Exergy and Environmental Assessment of a Steam Power Plant: A Case Study of Nigeria

M.N. Eke, D.C. Onyejekwe, O.V. Ekechukwu and C.C. Maduabuchi[*]

Department of Mechanical Engineering, University of Nigeria, Nsukka, Nigeria.

Abstract: Many electricity generating stations are concerned with the reduction of environmental pollution associated with the thermodynamic activities of power plants. Such environmental pollution includes emissions from exhaust gases, cooling tower blowdown, boiler blowdown and demineralization. In this paper, an exergo-environmental analysis was conducted using design data from the Egbin power plant for a 220MW steam power plant. Enhancement was carried out on the plant under varying pressure and temperature conditions to assess the plant's performance improvements that would lead to more reduction in environmental pollution. The exergy destruction efficiency value indicates that the boiler sub- system gave the highest exergy destruction in the power plant. Also, sustainability indicators such as environmental effect factor, waste exergy ratio and sustainability index factors have been performed and results presented with respect to the plant. The improvement options considered were: (i) increasing the inlet temperature of the high-pressure turbine at constant boiler pressure, and (ii) the second approach, simultaneous increase in inlet temperature of high-pressure turbine and boiler pressure. The result showed that the second improvement approach gave a better improvement approach than the former by reducing the environmental effect factor by 17.32% and increasing the sustainability index factor by 21.54%. These effects ultimately reduced the steam power plant emissions and improved efficient fuel utilization by the plant for sustainable development and for more power production.

Keywords: Exergy, Exergy destruction efficiency, Waste exergy ratio, Environmental pollution, Sustainability.

1. INTRODUCTION

Today's world major concerns are how to provide energy security, reduce CO_2 and other hazardous gaseous emissions. Improving energy system efficiency is an important option for future energy security and the elimination of harmful gaseous emissions. With civilization growing prosperity our energy consumption is growing very rapidly. Fossil combustibles, especially coal and natural gas, provide the bulk of the world largest electricity generation. Despite the rise of clean energy such as wind and solar power, it is expected that the strong reliance on fossil fuels will continue for decades [1]. Fossil fuels remain the country's dominant primary energy source, with their uses as a stable primary energy source in power generation, transport and manufacturing. Today, thermal power plants are among the most widely used plants in the world for electricity generation. Thermal power plants are the kind of systems that turn the chemical energy stored in solid, liquid and gaseous fuels into thermal energy, which is then converted into electrical energy, [2]. Thermal power plants include steam, gas turbine, nuclear power etc. Despite the rapid development of clean energy technology, their marginal costs and the present state of the art have not for some time, it is important that fossil fuel plants and their effects on the environment be

reduced by working more efficiently. Globally, fossil-fueled thermal power stations release a large share of man-made CO_2 emissions into the atmosphere, and efforts to minimize these emissions are complicated, varied and expensive [3]. Power plants emit particles directly into the air, lakes, lagoons, rivers and oceans. The main contributor to air pollution is SO_2 and NOx, pollutants, which are converted into sulfate and nitrate particles in the atmosphere. It emits specific kinds of waste such as liquid wastes, and pollutants of gases. These pollutants can impair human health where people are affected by the exhaust gas causing respiratory illness and the environment leading to dirty or dull walls, dirty tin roof as it also causes a rapid leak, [4]. In a thermal power plant, the conversion of energy is dominantly a thermodynamic operation. Improving energy efficiency in a thermodynamic process is usually based on energy analysis. The conventional approach to energy analysis is based on the first law of thermodynamics that focuses on conserving energy. The drawback of this study is that the depletion of energy content in the system is not taken into account. That is, it's not representative of the irreversibility of the system. Moreover, an interpretation of the first law sometimes casts false opinions on the efficiency of an energy conversion device, [5]. In order to get higher efficiency, therefore, requires exergy analysis that is based on the second thermodynamic law. This study helps us to understand the main cause of failure, and the need to improve performance. In addition to assessing the magnitudes, location and causes of

*Address correspondence to this author at the Department of Mechanical Engineering, University of Nigeria, Nsukka, Nigeria;
E-mail: chika.maduabuchi.191341@unn.edu.ng

plant's irreversibilities, the exergetic performance analysis can provide a more accurate measurement of the efficiency of the individual plant components. These aspects of the exergetic efficiency analyses are the underlying variation from energy analysis. Analysis of exergy is a powerful tool to explain the difference between energy losses in plants and intrinsic irreversibility [6]. Exergy can be defined as the maximum possible work that can be produced, as it is brought into balance with a given reference environment. Thus, exergy can be calculated as a measure of energy efficiency or utility (capacity to impact desired change). Exergy is not conserved, as opposed to energy, but it is destroyed in any practical process. Exergy key research goal is to determine where the exergy is destroyed. The exergy destruction during a process is proportional to the entropy generation in it which accounts for system inefficiencies due to irreversibility, [5]. Practical systems requiring energy conversion and transfer often comply with the law of conversion of energy, but the efficiency of energy degrades i.e. capacity for work is lost or exergy is destroyed. Exergy analysis helps to identify irreversibilities of the system that lead to loss of useful workability and thus identify areas where improvements can be sought, [7]. Degradation of the energy content is equal to the irretrievable exergy destruction due to the irreversibility of all the real processes. Many causes of irreversibilities include friction, heat transfer, mixing and throttling [8]. Exergy destruction provides a quantitative indicator of the inefficiency of the system. Destruction of exergy often contributes to thermal ambient emission. This energy generated pollution contributes to a range of human health and environmental issues. SO_2 and NOx combine with water to contaminate it as absorbed into the atmosphere, and other chemicals to create different acidic compounds and ozone. The sources of water pollution among other things include cooling tower blowdown, boiler blow-down, demineralization, etc. When drained, waste-water at higher temperatures will damage organisms' local aquatic life. Environmental risk is minimized by maintaining exergy by improved productivity (i.e. reducing as little as is required during a process), environmental damage is reduced, [9]. There is a broad body of researchers on the performance analyses of thermal power plants in relevant works of literature. Ehyaei *et al*. [10] studied the effect of an additional unit on the inlet of a typical power plant in Iran on the first and second law efficiencies. In addition, a new methodology for system optimization is proposed in their study. There are certain parameters within this new function that produce air pollution in the system, such as first law efficiency, energy costs and external costs.

Ultimately, it was observed that with the addition of a unit to the inlet of the plant, outlet power, first and second law efficiencies improved by 7%, 5.5%, 6% and 4%. In their work, Li and Liu [11] have defined comprehensive exergy losses of a 300MW power on the basis of the second law of thermodynamics. It has been established that with exergy analysis, the greatest exergy loss was found in the boiler unit of the plant. Ahmadi *et al*. [12] conducted thermodynamic simulation, exergy, exergo-economic analysis and optimization. It has been noted that the largest loss of exergy is in the boiler of the combined cycle plants. The explanation for this is due to extreme variation in temperature between combustion gasses and working fluid. Energy and exergy analysis studies for power generation systems are of scientific interest, as well as essential for the efficient use of energy resources. For this reason, exergy studies in recent years received a great deal of interest from scientists and system designers, some dedicated to studying component exergy analyses and improving performance. Exergy analysis has gained relatively widespread adoption in recent decades as a useful tool in the design, assessment, optimization and improve-ment of energy systems. Exergy analysis provides an effective technique for measuring and optimizing thermal system performance by accounting for energy quality. It can also be used for assessing energy sustainability levels.

A sustainable system by definition utilizes its resources efficiently which is achieved by reducing irreversible losses of the system [13]. Sustainability is necessary by evaluating certain sustainable indicators in order to overcome certain ecological problems that affect life and economic development. Sustainability means an energy resource supply that is available sustainably at a reasonable cost and does not cause minimal negative effects. Sustainability is required to address existing environmental, economic and growth challenges. Exergy sustainability indicators include exergy efficiency, waste exergy ratio, recoverable exergy rate, exergy destruction factor, environmental effect factor and exergy sustainability index, [13,14]. The study of efficient thermodynamic power cycle to increase its performance to get more significant power production with reduced fuel consumption, and therefore, decreasing emissions and eventually was conducted by [15-17] as reported in [18].

