficiency) supplies information about the efficiency in using energy resources to get the products. The exergetic efficiency (2nd law efficiency) is used to explain efficiency from the exergetic point of view. These two indicators (1st law efficiency and 2nd law efficiency) have wide range of application at thermal power system and component level (Mirandola et al., 2000). A complete analysis of the thermodynamic performance of a thermal power plant requires the use of both energy and exergy analyses. The exergy analysis is a method that combines both the first and second laws of thermodynamics for the design and analysis of thermal systems. The objective of exergy analysis is to identify areas in the system where exergy destruction and losses occur and their magnitude so that attention can be focused on the components of the system that offer the greatest opportunities for improvement. Exergy analysis, because it accounts for losses due to internal consumptions and external wastes, is regarded by many researchers to give more meaningful and illuminating results than energy analysis (Rosen and Scott, 1985; Ghazikhani, 2014; Badami, 2014).

The energy analysis which is based on the first law of thermodynamics provides no information about the irreversibility aspects of thermodynamic processes. Whereas, exergy analysis is a thermodynamic analysis technique based on the combined principles of conservation of mass and energy together with second law of thermodynamics, provides an alternative and illuminating means of assessing the locations, types and magnitudes of wastes and losses and to identify meaningful efficiencies of the system (Ahmadi and Dincer, 2011).

The majority of the causes of thermodynamic imperfection of thermal and chemical processes of thermal power plants cannot be detected by means of an energy analysis. For example, irreversible heat transfer, throttling, and adiabatic combustion are not associated with any energy loss, but they lead to decrease in the energy quality, reduce its ability to be transformed into other kind of energy, and, therefore, increase the operational cost of installation and environmental impact. These effects of aforementioned irreversible phenomenon can be detected and evaluated by second law of thermodynamics. The main purpose of the exergy analysis is to detect and evaluate quantitatively the causes of the thermodynamic imperfection of thermal processes (Kumar, 2009). Therefore, exergy analysis plays an important role in developing strategies and providing guidelines for more effective use of energy in the existing power plants (Kanoglu et al., 2007; Ahmadi and Dincer, 2010).

Recently, energy and exergy analyses have been used by many researchers in thermal systems, especially in performance evaluation of thermal power plants. These include but limited to the following studies: Mitrovic’ et al. (2010) presented the energy and exergy analysis of Kostolac power plant in Serbia. The performance of the plant was estimated by a component-wise modeling, and a detailed break-up of energy and exergy losses for the considered plant. Highest energy losses occurred in the condenser where 421 MW was lost to the environment while only 105.78 MW was lost from the boiler. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (88.2%) followed by the turbines (9.5%), and then the forced draft fan condenser (0.5%). In addition, the calculated thermal efficiency based on the lower heating value of fuel was 39% while the exergy efficiency of the power cycle was 35.77%. The energy and exergy analyses of Al-Hussein power plant in Jordan was performed by Aljund (2009). The performance of the plant was estimated by a component wise modeling and a detailed break-up of energy and exergy losses for the considered plant was presented. Energy losses mainly occurred in the condenser where 421 MW was lost to the environment while only 105.78 MW was lost from the boiler system. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (88.2%) followed by the turbine (9.5%), and then the forced draft fan condenser (0.5%). In addition, the calculated thermal efficiency based on the lower heating value of fuel was 39% while the exergy efficiency of the power cycle was 35.77%. The energy and exergy analyses of Al-Hussein power plant in Jordan was performed by Aljund (2009). The performance of the plant was estimated by a component wise modeling and a detailed break-up of energy and exergy losses for the considered plant was presented. Energy losses mainly occurred in the condenser where 134 MW was lost to the environment while only 13 MW was lost from the boiler system. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (77%) followed by the turbine (13%), and then the forced draft fan condenser (9%). The calculated thermal efficiency based on the lower heating value of fuel was 26% while the exergy efficiency of the power cycle was 25%. Sanjay et al. (2007) performed parametric energy and exergy analysis of reheat gas–steam combined cycle using closed-loop-steam-cooling. The reheat gas–steam
combined cycle plant with closed-loop-steam-cooling exhibits enhanced thermal efficiency (around 62%) and plant specific work as compared to basic steam–gas combined cycle with air-film cooling as well as closed-loop-steam cooling. With closed-loop-steam-cooling, the plant efficiency, reached an optimum value in higher range of compressor pressure ratio as compared to that in film air-cooling. Component-wise inefficiencies of steam cooled-reheat gas–steam combined cycle based on the second-law-model (exergy analysis) was found to be the maximum in combustion-chamber (∼30%), followed by that in gas turbine (∼4%). Sanjay and Prasad (2013) presented energy and exergy analysis of intercooled combustion-turbine based combined cycle power plant. An enhancement of 20% in plant work output in case of inter-cooled combustion-turbine based combined cycle in comparison to basic combustion-turbine based combined cycle was observed, while the plant efficiency was marginally enhanced. Exergy analysis showed that the rational-efficiency of the combustion-turbine was higher by about 3.13% as compared to basic combustion-turbine cycle. Exergy destroyed in components of inter-cooled cycle was found lower for all the components except the combustion chamber. The authors concluded that improved plant performance in terms of plant efficiency resulted in correspondingly reduced green-house-gas emissions. Kumar and Sanjay (2013) presented exergy analysis of a gas / steam combined cycle. The second law approach has been used to evaluate component wise exergy destruction for different values of investigated parameters. The cycle components was investigated with respect to the effect of varying values of air/fuel ratio and compression ratio on the performance parameters like component-wise percentage exergy destruction and rational efficiency of gas turbine and steam turbine. The higher values of compression ratio and low air/fuel ratio correspond to lower exergy destruction associated with combustion. The authors concluded that for maximising plant efficiency, turbine inlet temperature should be on the higher side (1600–1700 K) and turbine exit temperature should be kept as low as possible (< 750 K). Ebadi and Gorji-Bandpy (2005) performed an exergetic analysis on a 116-MW gas-turbine power plant. Mass and energy conservation laws were applied to each component of the system. Quantitative exergy balance for each component and for the whole system was considered. The effect of a change in the inlet turbine temperature on the exergetic efficiency and exergy destruction in the plant was evaluated. The crucial dependency of the exergetic efficiency and the exergy destruction on the change in the turbine inlet temperature was confirmed. Exergy analysis of the combined Brayton/Rankine power cycle of National Thermal Power Corporation (NTPC) Dadri India was performed by Tiwari et al. (2013). Theoretical exergy analysis was carried out for different components of the power plant. The greatest exergy losses occurred in the combustion chamber (35% of the total exergy losses), while the exergy losses in other plant components were between 7% and 21% of the total exergy losses at 1400 °C turbine inlet temperature and pressure ratio 10. There were clear effects in the exergy losses when varying pressure ratio and turbine inlet temperature. Ighodaro and Aburime (2011) presented exergetic appraisal of Delta IV Power Station, Ughelli, Nigeria. The first and second laws of thermodynamics were used to assess the performance of the plant. Mass and energy conservation laws were applied to each component. The compressor was found to have the largest exergetic efficiency while that of the total plant was 45.7%. The combustion chamber had the largest exergetic destruction (56%) while that of the total plant was 58.5%. Chen et al. (2014) presented the off design performance analysis of a combined cooling, heating, and power (CCHP) system consisting of a small-scale gas turbine, an exhaust-fired double-effect absorption chiller, and a heat exchanger. The energy and exergy analyses of the CCHP system were investigated under the rated and part-load conditions. The CCHP system was energy saving when the power output of the gas turbine exceeded 30% of the full load. It was also found that the CO2 emission of the CCHP system reduced by 66.7%-70.5%, compared with conventional separation system, when the power output of gas turbine increased from about 30% to 100%. Energy level results revealed that the combustor of the small-scale gas turbine mainly contributed to the deteriorated performance of the CCHP system. The case results indicated that using off-design data leads to a more realistic
evaluation of the CCHP system. Cihan et al. (2006) performed energy and exergy analyses for a combined-cycle power plant by using the data taken from its units in operation to identify the potential for improving efficiency of the system. Energy and exergy fluxes at the inlet and the exit of the devices in one of the power plant main units as well as the energy and exergy losses were determined. The results showed that combustion chambers, gas turbines and heat recovery steam generators (HRSG) were the main sources of irreversibilities representing more than 85% of the overall exergy losses. Some constructive and thermal suggestions were made to improve the efficiency of the system.

