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Performance Improvement of a Gas Turbine Power Plant in Nigeria by Exergy Analysis: A Case of Geregu 1

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Abstract

This paper presents the performance improvement study of gas turbine power plant in Nigeria by exergy analysis method using Geregu 1 gas turbine power plant as a case study. The study analyzed the system's components separately in order to identify the sites where the largest exergy losses occur and to quantify the amount. Results of the component efficiencies based on their thermodynamic models of energy and exergy analyses at design and operating years of the power plant revealed that the maximumexergy destruction efficiency occurred in the combustion chamber. Improvement approaches made on the performance of the combustion chamber included: 1) Increasing turbine inlet temperature at constant pressure ratio; 2) Increasing combustion chamber pressure ratio at constant turbine inlet temperature; and 3) Increasing both turbine inlet temperature and pressure ratio. Results of improvement conditions show that the first condition produced progressive decrease in exergy efficiency for the design and operating years when temperature was increased from 1060˚C - 1080˚C at 11 bar pressure. Second condition resulted to increase in exergy efficiency when pressure increased from 11 bar to 15 bar. Similarly, increasing turbine inlet temperature and pressure by 20˚C and 4 bar respectively increased the exergy efficiency by 0.22% on design condition. In addition the exergy efficiency of the operating years increased with increase in both temperature and pressure at the amount. Hence, pressure variation affects exergy efficiency of combustion chamber.

Subject Areas

Energy, Gas Turbines, Exergy

Keywords

Efficiency, Energy, Exergy, Exergy Destruction, Gas Turbine, Power Plant

1. Introduction

Even though there is high demand for energy (electricity) generation from cleaner sources such as solar, wind and hydro, the renewable energy resources alone cannot meet the current global electricity generation demand capacity with the present infrastructure. Hence, fossil fuels will continue to play key roles in electricity generation industry. Owing to this, there is need to improve the performance of the existing fossil energy transformation technologies, even if the construction of new facilities is discouraged, which will not only reduce the cost of electricity generation but also reduce the total energy input for cleaner environment, increase generation capacity and enhance reliability and sustainability. Counties that have sufficient electricity generation, through improved power plant performance, contribute immensely to the global reduction in the number of persons migrating in search of better living conditions.

Electricity generation in Nigeria, has been facing many challenges such as insufficient gas supply to thermal plants, poor transmission and distribution facilities and low performance of existing facilities. Where and when gas supply is available, many gas turbine plants perform below technical expectation due to lack of sufficient understanding of the thermodynamic performance improvement techniques that could be adopted.

The most commonly used method for evaluating the performance efficiency of an energy conversion process is the First law analysis. However, there is increasing interest in combined utilization of the First and Second laws of thermodynamics using concepts of exergy, entropy generation and exergy destruction in order to evaluate the efficiency with which the available energy is consumed. Both laws have been commended appropriate tools for the analysis of energy and exergy of power conversion systems [1].

Exergy analysis allows thermodynamic evaluation of energy conservation because it provides the tool for a clear distinction between energy losses to the environment and internal irreversibilities in the process. The second law of thermodynamics uses an exergy balance for the analysis of thermal systems. Thus, it can play an important role in developing strategies and in providing guidelines for more effective use of energy in the existing power plants. A thermal power plant is a good example of the utilization of exergy analysis [2]. Another important issue is to improve the performance of existing system through exergy and identify the component in which highest exergy destruction takes place. The majority of the causes of thermodynamic imperfection of thermal and chemical processes in thermal power plants cannot be detected by means of an energy analysis. For example, irreversible and heat transfer processes, throttling, and adiabatic combustion are not associated with an energy loss, but they lead to decrease in quality, reduce its ability to be transformed into other kinds of energy, and, therefore, increases the operation cost [3]. These effects can only be detected and evaluated by second law of thermodynamics. The efficacy of the energy-exergy analysis concept in power plants improvements can be attested by the continued interests and research on it by scholars such as [1] [4] [5] [6] [7].

Moran and Shapiro [8] provided a brief survey of exergy principles and analysis along with emphasis on areas of application. They concluded that the exergy balance can be used to determine the location, type and true magnitude of the waste of energy resources, and thus can play an important part in developing strategies for more effective fuel use. Sciubba and Wall [9] presented a brief critical and analytical account of the development of the concept of exergy and its applications. Ameri *et al.* [10] carried out exergy analysis of a combined cycle power plant and showed that the combustion chamber, the gas turbine, duct burner and the heat recovery steam generator are the main sources of irreversibility in the plant which if the components are optimized the power plant improvement can be enhanced. Aljundil [5] presented the energy and exergy analysis of Al-Hussein power plant in Jordan. The work analyzed the system components separately, identified and quantified the sites having the largest energy and exergy losses. In addition, the effect of varying the reference environment state on this analysis was presented. Kaushik *et al.* [6] reviewed some studies on energy and exergy analyses of thermal plants and showed that most gas turbines combustion chambers are most vulnerable to exergy destruction (irreversibilities). The study also showed that exergy methods are useful in assessing which improvements are worthwhile in a thermal power plant, and in addition be used alongside other information for efficiency improvement. Furthermore, Ameri and Enadi [11] also showed by exergy analysis that the gas turbine combustion chamber of a thermal power plant in Iran was the most exergy destructive component in the plant. Dincer and Rosen [12] demonstrated that, although energy and exergy values are dependent on the intensive properties of the dead state, the main results of energy and exergy analyses are usually not significantly sensitive to reasonable variations in these properties as observed in [13]. Most studies on thermal power plants in Nigeria have concentrated on evaluation of the performance of the system [14] [15], performance improvement by inlet air cooling [16] [17] and management and training of operational personnel [18]. There are few studies on the energy and exergy analyses of thermal plants in Nigeria which include the thermodynamic assessment of grid-based gas turbine power plants by Abam *et al.* [19] and the thermodynamic simulation of GT performance. In the face of the present low electricity generation capacity of the country, there is need for more studies on the energetic and exergetic performance of GT power plants in the country with a view to improving their performance. Comprehensive study of the thermal plants will ensure optimal performance of the existing plants and also form a benchmark for the installation of new facilities. The objective of this paper is to carry out a complete thermodynamic evaluation of one of the major GTs located at Geregu through energy and exergy analysis.