Jalili *et al*. [19] carried out the thermodynamic, economic, and environmental aspects trigeneration system driven by biomass and natural gas using the Eco- Indicator 99 method to quantify the environmental

impact. The result of their study revealed that the mixture of natural gas and synagas when their mass flow rates ratio increases from 0 to 0.5 enhances the exergy efficiency whereas the cost of the product per exergy unit and environmental impact per exergy unit of total products are seen to decline. Also, in that study, an increase in splitter separation ratio from 0 to 1 again enhances the exergy efficiency while the cost and environmental impact per exergy of the total products drop. Shamoushaki and Ehyaei [20] carried out exergy, economic and environmental analysis of Alibad Katoul power plant as well as multiobjective optimization using NSGA- II algorithm for a cycle with or without preheater. Two objective functions namely the total cost rate and environmental impact cost were considered. The result showed that the existence of an air preheater reduces both objective functions such that the values for total cost rate and environmental impact cost were higher for a cycle without air preheater than a cycle with an air preheater.

Although exergy analysis can usually be applied to energy conversion systems, it appears to be a more useful method for power cycle analysis than energy analysis because it helps to determine the true magnitudes of losses, their causes and locations, and assesses the overall system and its components. The largest fraction of conventional power plants in the world operates on a reheat-regenerative steam power cycle. Thus, in the last few decades, the exergy analysis of steam power plants has acquired significant popularity. However, the exergo environmental analysis of steam power plants operating in Nigeria climate is yet to be implemented. Thus, this paper aims to apply the exergy analysis in the reduction of environmental pollution from steam power. The beauty of this study is such that it will explore ways to reduce environmental pollution from the steam power plant through exergy analysis.

2. DESCRIPTION OF THE POWER PLANT USED AS A CASE STUDY

The plant analyzed is a 6-unit 220 (6X220) MW dual fired (natural gas and heavy oil) system with modern control equipment, single reheat; six stages regenerative feedwater heating whose technical specification is shown in Table **1**.

The plant uses a traditional Rankine cycle with regenerative feedwater heating. The power plant consists of a steam generator which includes an economizer, boiler, superheater, reheater and other accessories. It also contains the turbine, the condenser,

Table 1:   Egbin Steam Power Plant Design Condition

Initial steam pressure	125000kpa
Initial steam temperature	538^{0}C
Reheat temperature	538^{0}C
Exhaust steam pressure	8.5Kpa
Power rating	220MW
Steam flow rate	647504kg/hr
Fuel flow rate	43- 45t/hr
Airflow rate	572760kg/hr
No of Stages:	High-Pressure Turbine (HPT) Intermediate Pressure Turbine (IPT) Low-pressure turbine (LPT)
No of Heaters:	3-LP Heaters; 1 Deaerator; 2- HP Heaters Extraction
No of Extractions	1- HP; 2 –IP;3- LP
Cooling water	Lagoon
Cooling water inlet temperature	30^{0}C
Cooling water outlet temperature	36^{0}C
Condenser vacuum	8.5kpa
Condensate temperature leaving hot well	42.7^{0}C
Condenser water flow rate	32660m^3/h

generator, pumps, feedwater heaters, drain cooler and deaerator. Additional components are typically applied to improve the performance of the system and improve efficiency. Natural gas is used as the main energy supply and is burned with air under pressure to start the boiler. The gas is supplied to the plant directly from the Nigerian Gas Company (NGC) Lagos operations department which is annexed to the power station. The feedwater enters the boiler drum at a temperature of 236.6^{o}C and boiler pressure of 12. 5Mpa. It is heated from the burners in the boiler (BL) through the wall tubes and heat transfer during the burning process. This generates superheated steam at a temperature of 538°C with the same pressure and enters the first turbine cylinder, the high-pressure turbine (HPT). The steam is reheated in the boiler at the initial temperature of 538°C on leaving the high-pressure turbine and enters the second turbine cylinder, intermediate pressure turbine (IPT) at the same pressure and enters the third turbine cylinder, the low-pressure turbine (LPT) at a lower pressure and temperature. Steam leaves low-pressure turbine exhaust at about 0.92 quality and enters the condenser(C) which leaves as condensate. The heat of condensation is taken away from the lagoon by large circulating cooling water (CW)

located near the power plant back to the lagoon. After passing via a series of heaters, the condensed steam is redirected to the boiler through a condensate effective pump (CEP) and boiler feed pump (BFP). The extracted steam from turbine stages heats the liquid water in the heaters before going back to the boiler through the feedwater heaters. There are five surface feedwater heaters, two high-pressure heaters (HPH 6 and HPH 5), three low-pressure heaters (LPH 3, LPH 2 and LPH 1), a deaerator (DRT) and one drain cooler (DC). Thus, the thermodynamic process of the cycle is repeated continuously. Figure **1** shows the flow process diagram.

3. METHODOLOGY AND MODELLING APPROACH

For the power plant components, thermodynamic models are designed to achieve the elimination of environmental emissions from the steam power plant system using an exergy analysis method. Energy and exergy efficiencies, exergy destruction efficiency, thermal efficiency, plant waste exergy ratio, environmental effect factor and sustainability index factor are among the parameters discussed. Thermodynamic data for the thermodynamic properties of the plant were developed using Engineering Equation Solvers [21] with known thermodynamic intensive properties for each node as shown in Table **2**.

The generated data were fed into Scilab scientific and engineering software code, [22] which was used for computation. Thermodynamic models used in this study were based on mass, energy and exergy balance equations, [23]. The power plant's energy and exergy analysis and the system's component exergy analysis were investigated. By using formulated exergy model equations, each component of the plant was analyzed to determine the one with the highest exergy destruction. With the component identified, improvement would be made to enhance its efficiency. The assumptions made for the development of the power plant models were:

a) Evaluation of performance parameters at the design condition.

b) Steam and liquid water are the working fluid.

c) Each component operates at a steady state.

d) The combustion process is complete.

e) Turbine and pump mechanical efficiencies are 92% each.

f) Generator efficiency is 99%.

g) Kinetic, potential, electrical and nuclear components of exergy are neglected.
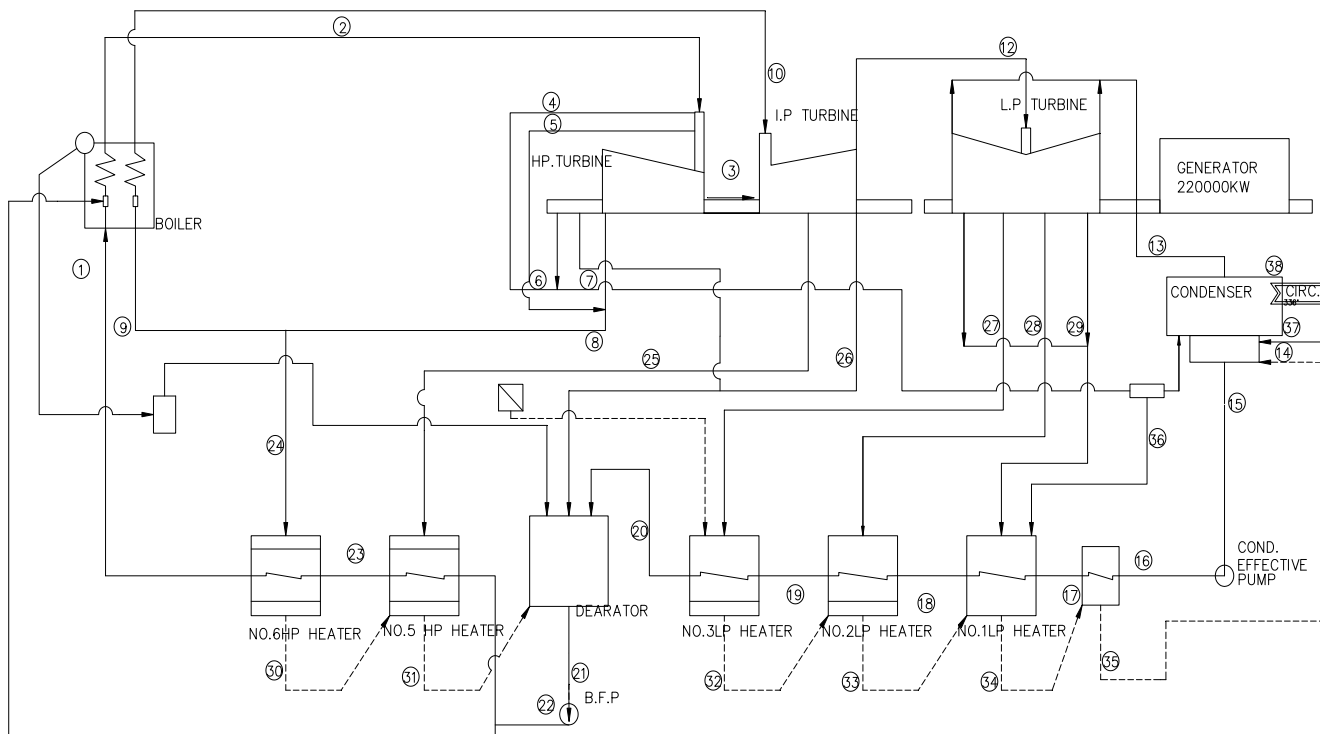

Figure 1: Egbin steam power plant process diagram.