Most of the past studies on energy and exergy analyses of thermal power plants were based on a single power unit. In the present work, energy and exergy analyses are performed on eleven (11) gas turbine units at three different stations in Nigeria.

The prime objectives of the study are:
1. To evaluate energy loss in each component of the selected gas turbine power plants using the first law of thermodynamics.
2. To establish exergy consumption and destruction in the same components of the gas turbine power plant using both the first and second laws of thermodynamics.
3. To identify the most significant sources (s) of exergy destruction in the power plants and the location(s) of occurrence and to demonstrate exergy analysis as a powerful tool to assist in the quest for sustainable development.
4. To identify the component that has potential for significant performance improvement.
5. To recommend possible ways to improve performance of the existing selected power plants based on the outcome of this study.

2. System description

Gas turbine power plants in Nigeria operate on simple gas turbine engine. The simple gas turbine power plant mainly consists of a gas turbine coupled to a rotary type air compressor and a combustor (or combustion chamber) which is placed between the compressor and turbine in the fuel circuit. Auxillaries, such as cooling fan, water pumps, etc. and the generator itself are also driven by the turbine. Other auxillaries 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

The energetic and exergetic efficiencies of the entire unit that make up the selected gas turbine plants were evaluated using MATLAB R2010a and Microsoft Excel. For the purpose of investigating the effect of interaction of the plant’s units on the energetic and exergetic efficiencies, the thermal power plant units were then grouped into subsystem and overall system, as clearly marked out in Figure 1. The energy and exergy balances on inlet and exit streams of each process unit were used in the estimation of their energetic and exergetic efficiencies.