2. Methodology

In order to analyze the energetic and exergetic performance of the power plant, thermodynamic models are applied to investigate each component of the power plant. Geregu 1 thermal power station Ajaokuta, Nigeria, is an open cycle gas turbine plant that runs on natural gas. It consists of three independent units, each rated at 138 MW giving a total installed capacity of 414 MW. The design parameters and the composition of the natural gas used are shown in Table 1 and Table 2 respectively while the operating process is shown schematically in Figure 1. The model equations were analyzed using high performance software, SCILAB code.

As an open cycle GT, air enters the compressor at a pressure of 1bar and at temperature of 15˚C and is compressed to a pressure and temperature of 11bar and 350˚C respectively. Heat addition takes place in the combustion chamber at constant pressure and temperature of 1060˚C by the combustion of the fuel. The exhaust gas exits to the atmosphere after useful work is done in the turbine. The combustion products are discharged to the atmosphere at a pressure of 1 bar and temperature 540˚C respectively. The component wise energy and exergy analyses of the system are presented in this section with the aid of Figure 1.

Table 1. Geregu gas turbine design conditions.

Parameter	Value	Unit
Compressor inlet pressure	1	bar
Compressor outlet pressure	11	bar
Compressor inlet temperature	15	˚C
Compressor outlet temperature	350	˚C
Compressor mass flow rate	500	kg/s
Fuel mass flow rate to the combustion chamber	9	kg/s
Combustion inlet temperature	350	˚C
Combustion chamber outlet temperature	1060	˚C
Combustion chamber pressure	11	bar
Turbine inlet temperature	1060	˚C
Turbine outlet temperature	540	˚C
Turbine inlet pressure	11	bar
Turbine outlet pressure	1	bar
Turbine mass flow rate	509	kg/s
Exhaust temperature	530	˚C
Exhaust pressure	1	bar
Power output	138	MW
Relative humidity	51.5	%
Compressor stages	10	-
Turbine stages	4	-
Power turbine speed	3600	rpm

Table 2. Composition of natural gas supply to Geregu gas turbine plant.

Component	Molecular Formula	Percentage by mole
Carbon dioxide	CO_2	0.61
Nitrogen	N_2	1.19
Methane	CH_4	93.56
Ethane	C_2H_6	4.03
Propane	C_3H_8	0.11
I-butane	C_4H_{10}	0.12
N-butane	C_4H_{10}	0.01
Neo-Pentane	C_5H_{12}	0.02
I-pentane	C_5H_{12}	0.08
N-pentane	C_5H_{12}	0.01
Hexanes	C_6H_{14}	0.08
M C pentane	C_6H_{12}	0.02
Cyclo hexane	C_6H_{12}	0.02
Heptanes	C_7H_{16}	0.03
M C hexane	C_7H_{14}	0.02
Toluene	C_7H_8	0.03
Octane	C_8H_{18}	0.01
MP Xylene	C_8H_{10}	0.02
Nonanes	C_9H_{20}	0.02
Decanes	$C_{10}H_{22}$	0.01
Gas density		0.7335 kg/m^3
Whole sample mole weight		17.28
Fuel Lower calorific Value		47,976.5 KJ/kg

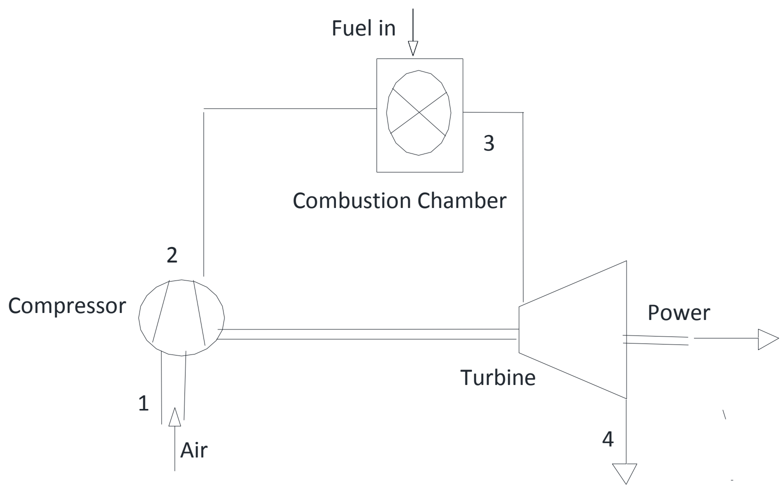

Figure 1. Schematic diagram of Geregu gas turbine power plant.

2.1. Mass and Combustion Equations

At several locations on the boundary through which mass enters and exits the plants, the mass balance at steady state is given by

$$\sum \dot{m}_i = \sum \dot{m}_e \quad (1)$$

where i and e represent inlet and exit respectively.

2.2. Energy Analysis of the Gas Turbine Plant Components

In an open flow system there are three types of energy transfer across the control volume namely work transfer, heat transfer, and energy associated with mass transfer in and out of the system. The first law of thermodynamics or energy balance for the steady flow process is

$$Q_{cv}+\sum_i \dot{m}_i\left(h_i+\frac{v_i^2}{2}+gz_i\right)=W_{cv}+\sum_e \dot{m}_e\left(h_e+\frac{v_e^2}{2}+gz_e\right) \tag{2}$$

To analyze the possible realistic performance, detailed energy analysis of the thermal power plant system has been carried out by ignoring the kinetic and potential energy changes. Here, we will apply all the fundamental principles required for the thermodynamic analysis of power generating systems (such as conservation of mass and conservation of energy principle and thermodynamic data) to individual plant component such as the compressor, the combustion chamber and the turbines to get their individual contribution to the overall power plant.

The energy or the first law efficiency η_1 of a system and/or system component is defined as the ratio of energy output to the energy input to a system/component.

$$\eta_1=\frac{\text{Desired Output energy}}{\text{Input energy}} \tag{3}$$

2.2.1. Energy Analysis of Compressor Sub-System

All the components shown in **Figure 1** are analyzed individually. For the compressor, there is no heat transfer, $Q_{cv}=0$, then the energy balance for the work intake by the air compressor subsystem is given by Equation (4)

$$W_C=\frac{\dot{m}_c\left(h_{2s}-h_1\right)}{\eta_{mc}\eta_c} \tag{4}$$

The energy or first law efficiency of the compressor sub-system is calculated from Equation (5)

$$\eta_{1,C}=\frac{\dot{m}_c\left(h_{2s}-h_1\right)}{\eta_{mc}W_{Comp.}} \tag{5}$$

2.2.2. Energy Analysis of Combustion Chamber Sub-System

At the combustion chamber, there is no work interaction in the system, the heat input into the system is given by Equation (6),

$$Q_{cv}=\dot{m}_f * LCV\,; \tag{6}$$

LCV is the Lower Calorific Value of the fuel.