Table 2: Egbin Plant Steam and Water Properties at Design Condition

State	P(kPa)	T(° C)	$\dot{x}$	h (kJ/kg)	s (kJ/kg-k)	$\dot{m}$ (kg/hr)
1	12500	236.6	0.00	1023.80	2.650	627504
2	12500	538	1.00	3443.82	6.600	647504
3	3298	468.9	1.00	3382.90	7.10	11107
4	12500	538	1.00	3443.80	6.60	388
5	12500	538	1.00	3443.80	6.60	1247
6	3343	351.9	1.00	3111.60	6.70	1762
7	3343	351.9	1.00	3111.60	6.70	3675
8	3343	352.2	1.00	3112.30	6.70	630572
9	3289	352.2	1.00	3113.50	6.80	579724
10	3076	538	1.00	3541.40	7.30	579724
11	694.1	332.2	1.00	3126.20	7.40	1081
12	694.1	332.2	1.00	3126.20	7.40	537215
13	8.50	42.7	0.92	2578.20	8.20	459191
14	8.50	42.6	0.00	178.40	0.61	459191
15	8.50	42.6	0.00	178.40	0.61	540986
16	1150	43.1	0.00	181.47	0.613	540986
17	1150	49.58	0.00	208.55	0.70	540986
18	1150	86.70	0.00	363.40	1.15	540986
19	1150	110	0.00	462.10	1.42	540986
20	1150	134.2	0.00	564.90	1.68	540986
21	666.4	163	0.00	688.70	1.973	647504
22	13750	165	0.00	704.9	1.98	647504
22b	0	0	0.00	0.0	0.0	20000
23	13750	196.6	0.00	842.50	2.28	627504
24	3209	350.5	1.00	3111.30	6.71	50308
25	1539	437	1.00	3335.40	7.40	29942
25^1	1447	436.4	1.00	3335.40	7.40	29942
26	694.1	332.2	1.00	3126.20	7.40	26268
26^1	666.4	331.9	1.00	3126.18	7.40	26268
27	365.4	257.1	1.00	2979.40	7.40	22332
27^1	343.4	256.8	1.00	2979.50	7.48	22332
28	174.1	180.1	1.00	2831.40	7.51	20953
28^1	163.7	179.8	1.00	2831.30	7.51	20953
29	76.49	105.7	1.00	2690.3	7.50	34739
29^1	71.88	105.7	1.00	2690.8	7.50	34739
30	3209	202.6	0.00	864.6	2.35	50508
31	1447	171.5	0.00	726.2	2.06	80250
32	343.4	118	0.00	495.4	1.51	22332
33	163.7	94.7	0.00	396.8	1.25	43285
34	71.88	90.6	0.00	371.5	1.20	79265
35	71.88	51.7	0.00	216.5	0.73	79265
36	694.1	347.6	1.00	3158.5	7.47	1241
37	100	30	0.00	125.8	0.40	32660000
38	100	36	0.00	150.9	0.50	32660000

h)   The environment conditions for pressure (P_0) and temperature (T_0) used in this analysis are 1bar and 25^0C.

3.1. Energy Balance Equation

At steady state, energy balance equation ignoring changes in both potential and kinetic energy is given by

$$Q_{cv} - W = \Delta H \tag{3}$$

3.1.1. Energy Analysis of the Boiler

The fuel type used for the computation was natural gas. The energy into the boiler can be calculated from the equation

$$\dot{Q}_{cv}\,\eta_{1,boiler} = \dot{m}_w(h_2 - h_1) - \dot{m}_s(h_{10} - h_9) \tag{4}$$

where $\dot{m}_w$ and $\dot{m}_s$ are mass flow rates of water and steam respectively, $\eta_{1,boiler}$ the efficiency of the boiler sub-system. The first law efficiency of the component is, therefore,

$$\eta_{1,boiler} = \frac{\dot{m}_w(h_2 - (h_1 + h_{22b}) + \dot{m}_s(h_{10} - h_9)}{\dot{Q}_{cv}} \tag{5a}$$

$$\dot{m}_w = \dot{m}_2 = \dot{m}_1 + \dot{m}_{22b} \text{ and } \dot{m}_s = \dot{m}_9 = \dot{m}_{10} \tag{5b}$$

$$\dot{Q}_{cv} = \dot{m}_f * HCV \tag{6}$$

where $\dot{m}_f$ and HCV are fuel mass flow rate and higher calorific value.

3.1.2. Energy Analysis of Turbine Subsystem

The Actual work developed by the turbine is given by

$$W_T = \eta_{mt}(\dot{m}_i h_i - P) \tag{7}$$

$$P = \sum_j \dot{m}_j h_j \tag{8}$$

For the high-pressure turbine work output, W_{HPT}, $i = 2, j = 3,4,6,7,8$. Also, for the intermediate pressure turbine work output, W_{IPT}, $i = in$, and $\dot{m}_{in}h_{in} = \dot{m}_3 h_3 + \dot{m}_{10}h_{10}, j = 11,12,25,26$. For the low-pressure turbine, W_{LPT}, $i = 12, j = 13, 27, 28, 29$.

3.1.3. Energy Analysis of Pump

Actual work input to the pump is given by the equation

$$W_p = \frac{\dot{m}_i(h_i - h_j)}{\eta_{mp}} \tag{9}$$

For the Condenser Effective Pump work, W_{CEP}, $i = 16, j = 15$. Similarly, for the Boiler Feed Pump work, W_{BEP}, $i = 22, j = 21$.

The cycle network is given by

$$W_{net} = \sum W_T - \sum W_P \tag{10}$$

The turbine work is the summation work developed in all turbine stages. Again, pump work is the summation of the work input of the condenser effective pressure and boiler feed pump.

3.1.4. Energy Analysis of the Plant

Cycle thermal efficiency of the plant,

$$\eta_{th} = \frac{(\sum W_T - \sum W_P)}{Q_{cv}} \tag{11}$$

Where $\sum W_T$ summation of work developed by a turbine, $\sum W_p$ summation of pump work and Q_{cv} is the energy input of the fossil fuel which in this case is natural gas.

Overall cycle thermal efficiency of the plant,

$$\eta_{overallth} = \frac{\eta_g(\sum W_T - \sum W_P)}{Q_{cv}} \tag{12}$$

3.2. Exergy Performance Analysis

Conservation of mass, degradation of energy quality and entropy generation was employed by exergy method in the analysis, design and improvement of energy systems. The exergy component of fluid (excluding nuclear effects, magnetism, electricity and surface tension) and ignoring kinetic and potential exergies in a steady flow is given by

$$Ex_{total} = Ex_{th} + Ex_{ch} \tag{13}$$

3.2.1. Power Plant Fuel Exergy

In calculating fuel exergy in the boiler, we first obtain the natural gas composition. The total exergy input to the plant is the thermomechanical exergy of the flowing streams and the chemical exergy of the fuel. The fuel characteristics and composition are shown in Table **3**.

3.2.2. Calculating Thermomechanical Exergy

The thermomechanical exergy of the fuel used in the boiler of the power plant was formulated using the first law energy balance and second law entropy balance. The estimation of the plant's thermomechanical exergy follows the model for an ideal gas described by

$$Ex_{th.} = \dot{m}_f C_p^h(T - T_0) - \dot{m}_f T_0 \left[C_P^{\hat{s}} \ln\left(\frac{T}{T_0}\right) - R \ln\left(\frac{P}{P_0}\right) \right] \tag{14}$$

The mean specific heat capacity for evaluating enthalpy is obtained from

Table 3: Characteristics Composition of the Fuel Used in Egbin Steam Plant

Parameter	Molecular Formula	Percentage by Volume
Methane	CH_4	88.18
Ethane	C_2H_6	5.25
Propane	C_3H_8	1.91
Iso-butane	C_4H_{10}	0.26
Normal butane	C_4H_{10}	0.33
Carbon dioxide	Co_2	3.81
Hexanes	C_6H_{14}	0.02
Nitrogen	N_2	0.15
Normal pentane	C_5H_{12}	0.04
Iso- pentane	C_5H_{12}	0.06
Molecular weight		18.90
Specific gravity		0.635
Gross Calorific Value		1014BTU/SCF OR 37766 KJ/NCM Average = 38000 kJ/NCM

$$C_p^h = \frac{1}{T-T_0}\int_{T_0}^{T} C_p\, dT \tag{15}$$

Also, the mean specific heat capacity for evaluating entropy is evaluated as follows,

$$C_p^s = \frac{1}{\ln\left(\frac{T}{T_0}\right)}\int_{T_0}^{T} C_p\, \frac{dT}{T} \tag{16}$$

In the above equations, the specific heat C_p is expressed as

$$C_p = \frac{\bar{C}_p}{M} = A + BT + CT^2 + DT^3 \tag{17}$$

The gas characteristics constants A, B, C and D are constants are chosen from the ideal gas table [24].

