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Performance evaluation of a major thermal power plant in Nigeria

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Abstract. Electric power is the bedrock of sustainable development in modern society. The demand for its adequate and reliable supply at a very competitive price is continuously increasing with population increase and the industrial revolution. A significant limitation to meeting this demand, is the inefficient operation of several of the existing power plants, resulting in their inability to generate electricity equivalent to their installed capacity. In this study, the exergy based performance evaluation of a major power plant in Nigeria was conducted to identify opportunities for thermodynamic improvement. Historical data of the plant was fed into HYSYS 8.8 to simulate its operations and obtain necessary thermodynamic data for assessing its performance. With the aid of codes embedded in HYSYS and the use of Ms Excel, the synthesized plant's thermodynamics data was used for its performance evaluation. Components-wise evaluation revealed that apart from the turbines, exergy efficiencies were lower than energy efficiencies. The overall energy efficiency of the plant was found to be 33.19% while the corresponding exergy efficiency was 31.94%. The boiler was identified as the unit with the highest irreversibility and most significant contributor to overall plant's inefficiency. It is posited that adequate knowledge of the effect of changes in operating parameters and load variation on performances will be handy in addressing inefficiencies in the boiler and other components of the plant.

1. Introduction

The importance of electricity to modern society cannot be overestimated. Among other things, electricity is used in modern societies, for operating various domestic appliances, cooking, lighting, operating educational aids, obtaining comfortable living temperature, piped water, essential health care, security, communication, and transportation. In the rural areas, it is vital to various agricultural practices such as land preparation, fertilization, irrigation, livestock rearing, processing and preservation of farm produce [1]. The generation of electricity, which supports modern-day living and economic activities, requires the consumption of fuel. The fuels which serve as the primary source of energy in electricity generation may exist in gaseous, liquid or solid forms; they may be renewable or non – renewable.

Electricity generation from any kind of primary energy source is associated with some levels of operational challenges, risks, and environmental effects. In Nigeria, fossil fuels have been the primary source of energy for power plants and several other sectors. Various projections indicate that the need for fossil fuels will continue and renewable sources of energy will not be sufficient in the short-to-medium term to replace them. The efficient operations of power plants firing fossil-based fuels is, therefore, germane. To a large extent, the efficiency of the plant determines emission levels, electricity availability to final consumers and cost of electricity to the consumers.

Nigeria, with a total installed capacity of about $12,522$ MW could barely boast of average available power of $4,996$ MW[2], [3]. The low power availability is partly due to the inefficient operation of several of the existing plants, which makes them perform far below their installed capacity [3]. Electricity blackouts, brownouts, excessive reliance on self-generated electricity from captive diesel and gasoline generators and economic under productivity are some of the adverse effects of the inefficient operation of power plants in Nigeria with a population of over 197 million[4]–[8]. Thermodynamic performances are investigated to improve the efficiencies of energy conversion systems. Energy analysis (which is based on the first law of thermodynamics) is typically used to examine energy conversion technologies. However, a better understanding is attained when a more thermodynamic view which uses the second law of thermodynamics in conjunction with energy analysis, via exergy methods is taken. This is because, by the first law of thermodynamics, energy cannot be lost but is conserved. However, exergy (useful energy) can be degraded due to irreversibilities. As such, exergetic efficiency gives the true efficiency of the system and points out areas with potentials for thermodynamic improvement [9]–[12]. Exergy losses, particularly when making use of non - renewable forms of energy should be minimized in order to foster sustainable development by lengthening the lives of existing resource reserves, generating more electricity from fixed amount of fuel and reducing the cost at which electricity gets to the final consumer.