The energy or first law efficiency of the combustion chamber is

$$\eta_{1,cc}=\frac{\dot{m}_t\left(h_3-h_2\right)}{\dot{m}_f * LCV} \tag{7}$$

2.2.3. Energy Analysis of Turbine Sub-System

For the turbine, the work developed by the sub-system is given by Equation (8)

$$W_T=\dot{m}_t\eta_{mt}\left(h_3-h_{4s}\right) \tag{8}$$

The first law efficiency of the turbine sub-system is calculated from Equation (9)

$$\eta_{1,T} = \frac{W_T}{\dot{m}_t \eta_{mt} \left(h_3 - h_{4s}\right)} \tag{9}$$

2.2.4. Gas Turbine Cycle Thermal Efficiency

The network W_{net} of the gas turbine plant is given by the equation

$$W_{net} = W_T - W_C \tag{10}$$

The thermal efficiency of the gas turbine cycle is determined using the equation,

$$\text{Cycle thermal efficiency} = \frac{W_{net}}{\eta_{cc} \dot{m}_f LCV} \tag{11}$$

The overall gas turbine power plant efficiency is thus

$$\text{overall power plant efficiency} = \frac{\eta_g W_{net}}{n_{cc} \dot{m}_f LCV} \tag{12}$$

2.3. Exergy Analysis of Gas Turbine Plant

Exergy is the maximum theoretical work or reversible work obtained as a system interacts with an equilibrium state. It is generally not conserved as energy but destroyed in the system. Exergy destruction is the measure of irreversibility which is the source of performance loss. Therefore an exergy analysis in assessing the magnitude of exergy destruction identifies location, magnitude and the source of thermodynamic inefficiencies in thermal systems [20].

The exergy efficiency or second law efficiency is defined as

$$\eta_2 = \frac{\text{Exergy Output}}{\text{Exergy Input}} \tag{13}$$

Formulation of Exergy Balance Equation:

The general exergy balance equation comprising both chemical exergy and thermo-mechanical exergy is given by the equation [21].

$$\begin{aligned} E^W &= E^{CH} + \left\{\sum\nolimits_{in} E_{in}^T - \sum\nolimits_{out} E_{out}^T\right\} + \left\{\sum\nolimits_{in} E_{in}^P - \sum\nolimits_{out} E_{out}^P\right\} \\ &\quad + T_0 \left\{\sum\nolimits_{in} S_{in} - \sum\nolimits_{out} S_{out} + \frac{Q_{cv}}{T_0}\right\} \end{aligned} \tag{14}$$

where,

E^W = Exergy or the maximum theoretical work developed by the system.

E^{CH} = Chemical exergy of the system.

$E_{in}^T - E_{out}^T$ = Change in thermal component of exergy as stream enters and exits the plant.

$E_{in}^P - E_{out}^P$ = Change in mechanical component of exergy as stream enters and exits the plant.

$\sum_{in} S_{in} - \sum_{out} S_{out}$ is the change in entropy as the stream enters and exits the plant.

Q_{cv} is the heat transfer between the plant and the environment.

The thermo-mechanical stream can be separated into thermal and mechanical components. The thermal and mechanical components of the thermo-mechanical exergy stream for an ideal gas with constant specific heat is written as [22]

$$E^T = \dot{m}c_p\left[\left(T - T_0\right) - T_0 \ln\left(\frac{T}{T_0}\right)\right] \tag{15}$$

$$E^P = RT_0 \ln\left(\frac{P}{P_0}\right) \tag{16}$$

The thermo-mechanical exergy stream of the ideal gas equation is given by Equation (17)

$$E^M = E^T + E^P = \dot{m}c_p\left(T - T_0\right) - T_0\left[\dot{m}c_p \ln\left(\frac{T}{T_0}\right) - R\ln\left(\frac{p}{P_0}\right)\right] \tag{17}$$

The c_p in Equation (17) can be obtained in a polynomial form as

$$c_p = \frac{\bar{C}_p}{M} = a + bT + cT^2 + dT^3 \tag{18}$$

where *a*, *b*, *c*, and *d*, are constants characteristics of gas obtained from selected ideal gas tables.

Chemical exergy for a mixture of ideal gas as written in Equation (19).

$$E^{CH} = \dot{m}\left(\sum_i y_i \bar{e}_i^{ch} + RT_0 \sum_i y_i \ln y_i\right) \tag{19}$$

The component wise exergy balance of gas turbine thermal power plant system is given as follows sections.

2.3.1. Exergy Analysis of Compressor Sub-System

The exergy balance for the air compressor system is given by

$$-W_C = \dot{m}_a\left(e_1 - e_2\right) - T_0\dot{\sigma} \tag{20}$$

The process irreversibility in the compressor is given by

$$I_{process} = T_0\sigma = T_0\left[\dot{m}\left(s_2 - s_1\right)\right] \tag{21}$$

where the entropy generated at each state in the compressor is given as in Equation (22)

$$s_2 - s_1 = C_{p_a} \ln\left(\frac{T_2}{T_1}\right) - R_a \ln\left(\frac{P_2}{P_1}\right) \tag{22}$$

The total irreversibility or exergy destruction in the compressor consists of two components, the mechanical irreversibility and process irreversibility [23].

$$I_{M,C} = \left\{\frac{1}{\eta_{mc.}} - 1\right\} E_{2-1comp.} \tag{23}$$

where

$E_{2-1comp.}$ = Internal power requirement of the compressor,

$$E_{2-1comp.} = \dot{m}_{air}\left(h_2 - h_1\right) \tag{24}$$

$\eta_{mc.}$ = compressor mechanical efficiency ≈ 0.99

The exergy destruction in the compressor

$$E_{D,C} = I_{M,C} + I_{P,C} \tag{25}$$

The second law efficiency of the compressor is

$$\eta_{2,C.} = 1 - \frac{\dot{E}_{D,C}}{W_C} = \frac{\dot{m}_a\left(e_2 - e_1\right)}{W_C} \tag{26}$$

2.3.2. Exergy Analysis of the Combustion Chamber Sub-System

The exergy balance for the combustion chamber sub system is given by the equation

$$0 = \sum_j \left[\left(\dot{m}e\right)_{f+a} - \left(\dot{m}e\right)_p\right] - T_0\sigma \tag{27}$$

where

m_{f+a} is the sum of the mass of fuel and air, e is the specific exergy and m_p is that of combustion products.