3.2.3. Calculating the Chemical Exergy

Chemical exergy for an ideal gases mix [24] is given as

$$Ex_{ch} = \dot{m}_f \left(\sum_i y_i\, \overline{ex}_i^{ch} + RT_0 \sum_i y_i \ln y_i\right) \tag{18}$$

The standard chemical exergies, molar mass, mole fraction, mass component, mass fraction and percentage by mass compositions of the fuel elements are shown in Table **4**.

3.2.4. Boiler/Steam Generator

The boiler exergy balance is given as

$$\sum_e (\dot{m}_e)_p = \sum_i (\dot{m}_i ex_i)_{f+a} + Ex_{dB} \tag{19}$$

where m_{f+a} is the mass flow rate of fuel and air, m_p the mass flow rate of products after combustion and Ex_{dB} exergy destruction in the boiler.

The exergy efficiency of the boiler is derived from

$$\eta_{2,Boiler} = \frac{\dot{m}_w(ex_2 - (ex_1 + ex_{22b}) + \dot{m}_s(ex_{10} - ex_9)}{(\dot{m}ex)_f} \tag{20}$$

where s and w are subscripts representing steam and water respectively.

When compositions of natural gas used as fuel and air combust in the boiler, combustion products include carbon dioxide, water vapor, nitrogen and oxygen for complete combustion. The combustion reaction which describes the combustion of natural gas used as fuel assuming 65% excess air used in the plant, becomes the equation

Table 4: Standard Chemical Exergy, Molar Mass, Mole Fraction, Mass Component and Mass Fraction of Natural Gas Supplied to Egbin Gas Turbine Plant

Component	Standard Chemical Exergy (kJ/kmol)	Molar Mass M_i	Mole Fraction y_i	Mass of Component (m)	Mass Fraction (m_i)	Percentage by Mass
Methane	831680	16.04	0.8818	14.1441	0.7557	75.5700
Ethane	1,495,955	30.07	0.0525	1.5787	0.0843	8.4300
Propane	2131880	44.097	0.0191	0.8423	0.0450	4.5000
Iso-butane	2800835	58.12	0.0026	0.1511	0.0081	0.8100
N- butane	2805885	58.12	0.0033	0.1918	0.0102	1.0200
Carbon dioxide	19870	44.01	0.0381	1.6768	0.0896	8.9600
Hexanes	4114495	86.18	0.0002	0.0172	0.0009	0.0900
Nitrogen	720	28.01	0.0015	0.0420	0.0022	0.2200
N-pentane	3459770	72.15	0.0004	0.0289	0.0015	0.1500
Iso- pentane	345920	72.15	0.0006	0.0433	0.0023	0.2300
			$\sum = 1.000$	$\sum m = 18.90$	$\sum m_i = 1.000$	$\sum = 100$

Table 5: Molar Mass, Mole Fraction, Mass and Percentage by Mass of Combustion Products

Component	Molecular Formula	Molar Mass (KJ/kg)	Mole Fraction,y_i	Mass of Component (m_i)	Percentage by Mass
Carbon dioxide	CO_2	44.01	1.112	48.9391	9.90
Water	H_2O	18.015	2.0344	36.6497	7.42
Nitrogen	N_2	28.01	12.966	363.1777	75.52
Oxygen	O_2	32.00	1.4141	45.2512	9.16
		494.0177		100.00	

$$[0.8818CH_4 + 0.0525C_2H_6 + 0.0191C_3H_8 + 0.0026C_4H_{10} + 0.0033C_4H_{10} + 0.0002\ C_6H_{14} + 0.0004\ C_5H_{12} + 0.0006\ C_5H_{12} + 0.0381CO_2 + 0.0015N_2 + (1.65) \times (2.09) \times (O_2 + 3.76N_2) = b\ CO_2 + cH_2O + 3.76dN_2 + eO_2 \qquad (21)$$

The molar mass, mole fraction, mass and percentage by mass of the combustion products leaving the plant are shown in Table **5**.

3.2.5. Turbine Sub-System

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

$$W_T\ =\ \dot{m}_j ex_j\ -\ U_i\ -\ Ex_{dT} \qquad (22)$$

The subscripts, i, j are stream flows in or out of the system sub-component.

For the high-pressure turbine, $U_i = \sum_{i=3,4,..8}^{8} \dot{m}_i\ ex_i$, for the intermediate pressure turbine,

$U_i = \sum_{i=11,12,25,26} \dot{m}_i\ ex_i$ and for the low-pressure turbine $U_i = \sum_{i=13,27,28,29} \dot{m}_i\ ex_i$.

Exergy flow into the turbine$= \dot{m}_j ex_j\ -\ U_i$

where $\dot{m}_j ex_j = \dot{m}_2 ex_2$, for the high-pressure turbine, $\dot{m}_j ex_j\ =\ \dot{m}_{in} ex_{in}$ for the intermediate pressure turbine, and for the low-pressure turbine,$\dot{m}_j ex_j\ =\ \dot{m}_{12} ex_{12}$

The subscripts, $2\ 3, ...,$ represents the number of steam extraction from the turbine. There are two components of total irreversibility in the turbine namely, mechanical irreversibility and process irreversibility, [25].

Mechanical irreversibility in the turbine sub- system

$$I_{mt} = (1 - \eta_{mt})P_T \qquad (23)$$

Internal power generated in the turbine sub-system is given by

$$P_T = \dot{m}_j h_j - Q \qquad (24)$$

For the high-pressure turbine, heat flows is, $= \sum_{i=3,4,..8} \dot{m}_i\ h_i$, for the intermediate pressure turbine,

$Q = \sum_{i=11,12,25,26} \dot{m}_i\ h_i$ and heat flow for the low-pressure turbine is $Q = \sum_{i=13,27,28,29} \dot{m}_i\ h_i$

Process irreversibility in the system is given by the equation

$$I_{process} = (\dot{m}_j ex_j - U_i - [W_T + I_{mt}]) \qquad (25)$$

The turbine sub-system exergy destruction is given by the equation

$$Ex_{dT} = I_{mT} + I_{process,T} \qquad (26)$$

The turbine sub-system exergy efficiency is determined by equation

$$\eta_{2,T} = \frac{W_T}{(\dot{m}_j ex_j - U_i)} \qquad (27)$$

3.2.6. Condenser Sub-System

Condenser exergy destruction is given by the equation

$$Ex_{dCond} = (\dot{m}_{13} ex_{13} - \dot{m}_{14} ex_{14}) - \sum_{j=1}\left(1 - \frac{T_0}{T_j}\right)Q_j \qquad (28)$$

The condenser exergy efficiency is determined from the equation

$$\eta_{2,condenser} = 1 - \frac{Ex_{d,cond}}{(\dot{m}_{13} ex_{13} - \dot{m}_{14} ex_{14})} \qquad (29)$$

3.2.7. Pump Sub-System

Work input of pump sub-system exergy balance is given by the equation

$$W_P = \dot{m}_i\left(ex_i - ex_j\right) - Ex_{d,P} \qquad (30)$$

Mechanical irreversibility in the pump is defined by the equation,

$$I_{m,p} = \left\{\frac{1}{\eta_{mp}} - 1\right\}P_p \qquad (31)$$

whereP_p is the internal power requirement of the pump.