3.1. Energy analysis

For any control volume at steady state with negligible potential and kinetic energy changes, energy balance can be expressed as:

\[ Q - W = \sum \dot{m}_{he} e - \sum \dot{m}_{hi} i \]  

(1)

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

- **Compressor**

At full load, Compressor work rate, \( W_c \), is given in terms of pressure ratio and inlet compressor temperature as:

\[ W_c = \frac{m_c c_{pa} T_1}{\eta_c} \left( r_p \gamma_a - 1 \right) \]  

(2)

Where \( c_{pa} \) is the specific heat capacity of air which is considered in this study as a temperature variable function and can be fitted by Equation (3)
for the range of 200 K < T < 800 K (Rahman et al., 2011; Kurt et al., 2009):

\[
c_{pa}(T) = 1.04841 - \frac{3.8371 T}{10^4} + \frac{9.4517 T^2}{10^7} - \frac{5.49031 T^3}{10^{10}} + \frac{7.9298 T^4}{10^{14}}
\]

(3)

Energy input to air compressor at ambient temperature is calculated by:

\[
\dot{Q}_{c1} = \dot{m}_a (c_{pa} T_2 - c_{pa} T_a)
\]

(4)

Where \(T_a\) is the ambient temperature

Energy input to air compressor at specific inlet temperature (\(T_1\)) is given as:

\[
\dot{Q}_{c2} = \dot{m}_a (c_v T_2 - c_v T_1)
\]

(5)

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

\[
\dot{Q}_{c loss} = \dot{m}_a (c_v T_2 - c_v T_a) - \dot{m}_a (c_v T_2 - c_v T_1)
\]

(6)

- **Combustion Chamber**

  The energy balance in the combustion chamber is expressed as:

  \[
  \dot{m}_a h_2 + \dot{m}_g LHV = \dot{m}_g h_3 + \dot{Q}_{icc}
  \]

  (7)

  Where

  \[
  \dot{Q}_{icc} = \dot{m}_g LHV \left( 1 - \eta_{cc} \right)
  \]

  (8)

  Energy loss in the combustion chamber is calculated as shown in equation (9):

  \[
  \dot{Q}_{c loss} = \dot{m}_a h_2 + \dot{m}_g LHV - \dot{m}_g h_3
  \]

  (9)

  Where, \(m_f\) is fuel mass flow rate (kg/s), \(m_a\) is air mass flow rate (kg/s), \(m_g\) is combustion product mass flow rate (kg/s), LHV is low heating value, \(h_2\) is enthalpy of compressed air, \(h_3\) is enthalpy of combustion product and \(Q_{icc}\) is heat added in combustion chamber.

Pressure drop across combustion chamber (\(\Delta P_{cc}\)) is usually around 2% (Adrian and Dorin, 2010; Barzegar et al., 2011). Turbine inlet pressure (\(P_3\)) can be calculated as:

\[
P_3 = P_2 \left( 1 - \Delta P_{cc} \right)
\]

(10)

Where \(P_3\), turbine entry level pressure in Pa; \(P_2\) is the combustion chamber inlet temperature, \(\Delta P_{cc}\) is pressure drop across the combustion chamber.

- **Gas turbine**

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

  \[
  W_T = m_g c_{pg} T_3 \eta_T \left[ 1 - \left( \frac{r_T}{\gamma_c} \right)^{1-\gamma_c} \right]
  \]

  (11)

  where, \(c_{pg}\) is the specific heat capacity of combustion product (gas). The \(c_{pg}\) in this work is considered to be a temperature variable function as [34]:

  \[
c_{pg}(T) = 0.991615 + \left( \frac{6.99703 T}{10^8} \right) + \left( \frac{2.7129 T^2}{10^7} \right) - \left( \frac{1.22442 T^3}{10^{10}} \right)
  \]

  (12)

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

  \[
  \dot{W}_n = m_g c_{pg} T_3 \eta_T \left[ 1 - \left( \frac{r_T}{\gamma_c} \right)^{1-\gamma} \right]
  \]

  \[- \frac{m_c c_{pa} T_1}{\eta_C} \left( \frac{\gamma - 1}{\gamma_c} \right) \left( \frac{\gamma - 1}{\gamma_c} - 1 \right) \]

  (13)

  Where

  \[
m_g = \dot{m}_a + \dot{m}_f
  \]

  (14)

  The overall power output from gas turbine plant in terms of pressure ratio, compressor inlet temperature and turbine inlet temperature is given as:

  \[
P = m_g \left[ c_{pg} T_3 \eta_T \left[ 1 - \left( \frac{r_T}{\gamma_c} \right)^{1-\gamma_c} \right] \frac{c_{pa} T_1}{\eta_C} \left( \frac{\gamma - 1}{\gamma_c} \right) \left( \frac{\gamma - 1}{\gamma_c} - 1 \right) \right]
  \]

  (15)
Energy input in the turbine is given as:

\[ \dot{Q}_T = \dot{m}_c n_3 T_3 \]  

(16)

Energy utilized for turbine work is calculated as follows:

\[ \dot{Q}_{NW} = \dot{m}_g (c_v T_3 - c_v T_{ET}) \]  

(17)

Where \( T_3 \) is the combustion chamber exit temperature and \( T_{ET} \) is turbine exhaust temperature. Energy loss from the turbine is given as:

\[ \dot{Q}_{loss} = \dot{m}_c (c_v T_{ET}) \]  

(18)

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

\[ \dot{Q}_{T loss} = \dot{Q}_{loss} + \dot{Q}_{C loss} + \dot{Q}_{T loss} \]  

(19)

The gas turbine thermal efficiency (\( \eta_{th} \)) can be determined by Equation (20):

\[ \eta_{th} = \frac{\dot{W}_c}{\dot{m} LHV} \]  

(20)

Equation (20) is also known as first law efficiency of gas turbine.

3.1.1. Thermal discharge index (TDI)

Thermal discharge index of power system is the number of thermal energy unit discharged in to the environment for each unit of electrical energy produced by the plant.