The irreversibility in the combustion chamber is given as

$$T_0\dot{\sigma} = \left(S_P\right)_3 - \left(S_R\right)_2 \tag{28}$$

and the entropy production rate is

$$\dot{\sigma} = \dot{m}\left[\left(s_P\right)_3 - \left(s_R\right)_2\right]$$

where $\left(S_R\right)_2 = \left(S_A\right)_2 + \left(S_F\right)_0$ and the subscripts P, R, A and F represents products, reactants, air and fuel respectively [24]. Combustion in the combustion chamber is assumed to take place at constant pressure. The irreversibility in the combustion chamber is

$$E_{D,cc} = T_0\left\{\left[\left(S_P\right)_3 - \left(S_P\right)_0\right] + \left(S_P\right)_0 - \left[\left(S_A\right)_2 - \left(S_A\right)_0 + \left(S_F\right)_0 + \left(S_A\right)_0\right]\right\}$$

where $\Delta S_0 = \left(S_P\right)_0 - \left[\left(S_F\right)_0 + \left(S_A\right)_0\right]$

$$E_{D,cc} = T_0\left\{\left[\dot{m}_g c_{p_g} \ln\frac{T_3}{T_0} - \dot{m}_g R_g \ln\frac{P_3}{P_0}\right] - \dot{m}_a c_{p_a} \ln\frac{T_2}{T_0} - \dot{m}_a R_a \ln\frac{P_2}{P_0} + \Delta S_0\right\} \tag{29}$$

The second law efficiency of the Combustion Chamber is

$$\eta_{2,CC} = 1 - \frac{E_{D,CC}}{\left(\dot{m}e\right)_{f+a}} = \frac{\left(\dot{m}e\right)_p}{\left(\dot{m}e\right)_{f+a}} \tag{30}$$

2.3.3. Exergy Analysis of Turbine Sub-System

The exergy balance for the work developed by the gas turbine sub-system is given by the equation

$$W_T = \dot{m}_p\left(e_3 - e_4\right) - T_0\sigma \tag{33}$$

where $\dot{m}_p\left(e_4\right)$ is exergy out of the turbine subsystem and $\dot{m}_p\left(e_3\right)$ is exergy in into the turbine subsystem.

The process irreversibility in the turbine is given by

$$I_{P,T} = T_0\sigma = T_0\left[\dot{m}_p\left(s_3 - s_4\right)\right] \tag{34}$$

The entropy changes at different states in the turbine is given by the Equation (35)

$$s_3 - s_4 = C_{p_g} \ln\left(\frac{T_3}{T_4}\right) - R_g \ln\left(\frac{P_3}{P_4}\right) \tag{35}$$

The internal power generated in the turbine is determined using the equation

$$P_T = \dot{m}_p \left(h_3 - h_4\right) \tag{36}$$

The mechanical irreversibility in the turbine is given by the equation

$$I_{M,T} = \left(1 - \eta_{mt}\right) P_T \tag{37}$$

Exergy out from the turbine sub system can be evaluated using Equation (38)

$$\dot{m}_p \left(e_4\right) = \left[\dot{m}_p \left(e_3\right) - \left\{\left(h_3 - h_4\right) - T_0 \left(s_3 - s_4\right)\right\}\right] \tag{38}$$

Exergy destruction in the turbine is given by the equation

$$E_{D,T} = I_{M,T} + I_{P,T} \tag{39}$$

The second law efficiency of the turbine sub system is given by the equation

$$\eta_{2,T} = 1 - \frac{E_{D,T}}{\dot{m}_p \left(e_3 - e_4\right)} = \frac{W_T}{\dot{m}_p \left(e_3 - e_4\right)} \tag{40}$$

2.3.4. The Gas Turbine Cycle Exergy Efficiency

The cycle second law or exegetic efficiency is given by

$$\eta_{2,cycle} = \frac{W_{net}}{Ex_{fuel}} \tag{41}$$

The overall power plant second law or exegetic efficiency is evaluated using the equation

$$\eta_{2(overall)} = \frac{\eta_g W_{net}}{Ex_{fuel}} \tag{42}$$

The thermochemical and standard chemical exergy of the natural gas supply to the power plant which are used in the exergy analysis are presented in Table 3.

Table 3. Thermochemical and standard chemical exergy of natural gas supplied to Geregu power plant.

Component	Mole fraction, y	Heat of formation h_f^0 (KJ/kmol)	Gibbs function (KJ/kmol)	Standard chemical exergy (KJ/kmol)
Carbon dioxide	0.006	−393,520	−394,380	19,870
Nitrogen	0.0119	0	0	720
Methane	0.9356	−74,850	−50,790	831,680
Ethane	0.0403	−84,680	−32,890	1,495,955
Propane	0.0011	−103,850	−23,490	2,131,880
I-butane	0.0012	−134,300	−20,760	2,800,835
N-butane	0.001	−126,150	−15,710	2,805,885
Neo-Pentane	0.0002	−167,800	−14,050	3,453,920
I-Pentane	0.0008	−154,400	−14,050	3,453,920

Continued

N-Pentane	0.0001	−146,440	−8200	3,459,770
Hexanes	0.0008	−166,920	150	4,114,495
MC Pentane	0.0002	−106,700	−8200	−3,459,770
Cyclo hexane	0.0002	−123,141	31,800	3,910,050
Heptanes	0.0003	−187,780	8165	4,768,855
M C hexane	0.0002	−154,700	27,480	4,141,825
Toluene	0.0003	50,170	122,050	3,938,390
Octane	0.0001	−208,450	17,320	5,424,451
MP Xylene	0.0002	17,300	118,760	4,581,425
Nonanes	0.0002	−229,300	11,900	6,065,370
Decanes	0.0001	−249,400	17,400	6,717,245