The pump sub-system's internal power requirement is given by the equation

$$P_p = \dot{m}_i(h_j - h_i) \tag{32}$$

Exergy flow through the pump is determined from the equation

$$\dot{m}_i(ex_i - ex_j) = \left[\dot{m}_i\left((h_j - h_i) - T_0(s_j - s_i)\right)\right] \tag{33}$$

Process irreversibility through the pump is given as

$$I_{process,P} = W - \left(\dot{m}_i(ex_i - ex_j) + I_{m,P}\right) \tag{34}$$

The exergy destruction in the pump is defined by the equation

$$Ex_{dP} = I_{m,P} + I_{process,P} \tag{35}$$

The pump sub-system's second law or exergy efficiency is determined from equation (36)

$$\eta_{2,P} = \frac{\dot{m}_i(ex_j - ex_i)}{W_P} \tag{36}$$

(i)   **For the condenser effective pump(CEP)**

$$\dot{m}_i = \dot{m}_{15}, ex_i = ex_{15}, ex_j = ex_{16}, h_j = h_{16}, h_i$$
$$= h_{15}, s_i = s_{15}, s_j = s_{16}$$

(ii)   **For the boiler feed pump(BFP)**

$$\dot{m}_i = \dot{m}_{21}, ex_i = ex_{21}, ex_j = ex_{22}, h_j = h_{22}, h_i$$
$$= h_{21}, s_i = s_{21}, s_j = s_{22}$$

3.2.8. Feedwater Heater Sub-System

Modeling the surface feedwater heaters as a heat exchanger; the objective is to maximize the exergy gained by the cold fluid at the expense of the exergy lost by the extracted steam or hot fluid (effectiveness of the heat transfer units). The equation accounts for exergy flow equation for the feedwater heaters

$$\dot{m}_j(ex_k - ex_p) - \dot{m}_i(ex_i - ex_j) = Ex_{d,h} \tag{37}$$

The subscripts, i, j, k and p are streams in and out of the heaters, $\dot{m}_j\left(ex_k - ex_p\right)$ is the exergy input,$\dot{m}_i(ex_i - ex_j)$ is the exergy output and $Ex_{d,h}$ is the exergy destruction of the heaters.

Rate of exergy input to the heater,

$$: Ex_{k-p} = \dot{m}_j\left(ex_k - ex_p\right) = \dot{m}_j[(h_k - h_p) - T_0(s_k - s_p)]$$

$$\tag{38}$$

Rate of exergy output of the heater,

$$: Ex = \dot{m}_i(ex_i - ex_j) = \dot{m}_i[(h_i - h_j) - T_0(s_i - s_j)] \tag{39}$$

The effectiveness or exergy efficiency of the heater is determined from the equation

$$\mathcal{E}_{heater} = \frac{\dot{m}_i(ex_i - ex_j)}{\dot{m}_j(ex_k - e_{p})} \tag{40}$$

(i)   **For the high-pressure feedwater heater 6 (HPH 6),**

$$\dot{m}_i = \dot{m}_{23}; \dot{m}_j = \dot{m}_{24}; ex_i = ex_1; ex_j = ex_{23}; ex_k$$
$$= ex_{24}; ex_p = ex_{30}$$

(ii)   **For the high-pressure feedwater heater 5 (HPH 5),**

$$\dot{m}_i = \dot{m}_{23}; \dot{m}_j = \dot{m}_{31}; ex_i = ex_{23}; ex_j = ex_{22a}; ex_k$$
$$= ex_{e1}; ex_p = ex_{31}$$

(iii)   **For the low-pressure feedwater heater 3 (LPH 3),**

$$\dot{m}_i = \dot{m}_{20}; \dot{m}_j = \dot{m}_{27}; ex_i = ex_{20}; ex_j = ex_{19}; ex_k$$
$$= ex_{27}'; ex_p = ex_{32}$$

(iv)   **For the low-pressure feedwater heater 2 (LPH 2),**

$$\dot{m}_i = \dot{m}_{19}; \dot{m}_j = \dot{m}_{33}; ex_i = ex_{19}; ex_j = ex_{18}; ex_k$$
$$= ex_{e2}; ex_p = ex_{33}$$

For the low-pressure feedwater heater 1 (LPH 1),

$$\dot{m}_i = \dot{m}_{18}; \dot{m}_j = \dot{m}_{34}; ex_i = ex_{18}; ex_j = ex_{17}; ex_k$$
$$= ex_{e3}; ex_p = ex_{34}$$

3.2.9. Drain Cooler

$$\dot{m}_i = \dot{m}_{34}; ex_i = ex_{34}; ex_j = ex_{35}; \dot{m}_j = \dot{m}_{16}; ex_k = ex_{17}; ex_p = ex_{16}$$

3.2.10. Cooling Water

For the cooling water

$$\dot{m}_i = \dot{m}_{38}; \dot{m}_j = \dot{m}_{37}; ex_i = ex_{38}; ex_j = 0; ex_k$$
$$= ex_{37}; ex_p = 0$$

3.2.11. Deaerator

The deaerator is a contact heat exchanger and the exergy balance is given as

$$[(\dot{m}_7 ex_7 + \dot{m}_{26} ex_{26} + \dot{m}_{20} ex_{20} + \dot{m}_{31} ex_{31}) - (\dot{m}_{21} ex_{21})] = Ex_{d,D} \tag{41}$$

Where$\dot{m}_7 ex_7 + \dot{m}_{26} ex_{26} + \dot{m}_{20} ex_{20} + \dot{m}_{31} ex_{31}$the exergy is input, $\dot{m}_{21} ex_{21}$ is the exergy output and $Ex_{d,D}$is the exergy destruction in the deaerator.

The effectiveness of the deaerator is

$$\varepsilon_{dearator} = \frac{\dot{m}_{21}ex_{21}}{(\dot{m}_{20}ex_{20}+\dot{m}_{26}ex_{26}+\dot{m}_{31}ex_{31})} \tag{42}$$

3.2.12. Exergy Efficiency of the Plant

Plant cycle exergy efficiency is given by the equation

$$\eta_{2cy} = \frac{(\sum W_T-\sum W_P)}{Ex_f} \tag{43}$$

Using equation (44), the overall cycle exergy efficiency of the plant is determined.

$$\eta_{2overallcycle} = \frac{\eta_g(\sum W_T-\sum W_P)}{Ex_f} \tag{44}$$

Waste exergy ratio: This represents the fraction of total exergy destruction to the total exergy into the system.

$$\tau = \frac{Ex_{d,Total}}{Exin_{Total}} \tag{45}$$

Environmental effect factor: The is the ratio of waste exergy ratio to the plant's exergy efficiency.

$$\gamma_{eef} = \frac{\tau}{\eta_2} \tag{46}$$

Exergetic sustainability index: This parameter is critical among exegetic sustainability indicators to measure the system's sustainability level, [14]. It ranges between 0 and ∞. The plant high exergy efficiency means low waste exergy ratio and low environmental effect factor and high exergetic sustainability index. It is determined according to equation (47)

$$\lambda = \frac{1}{\gamma_{eef}} \tag{47}$$

3.2.13. Validation of the Study

It is important to ensure that the numerical model employed in the analysis of a system is always validated. This eliminates all ambiguities pertaining to the accuracy of the result. The validation approach employed is the use of the present approach to reproduce the work of previous authors. The exergetic analysis approach and sustainability indicators studied were similar to the works of previous studies with similar results [13, 14]. The result of exergy destruction ratio is greater than zero as seen in the result of work reported in [14]. Our confidence in the exergy method adopted is hinged on the assumption that it produced a similar result to those reported in verified published journal articles, it can be trusted in accurately predicting novel work on exergy and environmental assessment of a steam power plant. The method adopted produced results similar to the work done by [13,14]. Hence, the model utilized in this study is deemed accurate and reliably and can be trusted invalidating the result obtained.

4. RESULT AND DISCUSSION

The (6x220MW) Egbin steam power plant at design condition was analyzed using exergy analysis and the ambient reference temperature and pressure used were 25°C and 1 bar, respectively. The thermodynamic properties of water and steam at different states shown in the process diagram Fig. **1** were determined using Engineering Equation Solver software [14]. The plant model equations developed for the components of the power plant were analyzed using SCILAB software code, [22] and the result summarized in Table 6 and Figure **2**, respectively. In Table 6, the plant's component efficiencies and sustainability indicators were presented. The boiler has exergy and exergy destruction efficiencies of 43.78 % and 56.2%, respectively. The turbine stages, namely, high-pressure turbine, intermediate pressure turbine, the low-pressure turbine has exergy and exergy destruction efficiencies of 83.84% and 16.16%; 82.60% and 17.40%; 86.83% and 13.07%, respectively. The power plant's power output is 202.92MW while its heat input and thermal efficiencies were 597.07MW and 33.98%., respectively. The exergetic efficiency of the cycle was found to be 33.07%.