The Comparative performance evaluation of some Kraftwerk Union (KWU) designed coal-fired power plants in India, using the first law and second law of thermodynamics have been conducted by [13]. The comparative assessment done at various loads show that in the same turbo-generator set, there is the possibility of energy and exergy efficiency improvement at specific loads. It was also found that proper maintenance of the major components and operating parameters of the understudied turbo-generator sets give the possibility of near design performance at various load ranges even after two decades of operation. A study of the effect of heat recovery steam generator (HRSG) configuration on performances of combined cycle power plants, using energy, exergy and economic indices have also been carried out. The outcome of the study shows that heat recovery from flue gas is enhanced at increased pressure levels of steam generation, leading to an increase in the energy efficiency of the cycle. In addition to this, as the number of pressure levels of steam generation in HRSG increases, the rate of exergy destruction in the cycle decreases[14]. Losses in boiler and turbine using exergetic criteria was measured by [15]. The exergy loss profile obtained reveals that highest exergy losses in the 32 MW coal-fired power understudied is attributable to the boiler and turbine. Although several studies have been conducted on the performance of various power plants using both the first and second law of thermodynamics. There appears to be a shortage of exergy based performance analysis on power plants operating in Nigeria. In this study, the performance assessment of a major power plant in Nigeria was carried out based on the first and second laws of thermodynamics. Areas of losses based on efficiency and irreversibility were pointed out for improvement purposes.

2. Methodology

This section highlights the materials used, gives a succinct description of the plant understudied and the approach adopted for the performance analysis.

2.1. Materials
The materials used for this research include the design and operating data of the thermal power plant, HYSYS V 8.8 simulation software and Microsoft EXCEL used for data evaluation.

2.2. Process Description
The power plant understudied operates a modified Rankine cycle. As such, the main components of the plant are boiler, turbines, pumps, condenser, and generator. The boiler, which is dual fired, is enabled to use either natural gas or high/low pour fuel oil (HPFO/LPFO). The turbine section of the plant comprises of the high-pressure turbine (HPT), intermediate pressure turbine (IPT) and the low-pressure turbine (LPT). The boiler produces superheated and reheated steams from the feed water (supplied by the boiler feed pump) and the HPT exhaust respectively. The superheated steam from the boiler at a pressure of about $12500\ kPa$ and a temperature of roughly $540°C$ turns the high-pressure turbine at a speed of $3000\ rpm$. Reheated steam is used to turn the intermediate turbine while the steam from the intermediate turbine turns the low-pressure turbine. The generator which is directly coupled to the rotor of the turbine also turns at 3000 rpm and generates a 3- phase AC power of 220 MW at full capacity through the cumulative mechanical energy from the turbines. The exhaust steam from the low-pressure turbine at a pressure of $8.5\ kPa$ is condensed to water in the condenser. Pretreated circulating water from a lagoon serves as the cooling water in the condenser. The condensed water from the condenser goes to the condensate polishing plant for treatment before being sent back to the boiler drum as feed water. The cooling water with raised temperature leaves the condenser through a discharge channel into the lagoon. The temperature of the effluent water is reduced in an auxiliary cooling system before eventually being released into the lagoon. Demineralized water is introduced as makeup water at the hot well to augment for inherent losses in the cycle. Table 1 gives the composition of natural gas used as fuel for this study, and a pictorial representation of the process and auxiliary equipment in the plant is given in Figure 1.