3. Results and Discussion

The 138 MW Geregu 1 gas turbine power plant was analyzed in terms of energy and exergy. For exergy analysis, the environment reference temperature and pressure of 25˚C and 1 bar respectively were considered. The thermodynamic properties of the input and output of the unit of Geregu gas turbine plant for both design and operating conditions were determined using Engineering Equation Solver (EES) software package. With the help of these properties, energy and exergy analyses of the gas turbine plant were performed based on the first and second laws of thermodynamics [25]. The components energy and exergy efficiencies, the gas turbine cycle efficiency and the second law efficiency were obtained. Also, the exergy destruction efficiencies of the system components was noted as a result of performing the analysis. The needed thermodynamic properties of the power plant both for design and operating conditions were fed into the formulated energy and exergy model equations for the components and analyzed using SCILAB software code [26]. The energy efficiency of the compressor sub-system at the design condition was 84.00%, whereas for the actual operating years, it ranges from 92.35% to 93.20%. Similarly, the exergy efficiency of the compressor at the design condition was 89.05%, and ranges from 92.83% to 93.19% for the operating years. The combustion chamber has energy efficiency of 96.96% at design and in the range of 98.60% to 99.50% for actual operating years. The exergy efficiency of the combustion chamber at design condition was 74.29%, and its efficiency ranges from 73.52% to 73.70% for the operating years. Also, the turbine sub-system energy efficiency at design condition was 87.49% and in the range of 87.02% to 87.61% for operating years. For the turbine, the exergy efficiency at the design condition was 93.60%. Its exergy efficiency for the operating years were in the range of 93.40% to 93.65%. The profile of energy and exergy efficiencies of the components of the power plant is illustrated in the bar chart shown in **Figure 2** and **Figure 3**.

The exergy destruction efficiency of the compressor at design was 10.94%, and

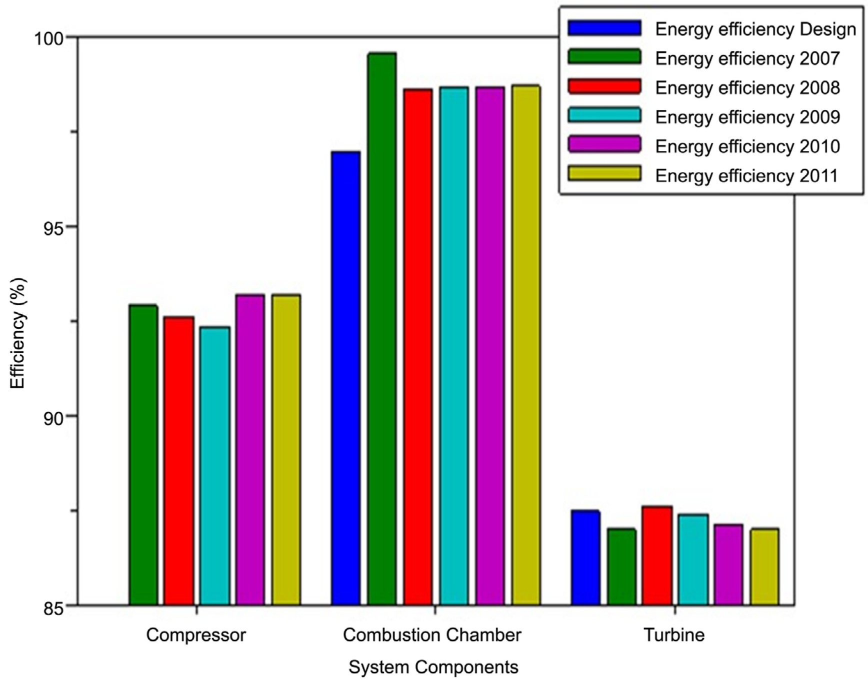

Figure 2. Energy efficiency of components at design and operating years.

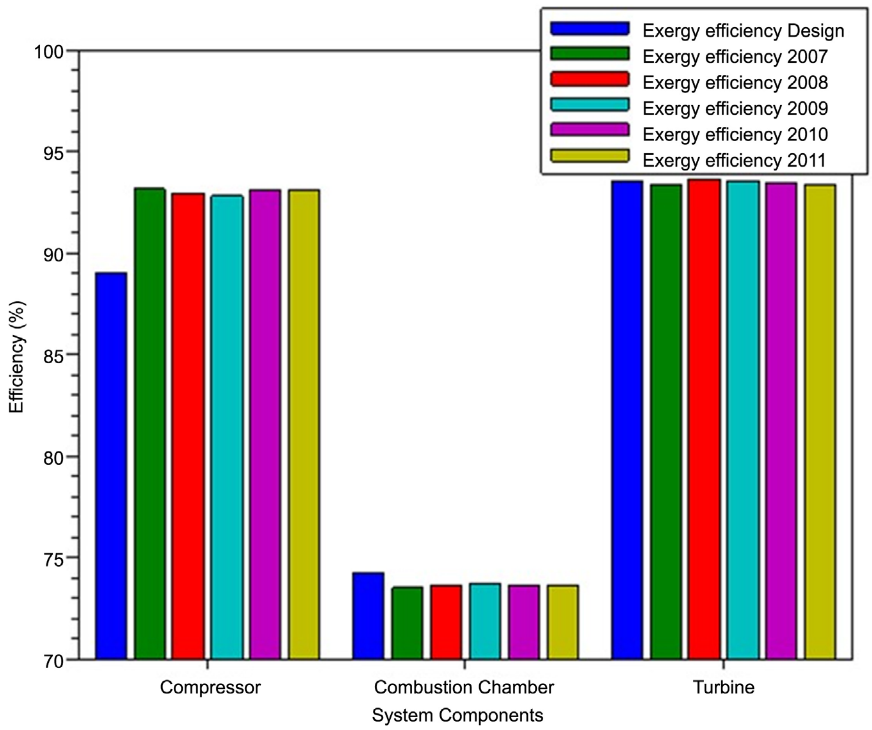

Figure 3. Exergy efficiency of components at design and operating years.

the efficiency at operating years ranges from 6.80% to 7.17%. Similarly, the exergy destruction efficiency of combustor chamber sub-system at design condition was 25.71%, while it ranges from 26.30% to 26.48% for actual operating years. The turbine exergy destruction efficiency was 6.40% at design and it was in the range of 6.35% to 6.60% for actual operating years. The thermal efficiency and the second law efficiency of the power plant at design condition were 31.36% and 29.13% respectively. For the actual operating years, it ranges from 32.60% to 33.04%, and 30.28% to 30.68%. The overall power plant thermal efficiency and the second law efficiency are also shown in Table 4.