This index cannot be zero or else the plant violates the second law of thermodynamics; but the index should be as low as possible to improve the efficiency of the plant as well as to keep the pollution level low (Mozafari et al., 2010; Rajput, 2005). The thermal discharge index is strongly dependent on the thermal efficiency and it can be expressed in terms of thermal efficiency as follows (Ofodu and Abam, 2002):

\[ \text{TDI} = \frac{P_{th}(1 - \eta_{th})}{P_{th} \eta_{th}} = \frac{1}{\eta_{th}} - 1 \]  

(21)

Where \( P_{th} \) is the thermal energy input and \( \eta_{th} \) is the thermal efficiency.

3.2. Exergy analysis

Exergy is composed of two important parts. The first one is the physical exergy and the second one is the chemical exergy. In this study, the kinetic and potential parts of exergy are considered negligible. The physical exergy is defined as the maximum theoretical useful work obtained as a system interact with an equilibrium state. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion process.

In order to carry out exergy analysis, mass and energy balances on the system are required to determine the flow rates and energy transfer rates at the control surface. By applying the first and second laws of thermodynamics, the exergy balance is obtained as follows (Mansouri et al., 2012):

In the absence of kinetic and potential exergy effects, the total exergy of a system becomes

\[ E^{\text{tot}} = E^{\text{PH}}_{\text{sys}} + E^{\text{CHE}} \]  

(22)

A general exergy - balance equation, applicable to any component of a thermal system may be formulated by utilising the first and second laws of thermodynamics (Abam and Moses, 2011). The thermo-mechanical exergy stream may be decomposed into its thermal and mechanical components so that the balance in rate form gives:

\[ \dot{E}_{i}^{\text{PH}} - \dot{E}_{e}^{\text{PH}} = \left( \dot{E}_{i}^{T} - \dot{E}_{e}^{T} \right) + \left( \dot{E}_{i}^{P} - \dot{E}_{e}^{P} \right) \]  

(23)

where the subscripts \( i \) and \( e \) represent inlet and exit states, \( \dot{E}_{i}^{PH} \) is the physical exergy of a material stream, \( \dot{E}_{i}^{T} \) is the thermal component of the exergy stream, \( \dot{E}_{i}^{P} \) is the mechanical component of the exergy stream, the term on the left – hand side of the equation represent the change in exergy of the flow stream, the first and second terms on the right – hand side of the equation represent the changes in the thermal and mechanical components of the exergy stream respectively.

The thermal and mechanical components of the exergy stream for an ideal gas with constant specific heat may be written respectively as (Oh et al., 1996; Ebadi and Gorji-Bandpy, 2005):

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

(24)
And

\[ \dot{E}^P = \dot{m}RT_0 \ln \frac{P}{P_0} \]  

(25)

Where \( P_0 \) and \( T_0 \) are the pressure and temperature, respectively, at standard state; \( \dot{m} \) is the mass flow rate of the working fluid; \( R \) is the gas constant, \( c_p \) is the specific heat at constant pressure.

The general exergy equation applicable to all the components of the gas turbine plant may be written, utilizing the decomposition defined in equation (22) as follows (Ebadi and Gorji-Bandpy, 2005):

\[ \dot{E}^W = \dot{E}^{CHE} + \left( \sum_{\text{inlet}} \dot{E}^T_i - \sum_{\text{exit}} \dot{E}^T_e \right) \]

+ \left( \sum_{\text{inlet}} \dot{E}^P_i - \sum_{\text{exit}} \dot{E}^P_e \right) \]

+ \( T_0 \left( \sum_{\text{exit}} \dot{S} - \sum_{\text{inlet}} \dot{S} + \frac{\dot{Q}_{CV}}{T_{\text{in, CV}}} \right) \)  

(26)

Where \( \dot{E}^W \) represents the exergy rate of power output; the term \( \dot{E}^{CHE} \) denotes the rate of chemical exergy flow of fuel in the plant; \( \dot{S} \) in the entropy transfer rate; \( T_{\text{in, CV}} \) is the temperature of the source from which the heat is transferred to the working fluid; the fourth right hand term is the exergy destroyed in the component; and \( \dot{Q}_{CV} \) in the fourth term denotes the heat transfer rate between the component and the environment.

For many fuels, the chemical exergy can be estimated on the basis of the Lower Heating value (LHV). The relation between the LHV and the estimated parameter of the performance of the system.

\[ \dot{E}^{WGT} = \left( \dot{E}^T_3 - \dot{E}^T_4 \right) \]

+ \left( \dot{E}^P_3 - \dot{E}^P_4 \right) + \dot{m}T_0 \left( \dot{S}_3 - \dot{S}_4 \right) \]

(30a)

\[ \dot{E}_{\text{ex}} = \dot{m}T_0 \left[ c_p\left( \frac{T_3}{T_2} \right) - R \left( \frac{P_2}{P_1} \right) \right] \]  

(30b)

3.2.1. Exergy improvement potential and exergetic efficiency of gas turbine plant

The exergy improvement potential of energy conversion system is a measure of how much and how easily the system could be improved for optimization purposes. It is a thermodynamic approach combining exergy losses and effectiveness to have a more complete parameter of the performance of the system (Rivero et al., 2004). The exergy improvement

\[ \dot{E}_{DAC} = T_0 \left( \dot{S}_2 - \dot{S}_1 \right) \]

\[ \dot{m}T_0 \left[ c_p\left( \frac{T_2}{T_1} \right) - R \left( \frac{P_2}{P_1} \right) \right] \]  

(28b)

Combustion Chamber

\[ \dot{E}_{CHE} + \left( \dot{E}^T_2 + \dot{E}^T_3 + \dot{E}^T_4 \right) \]

+ \left( \dot{E}^P_2 + \dot{E}^P_3 + \dot{E}^P_4 \right) + \dot{T}_0 \left( \dot{S}_3 - \dot{S}_2 + \dot{S}_3 + \frac{\dot{Q}_{cc}}{T_0} \right) = 0 \]

(29a)

Gas Turbine

\[ \dot{E}^{WGT} = \left( \dot{E}^T_3 - \dot{E}^T_4 \right) \]

+ \left( \dot{E}^P_3 - \dot{E}^P_4 \right) + \dot{T}_0 \left( \dot{S}_3 - \dot{S}_4 \right) \]

(30a)
potential makes it possible to determine the critical points of the system stating a hierarchy on its components in such a way that the measure be applied in the places where they will be most effective.