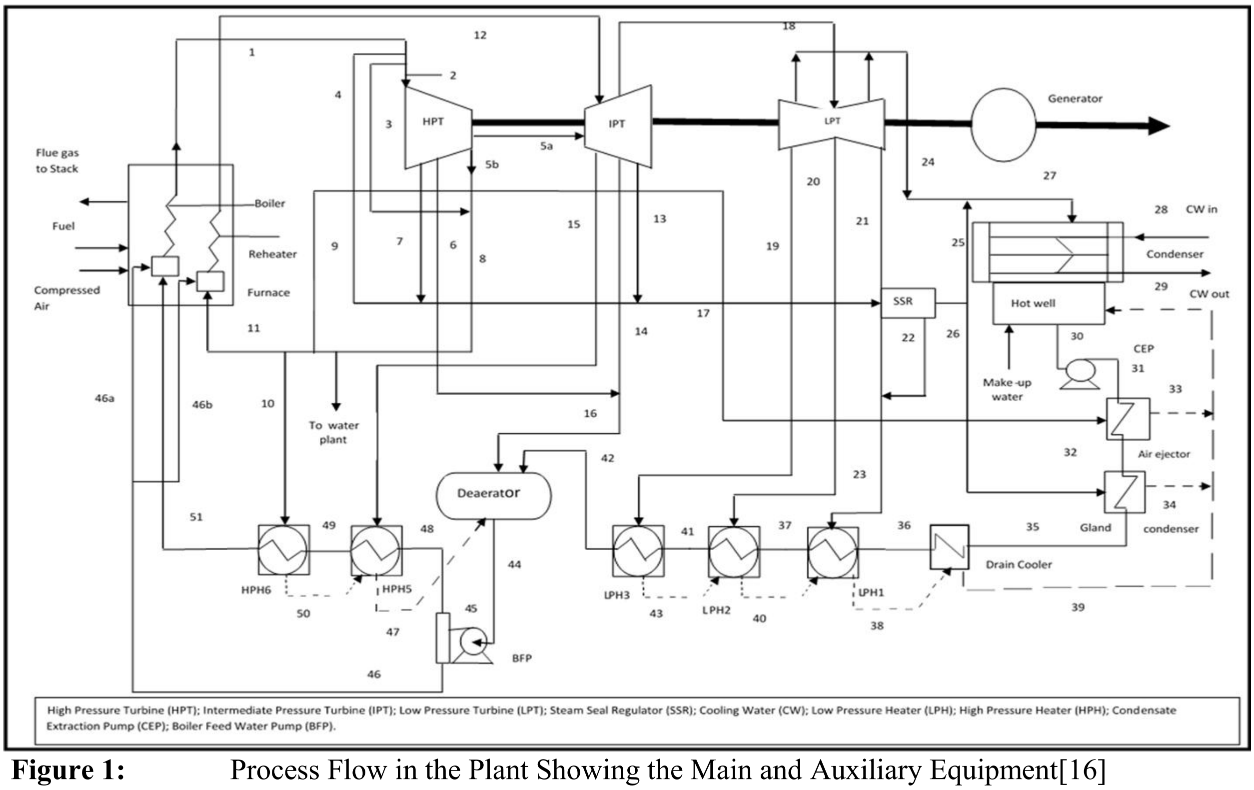

Figure 1: Process Flow in the Plant Showing the Main and Auxiliary Equipment[16]

Table 1: Natural Gas Composition

Component	CH_4	C_2H_6	C_3H_8	CO_2	N_2
Mole fraction	0.894	0.086	0.004	0.006	0.010

2.3. Method

The sub-systems and entire plant understudied is an open system with no material accumulation; therefore, the material and energy balances as set up in equation (1) and (2) were pivotal to the performance evaluation carried out on the plant. Having supplied the available thermodynamic data at strategic points, HYSYS returned the mass flow rates and enthalpies at other locations using those equations.

$$\sum_{i=1}^{n} \dot{m}_{in} - \sum_{i=1}^{n} \dot{m}_{out} = 0 \qquad (1)$$

$$\sum_{i=1}^{n} \dot{Q}_i + \sum_{i=1}^{n} \dot{W}_s = \sum_{i=1}^{n} \dot{m}_{out}\, h_{out} - \sum_{i=1}^{n} \dot{m}_{in}\, h_{in} \qquad (2)$$

Where $\dot{m}$is the mass flow rate of the stream; $\dot{Q}$, $\dot{W}_s$, and h is the heat flow rate, shaft work and specific enthalpy, respectively; i and n is the ith, and nth component, respectively; in and out represents the inlet and outlet stream, respectively.

In addition to the material and energy balances, exergy balance was also set up as given in equation (3) and the computation of specific physical exergy was done using equation (4)

$$\sum_{i=1}^{n} \dot{Q}_i \left(1 - \frac{T_o}{T_i}\right) + \sum_{i=1}^{n} \dot{W}_s + \sum_{i=1}^{n} \dot{m}_{in}\, ex_{in} - \sum_{i=1}^{n} \dot{m}_{out}\, ex_{out} = I \qquad (3)$$

$$ex_i^{Phy} = \Delta h - T_o \Delta s = (h_i - h_o) - T_o(s_i - s_o) \qquad (4)$$

Where T is temperature, o represent ambient condition, ex is specific exergy, s is specific entropy

In evaluating the exergy of streams and performance of the plant and component equipment. Two significant exergy forms were considered, the physical exergy and the chemical exergy. When computing the total specific exergy of a stream using equation (5) either of the two forms of exergy may be negligible.

$$ex_j^{Total} = ex_j^{Phy} + ex_j^{chem} \qquad (5)$$

As suggested by [17], the molar chemical exergy of a gas mixture was computed using equation (6) and the standard molar chemical exergy, $\bar{e}_i^{CH}$ of chemical substances was obtained from the literature [18].