A comparison of exergy destruction of power plant components as illustrated in Figure 4, shows that the highest exergy destruction occur in the combustion chamber with exergy destruction efficiency in the range of 25.71% to 26.48% for the years under study. The exergy loss in the combustion chamber is due to combustion reaction and large temperature difference during heat transfer between the

Table 4. Result of component efficiencies at design and operating years.

Component	Design	2007	2008	2009	2010	2011
Compressor						
Work input	176,767.68	172,147.84	171,468.94	171,964.44	171,017.76	170,847.05
Isentropic work input (kW)	148,484.85	159,924.32	158,823.74	1,558,814.22	159,380.45	159,221.36
Internal power consumed	175,000	1,740,426.36	169,754.25	170,244.8	169,307.58	169,138.58
Energy efficiency, η_1 (%)	84.00	92.90	92.63	92.35	93.20	93.20
Exergy efficiency, η_2 (%)	89.06	93.20	92.98	92.83	93.15	93.17
Exergy destruction efficiency, δ_c (%)	10.94	6.80	7.02	7.17	6.85	6.83
Exergy destruction (kW)	19,340.74	11,706.09	12,033.49	12,322.83	11,716.72	11,666.99
Exergy input (kW)	176,767.68	172,147.84	171,468.94	171,964.44	171,017.76	170,847.05
Exergy output (kW)	157,426.94	160,441.75	159,435.44	159,641.61	159,301.04	159,180.05
Combustion Chamber						
Heat input	431,788.5	421,233.67	421,713.44	421,233.67	421,713.43	420,274.14
Energy efficiency, η_1 (%)	96.60	99.53	98.60	98.65	98.66	98.70
Exergy efficiency, η_2 (%)	74.29	73.52	73.69	73.70	73.66	73.66
Exergy destruction efficiency, δ_{CC} (%)	25.71	26.48	26.31	26.30	26.34	26.34
Exergy destruction (kW)	160,023.61	162,598.81	161,425.72	161,274.82	161,523.62	161,105.95
Exergy input (kW)	622,310.00	613,961.00	613,471.23	613,160.87	613,336.83	611,652.8
Exergy output (kW)	462,286.39	451,362.19	452,045.51	451,886.04	451,813.21	450,546.85
Turbine						
Work output	308,053.28	308,817.97	308,842.17	308,144.04	307,317.36	306,108.60
Isentropic work (kW)	352,107.62	354,874.67	352,453.5	352,453.5	352,591.85	351,740.29
Internal power generated (kW)	311,164.93	311,937.35	311,961.79	311,256.61	310,421.57	309,200.61
Energy efficiency, η_1 (%)	87.48	87.02	87.62	87.43	87.16	87.03
Exergy efficiency, η_2 (%)	93.60	93.39	93.65	93.58	93.48	93.40
Exergy destruction efficiency δ_{Turb} (%)	6.40	6.61	6.35	6.42	6.52	6.60
Exergy destruction, δ_{Turb} (kW)	21,061.70	21,825.04	20,932.31	21,150.09	21,441.97	21,645.88
Exergy input (kW)	329,114.98	330,643.01	329,774.48	329,294.13	328,759.32	327,754.48
Exergy output (kW)	308,053.28	308,817.97	308,842.17	308,144.04	307,317.36	306,108.60
GT cycle						
Net work (kW)	131,285.61	136,670.13	137,373.23	136,179.6	136,299.6	135,261.55
Power generated (kW)	128,659.90	133,936.73	134,625.77	133,456.00	133,573.61	132,556.32
Energy efficiency, η_1 (%)	31.36	32.60	33.04	32.77	32.76	32.61
Exergy efficiency, η_2 (%)	29.13	30.28	30.68	30.44	30.43	30.29
Overall GT Plant						
Energy efficiency, η_1 (%)	30.73	31.95	32.38	32.11	32.10	31.95
Exergy efficiency, η_2 (%)	28.54	29.67	30.07	29.83	29.82	29.68

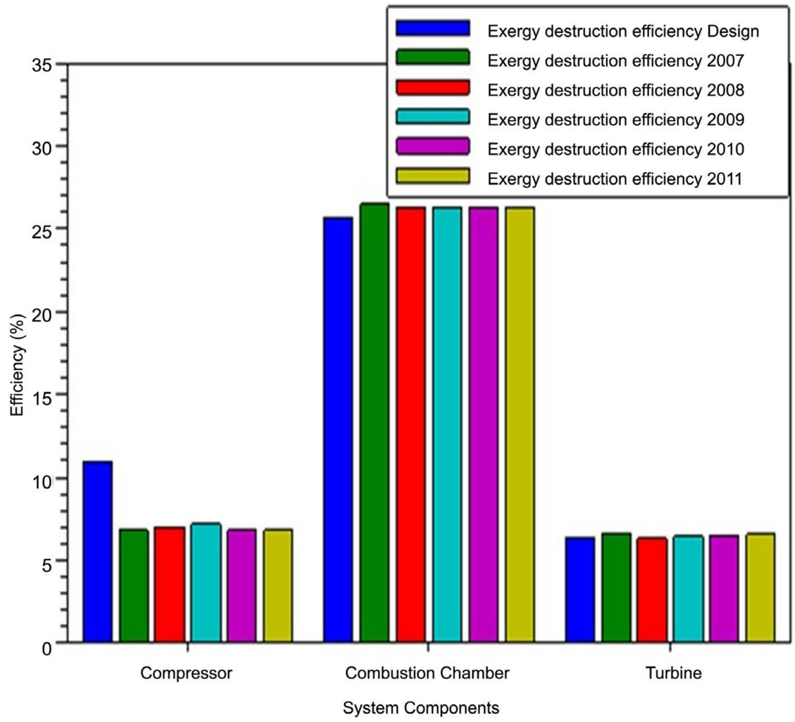

Figure 4. Exergy destruction efficiency of components at design and operating years.

combustion gases and air/fuel mixture. Other factors that may contribute to the high amount of irreversiblity are defective burners and fuel quality.