The exergetic improvement potential is obtained from the exergy losses and the efficiency of the system. It is calculated by the Equation (31) (Hammond, 2004):

\[ \text{ExIP} = (1 - \varepsilon)I \]  

(31)

where, \( \text{ExIP} \) is the exergetic improvement potential, \( \varepsilon \) is the exergetic efficiency (%) and \( I \) is the exergy loss or irreversibility rate.

\[ I = \Delta E_{i,w} = E_i - E_{i,w} > 0 \]  

(32)

In any real engineering system (which is irreversible) exergy is degraded and the exergy efficiency is consequently less than unity. According to Van Gool (1992) the maximum improvement in the exergy efficiency for a process or system is obviously achieved when \( \Delta E_{\text{lost}} \) is minimized.

The \( i^{th} \) component efficiency defect denoted by \( \delta_i \) is given by Equation (33) (Abam et al., 2011):

\[ \delta_i = \frac{\sum \Delta E_{i,w}}{\sum \Delta E_{i,w}} \]  

(33)

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

\[ \xi = \frac{E_{\text{D}} \text{Total}}{W_{\text{net}}} \]  

(34)

where \( \xi \) is the exergetic performance coefficient.

Exergy destruction rate and efficiency equations for the gas turbine power plant components and for the whole cycle are summarized in Table 1.

Exergetic efficiencies are used to evaluate the effectiveness of engineering measures taken to improve the performance of a thermal system. This is done by comparing the efficiency values determined before and after modifications have been made to show how much improvement has been achieved (Price, 1992). Moreover, exergetic efficiencies can be used to gauge the potential for improvement in the performance of a given thermodynamic system by comparing the efficiency of the system to the efficiency of like systems. A significant difference between these values would suggest that improved performance is possible (Moran and Shapiro, 2010; Khaldi and Adouane, 2011).

4. Results and discussion

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

4.1. Results of energy analysis

The energy loss experienced in the gas turbine components are shown in Table 3. The average operating data for period of six years (2005 – 2010) presented in Table 2 were used as inputs to the analytical models (Eqns. 6, 9, 18, 19 and 20). For the period of six years, the thermal efficiency of the plants as evaluated varied from 20.35 to 40.97%. From the results, it is seen that DEL4 and DEL3 have the highest thermal efficiency due to their high turbine inlet temperature, high pressure ratio and high fuel flow rate as observed from operating data
Table 2. Average Operating Data for Selected Gas Turbine Power Plants

<table>
<thead>
<tr>
<th>Plant/Average Operating Data</th>
<th>AES Station</th>
<th>Afam Station</th>
<th>Delta Station</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>PB204 (AES1)</td>
<td>PB209 (AES2)</td>
<td>PB210 (AES3)</td>
</tr>
<tr>
<td></td>
<td>GT17 (AF1)</td>
<td>GT18 (AF2)</td>
<td>GT19 (AF3)</td>
</tr>
<tr>
<td></td>
<td>GT20 (AF4)</td>
<td>GT9 (DEL1)</td>
<td>GT10 (DEL2)</td>
</tr>
<tr>
<td></td>
<td>GT18 (DEL3)</td>
<td>GT20 (DEL4)</td>
<td></td>
</tr>
<tr>
<td>Ambient temperature, $T_1$ (K)</td>
<td>303.63</td>
<td>302.31</td>
<td>305.28</td>
</tr>
<tr>
<td>Compressor outlet temperature, $T_2$ (K)</td>
<td>622.31</td>
<td>627.48</td>
<td>636.28</td>
</tr>
<tr>
<td>Turbine inlet temperature, $T_3$ (K)</td>
<td>1218.62</td>
<td>1256.86</td>
<td>1222.45</td>
</tr>
<tr>
<td>Turbine outlet temperature, $T_4$ (K)</td>
<td>750.00</td>
<td>755.00</td>
<td>827.05</td>
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<tr>
<td>Temperature of exhaust gas, $T_{exh}$ (K)</td>
<td>715.40</td>
<td>750.52</td>
<td>746.48</td>
</tr>
<tr>
<td>Compressor inlet pressure, $P_1$ (bar)</td>
<td>1.013</td>
<td>1.013</td>
<td>1.013</td>
</tr>
<tr>
<td>Compressor outlet pressure, $P_2$ (bar)</td>
<td>9.8</td>
<td>9.86</td>
<td>9.60</td>
</tr>
<tr>
<td>Pressure ratio</td>
<td>9.00</td>
<td>9.14</td>
<td>9.48</td>
</tr>
<tr>
<td>Mass flow rate of fuel (kg/s)</td>
<td>2.58</td>
<td>2.54</td>
<td>2.81</td>
</tr>
<tr>
<td>Inlet mass flow rate of air (kg/s)</td>
<td>122.16</td>
<td>122.20</td>
<td>121.93</td>
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<tr>
<td>Power output (MW)</td>
<td>29.89</td>
<td>29.37</td>
<td>31.52</td>
</tr>
<tr>
<td>LHV of fuel (kJ/kg)</td>
<td>47,541.57</td>
<td>47,541.57</td>
<td>47,541.57</td>
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</tbody>
</table>