The study also considered certain sustainability parameters such as waste exergy ratio, environmental effect factor and sustainability index. The values of these parameters from the result shown in Table 6 indicates that waste exergy ratio, environmental effect factor and sustainability index factor were 0.425; 1.2853 and 0.7780. The boiler's exergy destruction efficiency was dominant over all other major components in the plant displayed in the table. It accounts for 56.22%. That suggests that there are significant opportunities for improvement. However, owing to human, technical and economic limitations, much of the irreversibility can not be prevented. The boiler exergy loss was due to combustion reaction and large temperature difference during heat transfer between the combustion gases and the working fluid. Tubes fouling, faulty burners, fuel efficiency, unreliable soot blowers, valves steam traps and air heaters fouling are other causes that may add to the high amount of irreversibility.

4.1. Improvement Potential

Improved thermodynamic efficiency can entail the reduction of irreversibilities; measures in this direction

Table 6:   Components Efficiencies and Sustainability Indicators of Egbin Power Plant

Boiler		**High Pressure Heater 6 (HPH6)**	
Exergy efficiencyη_2(%)	43.78	Exergy efficiencyη_2(%)	93.53
Exergy destruction ratioδ_{boiler}(%)	56.22	Exergy destruction ratioδ_{HPH6}(%)	6.47
Exergy destruction (KW)	344818.96	Exergy destruction (KW)	856.923
Exergy in boiler (KW)	613318.2	Exergy in to the HPH6 (KW)	13239.668
High Pressure Turbine (HPT)		**High Pressure Heater 5 (HPH5)**	
Work output (KW)	54087.492	Exergy efficiencyη_2(%)	91.11
Exergy efficiencyη_2(%)	83.84	Exergy destruction ratio, δ_{HPH5}(%)	8.89
Exergy destruction ratio, δ_{HPT}(%)	16.16	Exergy destruction (KW)	819.144
Exergy destruction (KW)	10427.708	Exergy in to the HPH5 (KW)	9220.725
Exergy in HPT (KW)	64515.19		
Intermediate Pressure Turbine (IPT)		**Low Pressure Heater 3 (LPH3)**	
Work output (KW)	57704.369	Exergy efficiencyη_2(%)	87.00
Exergy efficiencyη_2(%)	82.60	Exergy destruction ratioδ_{LP} (%)	13.00
Exergy destruction ratio, δ_{IPT}(%)	17.40	Exergy destruction (KW)	568.663
Exergy destruction (KW)	12159.681	Exergy in to the LPH3 (KW)	4373.598
Exergy in IPT (KW)	69864.049		
Low Pressure Turbine (LPT)		**Low Pressure Heater 2 (LPH2)**	
Work output (KW)	94793.77	Exergy efficiencyη_2(%)	80.93
Exergy efficiencyη_2(%)	86.93	Exergy destruction ratio, δ_{LPH2}(%)	19.07
Exergy destruction ratio, δ_{LPT}(%)	13.07	Exergy destruction (KW)	646.056
Exergy destruction (KW)	14252.356	Exergy in to the LPH2 (KW)	3387.051
Exergy in to the LPT (KW)	109046.13		
Condenser		**Low Pressure Heater 1 (LPH1)**	
Heat rejection Q_1(KW)	22771.278	Exergy efficiencyη_2(%)	73.28
Exergy efficiencyη_2(%)	49.96	Exergy destruction ratio, δ_{LPH1}(%)	26.72
Exergy destruction ratio, δ_{cond}(%)	50.04	Exergy destruction (KW)	1133.971
Exergy destruction (KW)	8119.489	Exergy in to the LPH1 (KW)	4244.641
Exergy into the Condenser (KW)	16225.769		
Condenser Effective Pump (CEP)		**Deaerator**	
Work input (KW)	501.457	Exergy efficiencyη_2(%)	95.97
Exergy efficiencyη_2(%)	65.21	Exergy destruction ratio, δ_D(%)	4.03
Exergy destruction ratio, δ_{CEP}(%)	34.79	Exergy destruction (KW)	760.231
Exergy destruction (KW)	174.461	Exergy in to the deaerator (KW)	1880.63
Exergy in to the CEP(KW)	501.457		
Cycle		**Drain cooler**	
Heat input (KW)	597068.27	Exergy efficiencyη_2(%)	82.11
Network W$_{net}$(KW)	202917.03	Exergy destruction ratio, δ_{DC}(%)	17.89
Cycle thermal efficiency η_1(%)	33.98	Exergy destruction (KW)	0.00236
Cycle exergy efficiencyη_2(%)	33.07	Exergy into the drain cooler (KW)	0.01319
Power output (KW)	200887.93	**Cooling water**	
Overall power plantexergy efficiencyη_2(%)	38.79	Exergy efficiencyη_2(%)	0.00
Waste exergy ratio WER (τ)	0.4251		
Environmental effect factor EEF (γ_{eef})	1.2852		
Sustainability index factor SIF (λ)	0.7780		

are typically limited by a range of cost-related practical factors. An improvement strategy has been made on the process diagram of the boiler sub-system of Figure **3** which has the highest exergy destruction efficiency.

Low exergy efficiency in the boiler which gave rise to the highest exergy destruction is attributed to reasons such as incomplete combustion and high-temperature difference in the heat transfer process. The efficiency
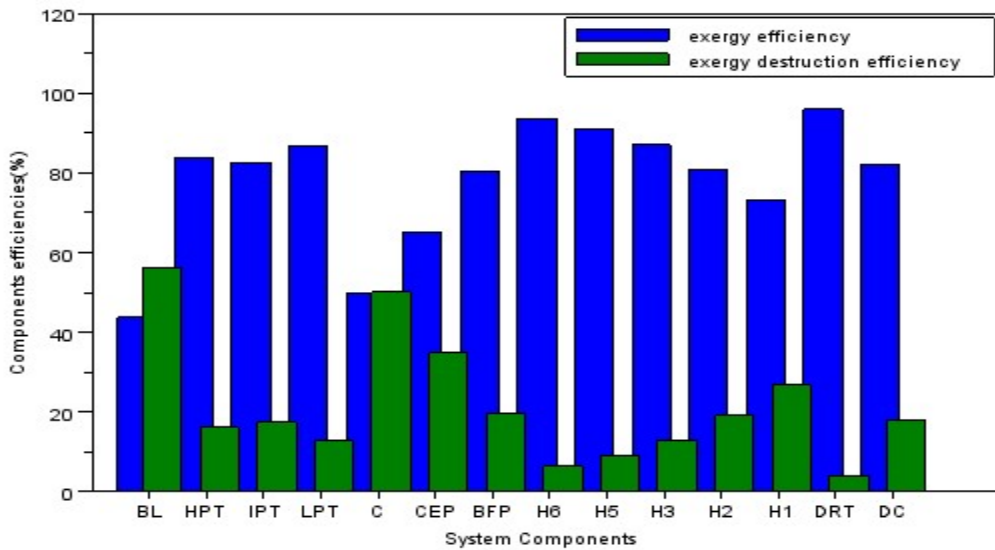
Figure 2: Exergy efficiencies and exergy destruction ratios of components at the design condition.
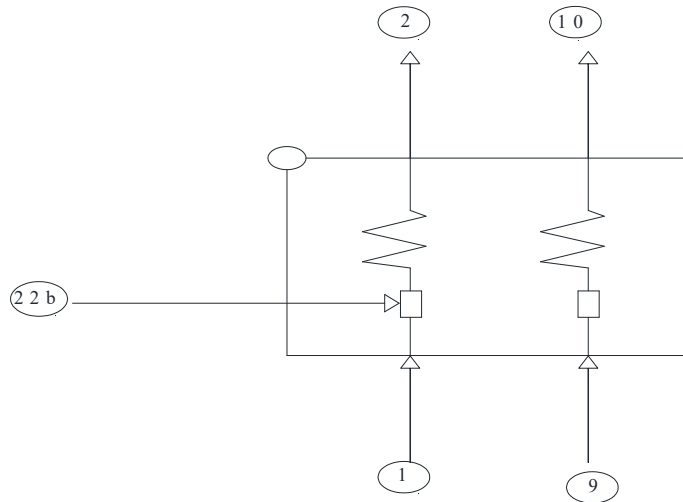

Figure 3: Boiler sub-system.

enhancement approach considered on the component is an increase in inlet temperature of the high-pressure turbine at constant initial boiler pressure and simultaneous increase in inlet temperature of high-pressure turbine and boiler pressure. For the first approach, we adjusted the inlet temperature of the high-pressure turbine (outlet temperature of boiler) between 538°C to 558°C in a step of four by 5°C at constant boiler pressure of 125 bar. Table **7** shows effect of optimizing efficiency on the system by increment in inlet temperature of high-pressure turbine at constant initial boiler pressure and other parameters such cycle exergy efficiency, waste exergy ratio, environmental effect factor, sustainability index factor.