Improvement Potential of the Combustion Chamber of the Gas Turbine Plant

In the plant, the combustion chamber has the largest exergy destruction efficiency from the study accounting for the range of 25.71% to 26.48% of the breakdown exergy of the plant components. This, of course, has the largest improvement potentials. The chemical reaction between the air and fuel in the combustion process is the main source of exergy destruction in the combustion chamber. An improvement approach was investigated to determine its effect on the exergy efficiency of the components. The improvement performance approach considered was increasing turbine inlet temperature at constant pressure ratio. In the simulation, the turbine inlet temperature was increased from 1060˚C to 1080˚C with an interval of 5˚C. For each temperature, the pressure ratio remained constant at 11 bar. The result of the effect of increase in turbine inlet temperature at constant combustion chamber pressure ratio on the exergy and exergy destruction efficiencies are shown in **Table 5**. It shows a progressive exergy efficiency reduction as temperature increases while the exergy destruction increased proportionately with temperature. Similarly, for a given temperature and pressure evaluation, the shift in exergy efficiency from design to operating years was positive continually which imply that age produced negative effect on the operation of the plant. This is possible because of deterioration of parts as years advanced. This performance effect on the exergy is further illustrated in **Figure 5** where the

Table 5. Effect of turbine inlet temperature increase at constant pressure ratio (P = 11 bar) on exergy efficiency of the combustion chamber.

TIT (˚C)	Parameter	Year					
		Design	2007	2008	2009	2010	2011
1060	Exergy Efficiency (%)	74.29	73.52	73.69	73.70	73.66	73.66
	Exergy Destruction Efficiency, γ (%)	25.71	26.48	26.31	26.30	26.34	26.34
1065	Exergy Efficiency (%)	74.17	73.40	73.57	73.58	73.55	73.54
	Exergy Destruction Efficiency, γ (%)	25.83	26.44	26.43	26.42	26.45	26.46
1070	Exergy Efficiency (%)	74.06	73.29	73.46	73.47	73.43	73.43
	Exergy Destruction Efficiency, γ (%)	25.94	26.71	26.54	26.53	26.43	26.57
1075	Exergy Efficiency (%)	73.95	73.18	73.35	73.36	73.32	73.32
	Exergy Destruction Efficiency, γ (%)	26.05	26.82	26.65	26.64	26.68	26.68
1080	Exergy Efficiency (%)	73.92	73.11	73.28	73.29	73.26	73.24
	Exergy Destruction Efficiency, γ (%)	25.08	26.89	26.72	26.71	26.74	26.76

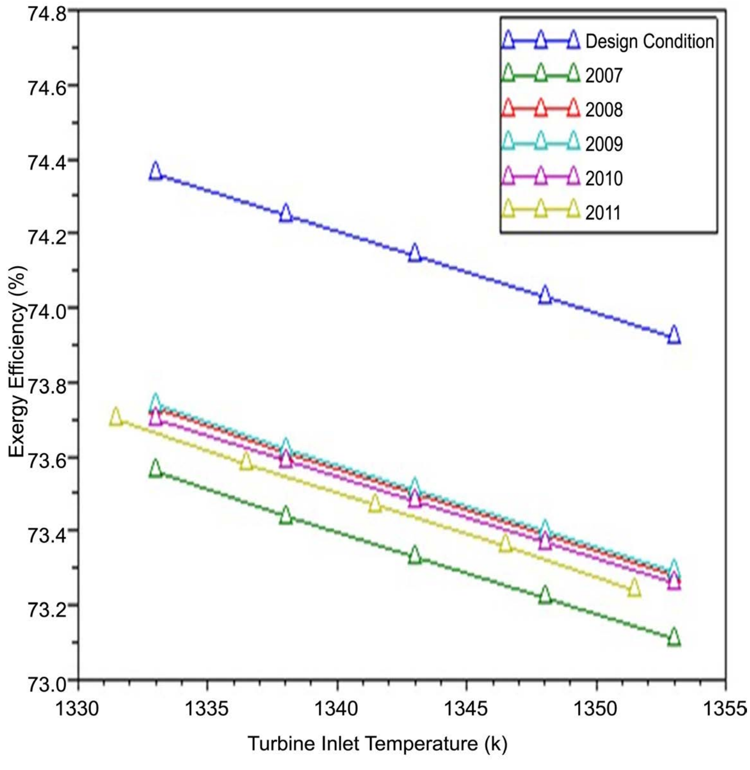

Figure 5. Effect of increase in turbine inlet temperature (K) at constant pressure ratio on exergy efficiency.

increase in TIT resulted to constant exergy efficiency decrement irrespective of the year of operation. That is the change in exergy efficiency from the reference state (P = 11 bar, T = 1060˚C) to subsequent states maintained constant value each year of operation as well as the design year. Similarly, the exergy destruction efficiency was also in the increase as temperature increased (**Table 5**).

The second improvement performance approach considered was increasing combustion chamber pressure ratio at constant turbine inlet temperature (TIT). The combustion chamber pressure ratio was increased from 11 bar to 15 bar at an interval of 1 bar while the temperature remained constant at 1060˚C. The result is shown in **Table 6**. It shows that increasing pressure ratio enhances the exergy of the system as the efficiency maintained a constant increase from the reference temperature and pressure.

Figure 6 shows the effect of increasing pressure ratio on the exergy efficiency

Table 6. Effect of pressure ratio increase on exergy efficiency of combustion chamber at constant turbine inlet temperature of 1060˚C.

Pressure (bar)	Parameter	Year					
		Design	2007	2008	2009	2010	2011
11	Exergy Efficiency (%)	74.29	73.52	73.69	73.70	73.66	73.66
	Exergy Destruction Efficiency, γ (%)	25.71	26.48	26.31	26.30	26.34	26.34
12	Exergy Efficiency (%)	74.46	73.70	73.87	73.88	73.85	73.84
	Exergy Destruction Efficiency, γ (%)	25.54	26.30	26.13	26.12	26.15	26.16
13	Exergy Efficiency (%)	74.66	73.91	74.08	74.09	74.05	74.05
	Exergy Destruction Efficiency, γ (%)	25.34	26.09	26.92	26.91	25.95	25.95
14	Exergy Efficiency (%)	74.78	74.04	74.22	74.21	74.18	74.16
	Exergy Destruction Efficiency, γ (%)	25.22	25.96	25.78	25.79	25.82	25.84
15	Exergy Efficiency (%)	74.90	74.16	74.34	74.34	74.31	74.30
	Exergy Destruction Efficiency, γ (%)	25.10	25.84	25.66	25.66	25.69	25.70