in Table 2. Energy performance analysis shows that the combustion chamber has the highest proportion of energy loss in AES1 (33.31%), AES2 (35.99%), AES3 (36.72%), DEL1 (39.61%) and DEL2 (39.95%). The turbine has the highest proportion of energy loss in AF1 (32.40%), AF2 (33.39%), AF3 (31.86%), AF4 (30.83%), DEL3 (35.22%) and DEL4 (35.24%).

AES 1 (i.e unit PB 204) is used as an example to illustrate the effect of operating parameters on energy loss of gas turbine components for the selected power stations. Figure 2 shows the percentage energy loss in the air compressor against compressor inlet temperature. The energy losses in air compressor increase at high ambient temperature. The air compressor work increases as inlet air temperature increases which leads to a decrease in net work of the gas turbine. Air compressor work can be minimized when the air inlet temperature and mass flow rate are reduced. This shows that compressor work can be managed by the compressor inlet air temperature.

Another parameter besides the compressor inlet air temperature is the compression ratio. Figure 3 shows the effect of pressure ratio on energy loss in air compressor. Increase in pressure ratio brings about decrease in energy loss in air compressor. Moreover, compressor work can be reduced by decreasing the compression ratio.

Figure 4 shows that heat energy loss in the combustion chamber decrease with increase in air mass flow rate. This implies that high mass flow rate of air can minimize the energy losses in combustion chamber as this would introduce more air for combustion. The unburnt air in combustion chamber serves as coolant. Therefore, the energy losses decrease as the temperature of mass flow rate of hot gases is decreased. This is due to high quantity of air mass flow which lowers the temperature of the hot gases.

From Table 3, the thermal efficiencies of the selected gas turbine plants vary between 20.35% and 40.97%. AES 2 unit has the least thermal efficiency (20.35%) and DEL4 unit has the highest...
The results show that over 50% of the selected plants have thermal efficiency below 30%. This is attributed to the high-energy degradation experienced in different components of the thermal plants. The results also show that about 1.44 to 3.83 MW of thermal energy is constantly discharged from gas turbine engine to the environment at various stations for each MW of electrical energy produced. The thermal discharge index (TDI) should be as low as possible to improve the efficiency of the plant as well as to keep the pollution level low (Culp, 1991; Rajput, 2005).

Arriving at a decision for plant performance improvement based on energetic performance

### Table 3.
Result of Energy Performance Analysis

<table>
<thead>
<tr>
<th>Energy Performance Indicator</th>
<th>PB204 (AES1)</th>
<th>PB209 (AES2)</th>
<th>PB210 (AES3)</th>
<th>GT17 (AF1)</th>
<th>GT18 (AF2)</th>
<th>GT19 (AF3)</th>
<th>GT20 (AF4)</th>
<th>GT9 (DEL1)</th>
<th>GT10 (DEL2)</th>
<th>GT18 (DEL3)</th>
<th>GT20 (DEL4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Installed Rated Power (MW)</td>
<td>33.5</td>
<td>33.5</td>
<td>33.5</td>
<td>75.0</td>
<td>75.0</td>
<td>138.0</td>
<td>138.0</td>
<td>25.0</td>
<td>25.0</td>
<td>100.0</td>
<td>100.0</td>
</tr>
<tr>
<td>Energy loss of Compressor (MW)</td>
<td>0.71</td>
<td>0.54</td>
<td>0.91</td>
<td>0.84</td>
<td>1.25</td>
<td>0.96</td>
<td>1.16</td>
<td>0.36</td>
<td>0.47</td>
<td>1.19</td>
<td>1.43</td>
</tr>
<tr>
<td>Energy loss of combustion chamber (MW)</td>
<td>99.28</td>
<td>112.47</td>
<td>115.68</td>
<td>43.78</td>
<td>47.49</td>
<td>97.58</td>
<td>78.68</td>
<td>144.74</td>
<td>146.63</td>
<td>93.12</td>
<td>91.74</td>
</tr>
<tr>
<td>Energy loss of turbine (MW)</td>
<td>105.34</td>
<td>106.42</td>
<td>104.53</td>
<td>267.75</td>
<td>273.07</td>
<td>327.95</td>
<td>346.47</td>
<td>100.08</td>
<td>105.90</td>
<td>272.62</td>
<td>272.60</td>
</tr>
<tr>
<td>Total energy loss (MW)</td>
<td>205.33</td>
<td>219.43</td>
<td>221.12</td>
<td>312.38</td>
<td>321.81</td>
<td>426.48</td>
<td>426.31</td>
<td>245.87</td>
<td>253.01</td>
<td>366.93</td>
<td>365.77</td>
</tr>
<tr>
<td>Net Work of Turbine (MW)</td>
<td>46.37</td>
<td>46.35</td>
<td>47.79</td>
<td>114.06</td>
<td>139.07</td>
<td>139.60</td>
<td>130.20</td>
<td>63.02</td>
<td>57.88</td>
<td>174.50</td>
<td>174.97</td>
</tr>
<tr>
<td>% Energy loss of compressor</td>
<td>1.50</td>
<td>1.15</td>
<td>1.94</td>
<td>0.64</td>
<td>0.92</td>
<td>0.64</td>
<td>0.78</td>
<td>0.63</td>
<td>0.84</td>
<td>0.79</td>
<td>0.95</td>
</tr>
<tr>
<td>% Energy loss of combustion chamber</td>
<td>33.31</td>
<td>35.99</td>
<td>36.72</td>
<td>7.85</td>
<td>7.98</td>
<td>13.67</td>
<td>11.18</td>
<td>39.61</td>
<td>39.95</td>
<td>13.48</td>
<td>13.30</td>
</tr>
<tr>
<td>% Energy loss of turbine</td>
<td>31.98</td>
<td>31.88</td>
<td>32.24</td>
<td>32.40</td>
<td>33.39</td>
<td>31.86</td>
<td>30.83</td>
<td>35.20</td>
<td>34.19</td>
<td>35.22</td>
<td>35.24</td>
</tr>
<tr>
<td>Total % energy loss in the plant</td>
<td>66.78</td>
<td>69.01</td>
<td>70.90</td>
<td>40.89</td>
<td>42.30</td>
<td>46.18</td>
<td>42.83</td>
<td>75.45</td>
<td>74.98</td>
<td>49.49</td>
<td>49.49</td>
</tr>
<tr>
<td>Energy input (MW)</td>
<td>298.08</td>
<td>312.53</td>
<td>315.05</td>
<td>557.94</td>
<td>594.93</td>
<td>713.61</td>
<td>703.82</td>
<td>365.43</td>
<td>367.08</td>
<td>690.64</td>
<td>689.88</td>
</tr>
<tr>
<td>Thermal Efficiency</td>
<td>21.72</td>
<td>20.35</td>
<td>20.72</td>
<td>35.85</td>
<td>39.52</td>
<td>31.34</td>
<td>29.89</td>
<td>23.63</td>
<td>21.55</td>
<td>40.77</td>
<td>40.97</td>
</tr>
<tr>
<td>Thermal discharge index</td>
<td>3.60</td>
<td>3.79</td>
<td>3.83</td>
<td>1.79</td>
<td>1.53</td>
<td>2.19</td>
<td>2.35</td>
<td>3.23</td>
<td>3.64</td>
<td>1.45</td>
<td>1.44</td>
</tr>
</tbody>
</table>