This approach showed boiler exergy efficiency increment from 43.78% to 44.85% and the exergy destruction efficiency decrease from 56.22% to 55.15%. The cycle exegetic efficiency of the steam plant considered increased from 33.08% to 34.48% indicating a gain of 4.23 %. Parameters such as waste exergy ratio decreased to 0.4130 from 0.4251 suggesting a drop of 2.85% wastes entering the lagoon. Also, the environmental effect factor was decreased from 1.2852 to 1.1979 showing a 6.79% reduction in enviromental pollution. Reducing waste exergy ratio and environmental effect factor would boost efficiency and increase sustainability in power plant. The sustainability index factor was increased from 0.7780 to 0.8347 indicating 7.29% effective energy resource utilization by the plant.

Figure **4** revealed the plot of boiler exergy efficiency against the increase in inlet temperature of a high-pressure turbine at constant boiler pressure. Figure **5**

Table 7:   Increase in inlet temperature of high pressure turbine at constant boiler pressure.

Component	T= 538°C P = 125 bar	T= 543°C P = 125 bar	T= 548°C P = 125 bar	T= 553°C P = 125 bar	T= 558°C P = 125 bar
Boiler					
Energy eff, $\eta_{1boiler}(\%)$	84.44	84.84	85.23	85.62	86.01
Exergy eff,, $\eta_{2boiler}(\%)$	43.78	44.07	44.27	44.56	44.85
Exergy des.eff $\delta_{boiler}(\%)$	56.22	55.93	55.73	55.44	55.15
Steam power plant					
Cycle thermal eff, $\eta_1(\%)$	33.98	34.35	34.71	35.07	35.42
Cycle exergy eff $\eta_2(\%)$	33.08	33.43	33.78	34.13	34.48
Overall thermal eff $\eta_{1overall}(\%)$	33.64	34.00	34.36	34.72	35.07
Overall exergy eff $\eta_{2overall}(\%)$	32.75	33.10	33.44	33.78	34.13
Waste exergy ratio WER(τ)	0.4251	0.4220	0.4192	0.4160	0.4130
Enviromentaleffect factor EEF(γ_{eef})	1.2852	1.2623	1.2410	1.2192	1.1979
Sustainability index factor SIF(λ)	0.7780	0.7922	0. 8059	0.8202	0.8347

showed the plots of cycle exergy efficiency against environmental effect factor and sustainability index. Figure **5**(**a**) showed that increasing the cycle exergy efficiency of the plant decreases the impact of emissions on the environment. This means that steam power plant is more efficient when there are fewer emissions to the environment. Figure **5**(**b**) showed that increasing cycle exergy efficiency improves the steam power plant's sustainability which indicates efficient fuel utilization in the boiler.

The second improvement performance approach considered is a simultaneous increase in inlet temperature of high-pressure turbine and boiler pressure. The inlet temperature of the high-pressure turbine was raised from 538°C to 558°C in a step of four

by 5°C and the boiler pressure was increased from 125 bar to 165 bar in a step of four by 10 bar. The result is presented in Table **8**.

The exergy efficiency of the boiler increased from 43. 78 percent to 45.13 percent. The percentage exergy destruction decreased from 56.22 percent to 54.87 showing 2.40 % reduction. The cycle exergy efficiency increased from 33.08 percent to 39.36 percent indicating 18.98% improvement. The waste exergy ratio and environmental effect factor decreased from 0.4251 to 0.4162 and 1.2852 to 1.0573 showing 2.09% and 17.32% decrease, respectively. The sustainability index factor increased rapidly from 0.7780 to 0.9456 showing 21.54% efficient energy resource utilization. Figure **6** showed the plots of cycle exergy efficiency against
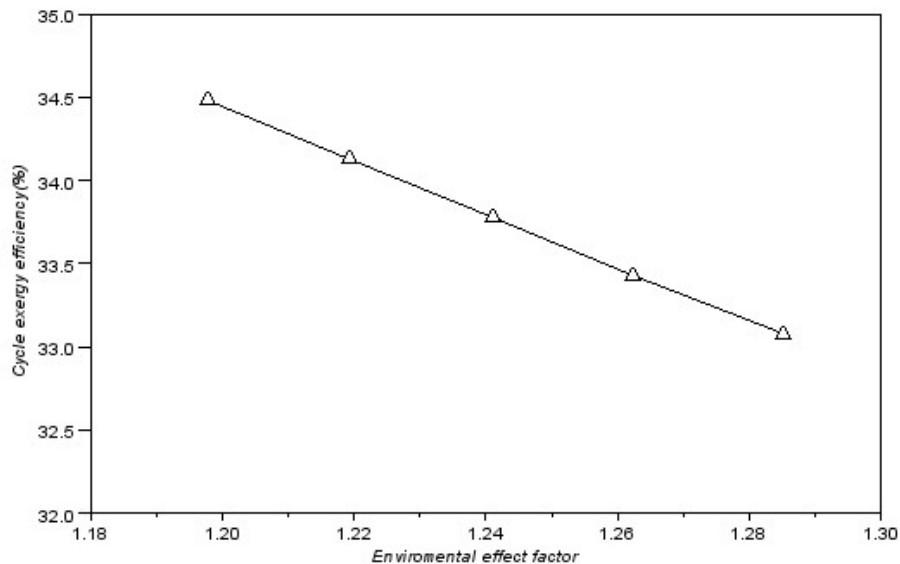

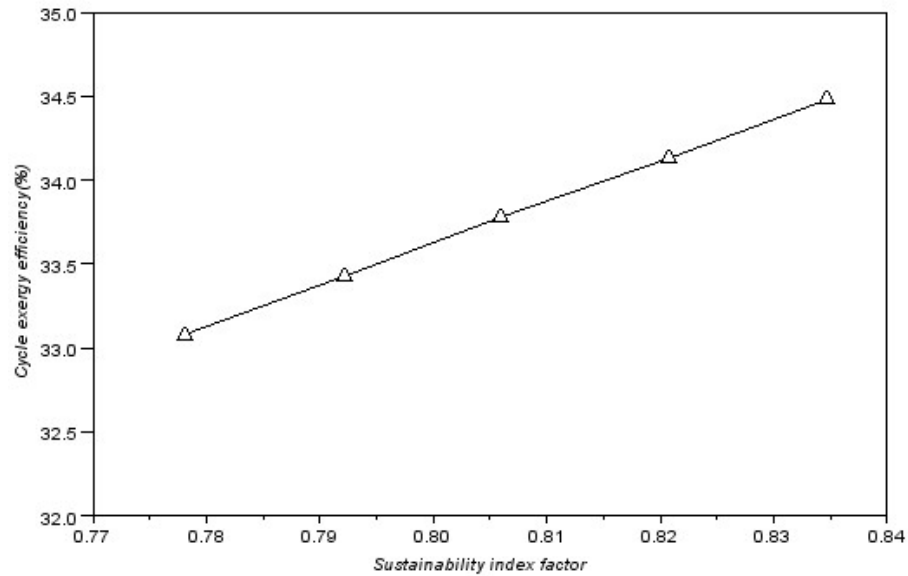**Figure 5:** (a) cycle exergy efficiency versus environmental effect factor.

Figure 5: (b) cycle exergy efficiency versus sustainability index factor.

Table 8:   Simultaneous Increase in Inlet Temperature of High-Pressure Turbine and Boiler Pressure

Component	T= 538°C P = 125 bar	T= 543°C P = 135 bar	T= 548°C P = 145 bar	T= 553°C P = 155 bar	T= 558°C P = 165 bar
Boiler					
Energy eff, $\eta_{1boiler}$(%)	84.44	84.51	84.60	84.69	84.78
Exergy eff,, η_{2boil} (%)	43.78	44.18	44.52	44.78	45.13
Exergy des.eff δ_{boiler}(%)	56.22	55.82	55.48	55.22	54.87
Steam power plant					
Cycle thermal eff, η_1(%)	33.98	40.28	40.34	40.39	40.45
Cycle exergy eff η_2(%)	33.08	39.21	39.26	39.31	39.36
Overall thermal eff $\eta_{1overal}$ (%)	33.64	39.88	39.93	39.99	40.04
Overall exergy eff $\eta_{2overall}$(%)	32.75	38.81	38.86	38.91	38.96
Waste exergy ratio WER(τ)	0.4251	0.4237	0.4209	0.4187	0.4162
Environmental effect factor EEF(γ_{eef})	1.2852	1.0807	1.0722	1.0651	1.0573
Sustainability index factor SIF(λ)	0.7780	0.9253	0. 9327	0.9388	0.9456

environmental effect factor and sustainability index factor with a simultaneous increase in inlet temperature of high-pressure turbine and boiler pressure. Figure **6** (**a**) showed that increasing the cycle exergy efficiency of the plant further decreases the impact of emissions on the environment. Figure **6** (**b**) showed that with both increases in inlet temperature of high-pressure turbine and boiler pressure, the cycle exergy efficiency increases with sustainability index factor which, of course, means better fuel utilization on this improvement approach.