$$. ex_j^{chem} = \textstyle\sum_{i=1}^{j} y_i \bar{e}_i^{CH} + RT_o \sum_{i=1}^{j} y_i \ln y_i \qquad (6)$$

Where y_i is the mole fraction of component i and R is the universal gas constant

The exergy rate of a stream j was obtained from its specific value using equation (7)

$$\dot{Ex}_j = \dot{m}_j \left(ex_j^{Total}\right) \qquad (7)$$

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Irreversibility, I, in the plant and sub-systems, was calculated according to [19] by setting up exergy balance on the subsystem or plant (as the case may be), and taking the difference between all incoming and all outgoing exergy flows using equation (8).

$$I = \sum_{in} \dot{E}x_j - \sum_{out} \dot{E}x_j \tag{8}$$

Where I is the irreversibility rate and is the $\dot{E}x$ exergy rate.

The exergy efficiency was obtained according to [20] and [21] using equation (9) while energy efficiency was computed using equation (10)

$$\psi = \frac{\sum \dot{E}x_{out}}{\sum \dot{E}x_{in}} = \frac{\sum \dot{E}x_{sink}}{\sum \dot{E}x_{source}} \tag{9}$$

$$\eta = \frac{\sum \dot{E}_{out}}{\sum \dot{E}_{in}} = \frac{\sum \dot{E}_{sink}}{\sum \dot{E}_{source}} \tag{10}$$

Where $\dot{E}_{sink}$ is the energy rate of the sink and $\dot{E}_{source}$ is the energy rate of the source.

The resulting component model equations by the adaptation of equations 1 through 10 for each equipment making up the plant were solved using EXCEL spreadsheet alongside a modified form of HYSY VBA codes put forward by [22].

3. Results

The boiler and the steam cycle, as described by Figure 1, was simulated in HYSYS using the operating conditions obtained from the plant. To evaluate the performance of the power plant based on the given operating conditions, the Peng Robinson fluid package and ASME fluid package was used to obtain necessary and accurate thermodynamic data for the combustion processes and the steam cycle, respectively. The process flow diagram of the simulation is shown in Figure 2; The results obtained by solving the model equations obtained from 1 through 10 are given in Table 2 and Figure 3. Table 2 shows the thermodynamic data of the steam cycle and natural gas combustion. Based on these thermodynamic data, comparison of energy and exergy efficiencies, component-wise performance and overall plant's performance are given in Figure 3.

4. Discussion

The analysis of the thermodynamic performance of the plant at full capacity was carried out with reference to a dead state of 25 ℃ and 1 atm using recent operational data from the plant. The thermodynamic properties of streams at various points in the plant with reference to this dead state is as shown in Table 2. As expected, the specific enthalpies computed using VBA codes embedded in HYSYS were higher than the specific exergies for each stream in the steam cycle. This is because energy from heat source such as steam is not entropy free; the exergy, is thus, an indication of the grade of energy possessed by the steam at its temperature and pressure [23]. In the steam cycle where no chemical interaction takes place, the changes in physical exergy accompanying the physical interactions were accounted for. For such processes where chemical changes do not occur, the contribution of the chemical exergy to change in the total exergy possessed by the stream as physical transformation takes place is usually negligible.