Fig. 2. Heat Energy Loss in Air Compressor (%) Against Air Compressor Inlet Temperature (K).

Fig. 3. Heat Energy Loss in Air Compressor (%) Against Compression Ratio.
results only may not be sufficient. For complex systems like gas turbine plant with multiple components this may be misleading as quantifying actual losses in the different system control volumes might not be accurately achieved. Only energetic analysis for decision making is lopsided, since it does not reveal explicit presentation of plant performance. Therefore, the results obtained from energetic performance analysis should be considered with those of exergetic analysis allowing an improved understanding by quantifying the effect of irreversibility occurring in the plant and the locations.

4.2. Results of exergy analysis

An exergy balance for the components of the gas turbine plants and of the overall plants is at this point performed and the net exergy flow rates crossing the boundary of each component of the plants, together with the exergy destruction, exergy defect and exergy efficiency in each component are calculated and are presented in Table 4. The two most important performance criteria are exergy efficiency and exergetic performance coefficient ($\xi$). The values of exergetic efficiency and exergetic performance coefficient varied from 18.22–32.84% and 1.32–2.01 respectively for the considered plants. Since the condition of good performance is derived from a higher overall exergetic efficiency but lower exergetic performance coefficient for any thermal system, hence, it can be inferred that AF2, DEL3 and DEL4 gas turbine plants have good performance. Table 4 also shows the component exergy efficiencies, exergy destruction rates, efficiency defects and total exergy destruction rates of the plants considered in this study. The exergy destruction rates varied from 61.28 to 210.13 MW. AF4 has the highest value (210.13 MW) and AES3 has the least value (61.28 MW). The total efficiency defects and overall exergetic efficiency varied from 67.16 to 81.78% and 18.22 to 31.08% respectively. The efficiency defects are higher for AES3 (81.78%), AES2 (81.77%) and DEL1 (81.66%) than other units. Units AF2 and DEL3 have the highest overall exergetic efficiency and least efficiency defect.

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.

By comparing information in Table 3 and Table 4 together, the total plant loss for the plants varies from 40.89 to 75.49% with average of 57% for energetic consideration and 67.16 to 81.78% with average of about 76% for exergetic case. This shows that using only energetic analysis for decision making is lopsided as it does not reveal explicit presentation of plant performance. Therefore, the result obtained from energy considerations should be considered along with those from exergy analysis. This allows an improved understanding by quantifying the effect of irreversibility occurring in the plant and the locations of occurrence.