5. CONCLUSION

In this research, the application of exergy analysis to determine the level of wastes and environmental

pollution reduction on Egbin steam power plant at design condition was performed. The primary purpose of this analysis was to evaluate the components of the system individually and determine the site with the greatest efficiency in exergy destruction. The exergy analysis identified and quantified exergy destruction in each component of the plant. Due to various internal and external irreversibilities associated with the component, the boiler was identified as the system that has the highest value of exergy destruction. The exergy destruction efficiency in the boiler was more compared to exergy destruction efficiencies in the turbine, condenser, feedwater heaters and other components. It is obvious from the study that enhancement of power plant performance can be achieved through increasing
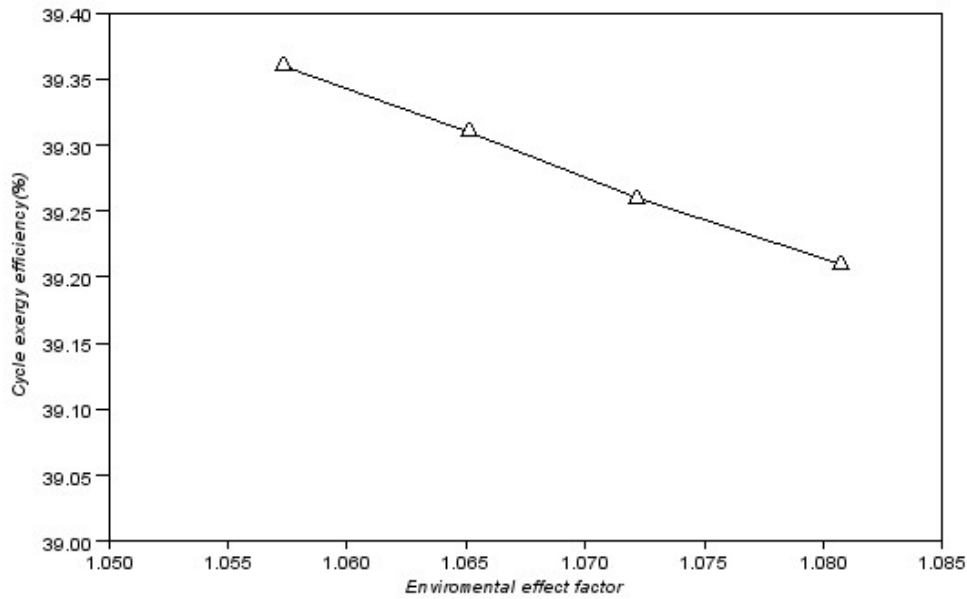
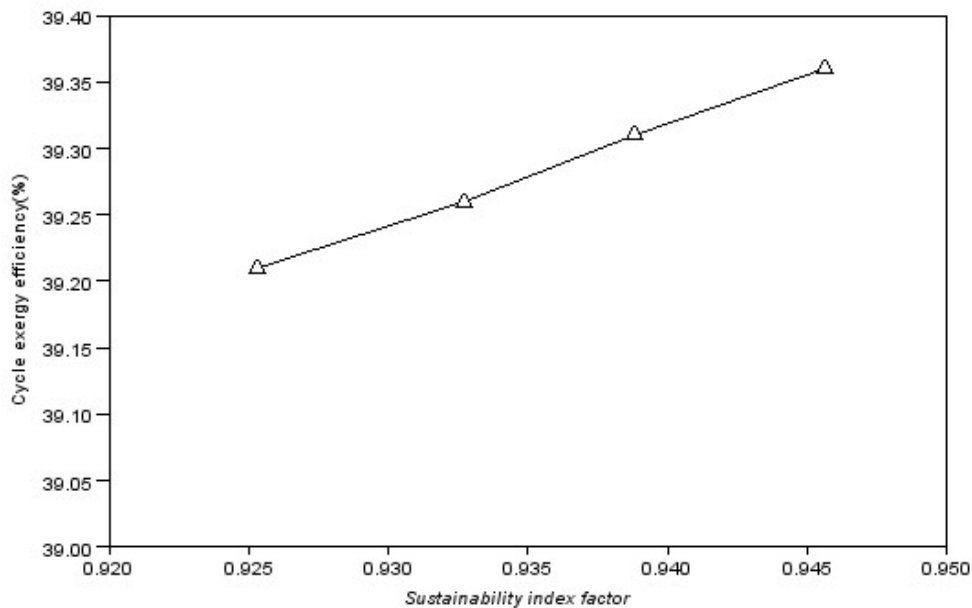**Figure 6:** (a) cycle exergy efficiency versus environmental effect factor

Figure 6: (b) cycle exergy efficiency versus sustainability index factor.

the inlet temperature of a high-pressure turbine with constant boiler pressure and simultaneous increase in inlet temperature of high-pressure turbine and boiler pressure. The two improvement performance criterion reduced the pollution effect on the environment and increases efficient energy resource utilization thus increasing sustainability. The first improvement performance criterion, the effect of increasing the inlet temperature high-pressure turbine at constant boiler pressure reduced the environment effect factor by 6.79% and increased the sustainability index factor by

7.29%. Furthermore, the second improvement approach, simultaneous increase in inlet temperature of high-pressure turbine and boiler pressure reduced further the environmental effect factor and enhanced the sustainability index factor by 17.32% and 21.54%, respectively. The improvement approach which gave the highest reduction on environmental impact on the environment is the second improvement approach. The novelty of this study revealed that gaseous emission from the steam power plant can be reduced by exergy analysis with a simultaneous increase in inlet tem-

perature of high-pressure turbine and boiler pressure. providing a better improvement option between the two options considered.

The findings revealed that reduction in thermal emissions or environmental effect factor improves both cycle exergy efficiency and sustainability index factor meaning that the system uses well its energy well for more power production. Exergy method is important in examining which enhancement options are recommended. It should be used in conjunction with other technical information to guide efforts to improve the performance of steam power plants to reduce pollution.

CONFLICT OF INTEREST STATEMENT

On behalf of all authors, the corresponding author states that there is no conflict of interest.

DATA AVAILABILITY STATEMENT

The data presented in this work can be obtained from the corresponding author upon reasonable request.

NOMENCLATURE

c_p	=	Specific heat at constant pressure (KJ/KgK)
$\bar{c}_p$	=	Molar specific heat at constant pressure (kmol/KgK)
i	=	Inlet
e	=	Exit
$E_{x\,(tot)}$	=	Total exergy (KW)
Ex_{th}	=	Thermomechanical exergy (KW)
Ex_{ch}	=	Chemical exergy (KW)
$\overline{ex}_i^{ch}$	=	Standard chemical exergy of the constituents
y_i	=	Mole fraction of the constituents
$\dot{m}_f$	=	Fuel mass flowrate (Kg/s)
$\dot{m}_w$	=	Mass flow rate of water (Kg/s)
$\dot{E}x_d$	=	Exergy destruction rate (KW)
HCV	=	Higher Calorific Value
h	=	Specific enthalpy in KJ/kg
$\bar{h}$	=	Molar enthalpy in Kmol/k
h_0	=	Enthalpy at the reference or environmental temperature
$\dot{Q}_{cv}$	=	Heat transfer rate in KJ/s

ΔH	=	Enthalpy Change
R	=	Specific gas constant
$\bar{R}$	=	Molar gas constant
S	=	Specific entropy (kJ/kgK)
$\bar{S}$	=	Molar entropy (Kmol/K)
s_0	=	Entropy at reference or environmental temperature
T_0	=	Ambient temperature, K
$\dot{W}$	=	work rate in KJ/s

Greek Words

η_1	=	First law efficiency or energy efficiency
η_2	=	Second law efficiency or exergy efficiency
δ_i	=	Exergy destruction efficiency
$\bar{g}$	=	Gibbs function on molar basis
τ	=	Waste exergy ratio
γ_{eef}	=	Environmental effect factor
λ	=	Exergy sustainability index
$\dot{\sigma}$	=	Entropy production rate (KJ/kg)

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