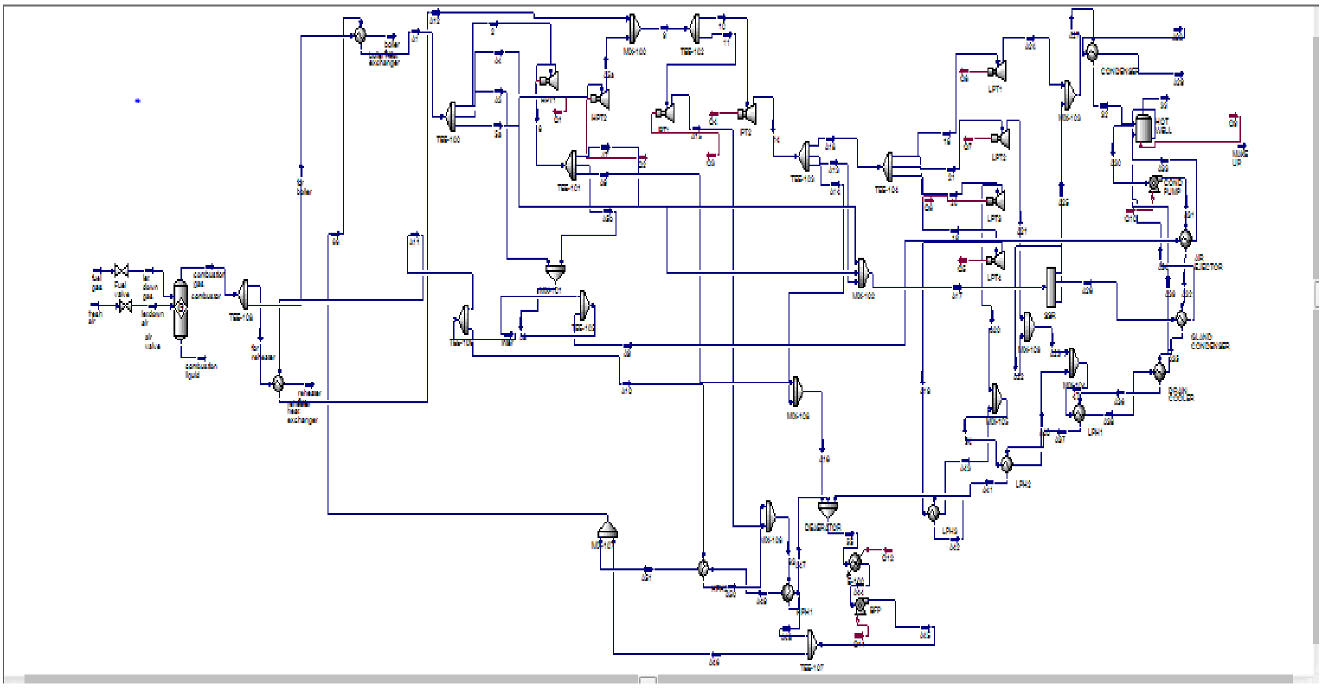

Figure 2: The Plant Simulated using HYSYS

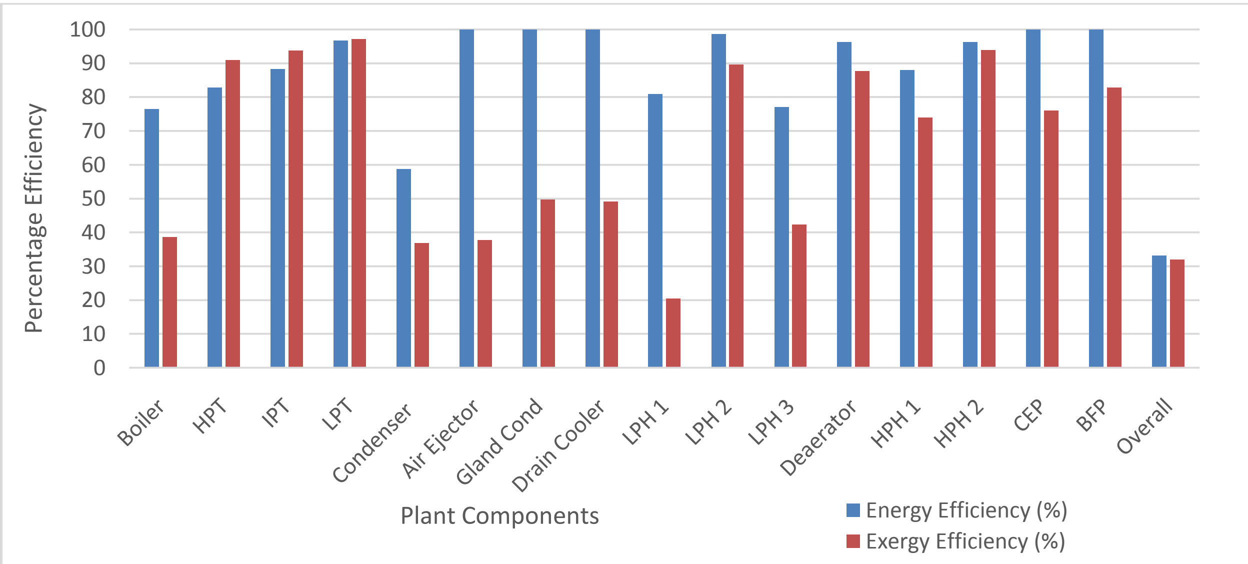

Figure 3: Comparison of the Energetic and Exergetic Efficiencies of Plant Component units