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 1000 to 1400 \(^{\circ}\)K, respectively. Figure 5 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. The overall
exergetic efficiency decreased from 18.53 to 17.26% for ambient temperature range of 290 – 320K. It was found that a 5\(^\circ\)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

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

**Exergy improvement potential:**

<table>
<thead>
<tr>
<th>Air Compressor (MW)</th>
<th>3.94</th>
<th>4.14</th>
<th>4.71</th>
<th>7.83</th>
<th>6.88</th>
<th>8.59</th>
<th>14.95</th>
<th>2.89</th>
<th>3.30</th>
<th>11.69</th>
<th>11.02</th>
</tr>
</thead>
<tbody>
<tr>
<td>Combustion Chamber (MW)</td>
<td>38.48</td>
<td>40.33</td>
<td>43.82</td>
<td>69.49</td>
<td>68.52</td>
<td>85.81</td>
<td>88.86</td>
<td>30.47</td>
<td>30.21</td>
<td>78.48</td>
<td>78.67</td>
</tr>
<tr>
<td>Turbine (MW)</td>
<td>0.015</td>
<td>0.023</td>
<td>0.13</td>
<td>0.19</td>
<td>0.22</td>
<td>1.05</td>
<td>2.50</td>
<td>0.15</td>
<td>0.17</td>
<td>1.00</td>
<td>0.38</td>
</tr>
<tr>
<td>Entire Plant (MW)</td>
<td>119.44</td>
<td>147.15</td>
<td>159.28</td>
<td>98.30</td>
<td>89.99</td>
<td>124.18</td>
<td>59.99</td>
<td>54.04</td>
<td>56.46</td>
<td>159.88</td>
<td>143.28</td>
</tr>
</tbody>
</table>
destruction in the combustion chamber (Kotas, 1985).

The exergetic efficiency (or second law efficiency) of the plant was also found to depend significantly on a change in turbine inlet temperature. Figure 6 shows that the second-law efficiency of the plant increases steadily as the turbine inlet temperature increases. The increase in exergetic efficiency with 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 (1200K), after which it increases with TIT. This shows degradation in performance of gas turbine plant at high turbine inlet temperature.

4.2.1. Exergy improvement potential

Table 4 shows the exergy improvement potential of the selected plants. The total improvement potential of the plants varied from 54.04 MW to 159.88 MW. The component with the highest exergy improvement potential is the combustion chamber, which has value vary from 30.21 MW to 88.86 MW. This is followed by the air compressor which has value vary from 3.30 MW to 14.95 MW. This high improvement potential in the combustion chamber is 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. Exergy improvement potential can be afforded in the combustion chamber by preheating the reactants and by reducing the heat loss and the excess air entering the combustion chamber. The lower improvement potential in the air compressor when compared with the combustion chamber is due to relatively heat loss from the air compressor through friction as compared to large temperature difference between the air entering the combustion chamber and the flame temperature. These results have made it possible to determine the critical points of the gas turbine system stating hierarchy on its components in such a way that the measure be applied in the places where they will be most effective.

5. Conclusions and recommendations

In the present study, energetic and exergetic analyses were performed for eleven selected gas turbine power plants in Nigeria.

Energy analysis reveals that thermal efficiency of the selected power plants varied from 20.35 to 40.97%. Also, energy performance analysis shows that combustion chamber has the highest proportion of energy loss in some gas turbine units (5-units) while turbine has the highest proportion of energy loss in other gas turbine units (6-units). Energy loss in combustion chamber varied from 33.31 to 39.95% and the energy loss in turbine varies from 30.83 to 35.24%. Results of energy analysis further show that heat energy loss in air compressor increases with air compressor inlet temperature but decreases with compression ratio. In combustion chamber, heat energy loss decreases with increase in air mass flow rate.
The results from the exergy analysis show that the combustion chamber is the most significant exergy destructor in the selected power 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 the turbine inlet temperature (TIT) 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 exergy analysis results also show that percentage exergy destruction in the combustion chamber varied from 86.05 to 94.67%. The exergy destruction rates varied from 61.28 to 210.13 MW. The total efficiency defects and overall exergetic efficiency of the selected power plants varied from 67.16 to 81.78% and 18.22 to 32.84% respectively. The exergetic performance coefficient varies from 1.32 to 2.01.

The result of this study shows that 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.

### 5.1. Recommendations to improve performance of the selected power plants

Results of this research provide insight into the performance of the selected gas turbine power plants in Nigeria from energy and exergy analyses perspective. Based on the results of this research work, 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.
- The integration of inlet air cooling techniques (IAC) and the steam injection gas turbine (STIG) technologies can be employed with regenerative gas turbine cycle in order to boost power output and generation efficiency of the power generation unit which consequently the most effective way to use retrofitted techniques in the selected power plants.

- 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.
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REFERENCES

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]

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]

$T$  temperature, either [K] or [oC]

$T_{p,z}$  primary zone combustion temperature

$TDI$  thermal discharge index

$\gamma_c$  compressor work (W)

$\gamma_T$  turbine work (W)
$
\gamma_D$  exergy destruction rate ratio

Greek Symbols

$\eta_c$  isentropic efficiency of compressor

$\eta_T$  isentropic efficiency of turbine

$\eta_h$  thermal efficiency

$\varepsilon$  exergetic efficiency

$\Phi$  rational efficiency

$\delta$  component efficiency defect

$\psi$  overall exergetic efficiency

$\xi$  exergetic performance coefficient

Subscripts

$i$  inlet
$e$  exit or outlet
$p$  pressure
$a$  air
$pg$  combustion product
$f$  fuel
$T$  turbine
$cc$  combustion chamber
$th$  thermal
$sys$  system
$0$  ambient
$cv$  control volume
$D$  destruction
$gen$  generation
$ac$  air compressor
$gt$  gas turbine
$k$  Component

Superscripts

$tot$  total

$PH$  physical
$KN$  kinetic
$PT$  potential
$CHE$  chemical
$T$  thermal
$P$

Abbreviation

LHV  lower heating value

TET  turbine exit temperature