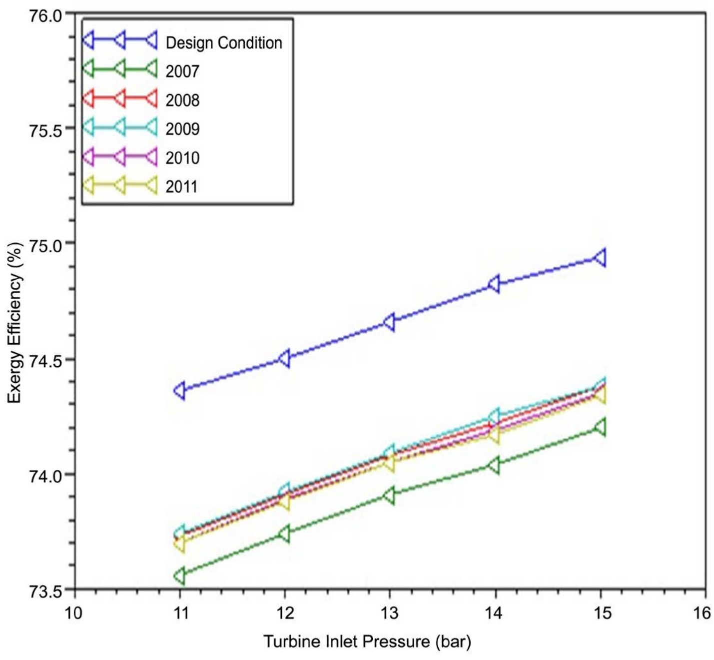

Figure 6. Effect of pressure ratio change on exergy efficiency of the combustion chamber at constant inlet temperature.

Table 7. Effect of changes in turbine inlet temperature and pressure on exergy efficiency of the combustion chamber.

Pressure (bar)/ TIT (˚C)	Parameter	Year					
		Design	2007	2008	2009	2010	2011
11/1060	Exergy Efficiency (%)	74.29	73.52	73.69	73.70	73.66	73.66
	Exergy Destruction Efficiency, γ (%)	25.71	26.48	26.31	26.30	26.34	26.34
12/1065	Exergy Efficiency (%)	74.35	73.59	73.76	73.77	73.73	73.72
	Exergy Destruction Efficiency, γ (%)	25.65	26.41	26.24	26.23	26.27	26.28
13/1070	Exergy Efficiency (%)	74.41	73.64	73.81	73.82	73.79	73.78
	Exergy Destruction Efficiency, γ (%)	25.59	26.36	26.19	26.18	26.21	26.22
14/1075	Exergy Efficiency (%)	74.48	73.69	73.88	73.91	73.85	73.83
	Exergy Destruction Efficiency, γ (%)	25.52	26.31	26.12	26.09	26.15	26.17
15/1080	Exergy Efficiency (%)	74.51	73.76	73.93	73.94	73.91	73.90
	Exergy Destruction Efficiency, γ (%)	25.49	26.24	26.07	26.06	26.09	26.10

of the combustion chamber indicating positive exergy efficiency gradient. This implies that whereas temperature increase reduces the exergy efficiency, the pressure enhances it. On the other hand, the exergy destruction efficiency reduced in the same range for both design and operating conditions.

The resultant effects of increasing both turbine inlet temperature and pressure ratio from 1060˚C to 1080˚C and 11 bar to 15 bar respectively, on the exergy efficiency of the combustion chamber is presented in Table 7. Exergy efficiency improvement range of 0.22% to 0.25% was recorded for the power plants turbine combustion chamber. It also resulted to the reduction in exergy destruction of the component in the same range for both design and operating conditions. The determining parameter in the improvement of the system as both inlet temperature and pressure increased, is the combustion chamber pressure ratio. Increase in pressure ratio has more effect than increase in turbine inlet temperature.

4. Conclusion

In this study, the performance improvement of Geregu 1 gas turbine power plant in Nigeria was performed by exergy analysis. The primary objective of this study was to analyze the system components separately and to identify the sites having the largest exergy losses and to quantify the amount of losses. Specific data at design and operating conditions from Geregu1 gas turbine plant in Nigeria was used. The maximum exergy destruction efficiency was found in the combustion chamber to be in the range of 25.71 to 26.48 percent for the design and operating years under review. This is because of chemical reaction and heat transfer processes in the combustion chamber. Improvement on the performance of the combustion chamber made included 1) Increasing turbine inlet temperature at constant pressure ratio. In this, the turbine inlet temperature was increased from 1060˚C to 1080˚C with an interval of 5˚C in temperature and the pressure ratio remained constant at 11 bar. There is decrease in exergy efficiency in the range of 0.37 percent to 0.45 percent for both design and operating conditions. 2) Increasing combustion chamber pressure ratio at constant turbine inlet temperature (TIT). In the simulation, the combustion chamber pressure ratio was increased from 11 bar to 15 bar with an interval of 1 bar pressure increase and the turbine inlet temperature remained constant at 1060˚C. The improvement analysis indicates increase in exergy efficiency in the range 0.61 percent to 0.65 percent. On the other hand, the exergy destruction efficiency reduced in the same range for both design and operating conditions. 3) Increasing both turbine inlet temperature and pressure ratio. Improvement on the performance of the combustion chamber with both increase in turbine inlet temperature (TIT) from 1060˚C to 1080˚C at 5˚C interval, and pressure increase from 11bar to 15bar at interval of 1 bar showed exergy destruction efficiency reduction in the range of 0.22 percent to 0.25 percent. This efficiency reduction in exergy destruction resulted to combustion chamber exergy efficiency increase of 74.29 percent to 74.51 percent from initial design pressure and temperature of 11 bar and 1060˚C to 15 bar and 1080˚C respectively. This also resulted in performance improvement in exergy efficiency in the range of 73.76 percent to 73.93 percent from the range of 73.52 percent to 73.70 percent for the operating years after the improvement. The performance improvement was made by taking note of the temperature limit of 1700K (1427C), (Moran and Shapiro, 2006) imposed by the metallurgical considerations on the maximum permissible temperature at the TIT. The cycle thermal and exergy efficiency of the plant at design were 31.36 percent and 29.13 percent respectively, while it ranges from 32.60 percent to 33.04 percent for the operating years. The overall energy and exergy efficiency of the plant at design were 30.78 percent and 28.54 percent and the overall energy and exergy efficiencies ranges from 30.28 percent to 30.68 percent for the operating years. In conclusion, energetic and exergetic analysis and results obtained from this study will guide engineers and researchers on the area to focus on the plant performance improvement

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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