Table 2: Thermodynamic Data on the Plant

Stream	Mass Flow (kg/h)	T (℃)	P(kPa)	ΔS (kJ/kg.℃)	ΔH (kJ/kg)	Energy Rate (kW)	ΔEx (kJ/kg)	Physical Exergy Rate (kW)
S1	653902	538	12500	6.22667	3339.50	606585.17	1483.02	269375.075
S2	652250.84	538	12500	6.22667	3339.50	605053.493	1483.02	268694.881
S3	1259.32	538	12500	6.22667	3339.50	1168.196	1483.02	518.778
S4	391.84	538	12500	6.22667	3339.50	363.482	1483.02	161.417
S5a	11216.75	469.91	3336	6.72764	3280.93	10222.615	1275.09	3972.872
S5b	635543.37	351.30	3336	6.32892	3009.23	531249.723	1122.27	198124.984
S6	3711.31	351.30	3336	6.32892	3009.23	3102.280	1122.27	1156.969
S7	1779.41	351.30	3336	6.32892	3009.23	1487.406	1122.27	554.715
S8	636802.69	351.58	3336	6.32996	3009.89	532417.919	1122.61	198577.951
S9	545.34	351.58	3336	6.32996	3009.89	455.944	1122.61	170.055
S10	50805.09	351.58	3336	6.32996	3009.89	42477.117	1122.61	15842.853
S11	585452.27	351.58	3336	6.32996	3009.89	489484.858	1122.61	182565.043
S12	585452.27	538	3069	6.96562	3436.59	558877.828	1359.79	221136.894
S13	1091.68	331.70	691.3	7.05320	3022.23	916.477	919.32	278.780
S14	22816.24	331.70	691.3	7.05320	3022.23	19154.453	919.32	5826.521
S15	30237.86	436.77	1535	7.00927	3231.33	27141.286	1141.52	9588.099
S16	26527.55	330.83	691.3	7.05019	3020.42	22256.733	918.40	6767.478
S17	3262.93	346.55	691.3	7.10385	3053.24	2767.365	935.23	847.664
S18	542523.23	331.70	691.3	7.05320	3022.23	455453.443	919.32	138542.667
S19	22552.66	256.52	363.3	7.08655	2875.03	18011.023	762.18	4774.775
S20	21160.04	179.45	172.9	7.12209	2726.33	16024.816	602.89	3543.629
S21	35082.26	104.92	75.81	7.15731	2584.93	25190.371	450.98	4394.866
S22	1253.26	346.55	691.3	7.10385	3053.24	1062.922	935.23	325.581
S23	36335.52	113.60	75.81	7.19957	2601.09	26253.293	454.53	4587.701
S24	463728.27	50.27	12.5	7.11740	2297.43	295940.296	175.38	22591.579

Stream	Mass Flow (kg/h)	T (℃)	P(kPa)	ΔS (kJ/kg.℃)	ΔH (kJ/kg)	Energy Rate (kW)	Specific Physical Exergy (kJ/kg)	Physical Exergy Rate (kW)
S25	1050.28	346.55	691.30	7.10385	3053.24	890.764	935.23	272.848
S26	959.39	346.55	691.30	7.10385	3053.24	813.679	935.23	249.236
S27	464778.55	50.27	12.50	7.12268	2299.14	296831.061	175.52	22659.972
S28	32594223.51	30.00	4.24	0.06950	20.81	188427.630	0.08	687.457
S29	32594223.51	37.60	6.48	0.17304	52.58	476049.047	0.99	8938.861
S30a	464778.55	42.00	8.20	0.23296	71.33	9209.644	1.88	242.463
S30	546331.49	42.00	8.20	0.23181	70.97	10770.746	1.86	282.047
S31	546331.49	42.00	8.70	0.23181	70.97	10770.848	1.86	282.125
S32	546331.49	42.10	29.70	0.23314	71.41	10837.404	1.90	288.766
S33	545.34	99.10	97.47	7.00438	2570.53	389.389	482.17	73.040
S34	959.39	99.10	94.47	7.01965	2570.92	685.141	478.01	127.388
S35	546331.49	42.30	54.70	0.23574	72.26	10965.942	1.97	299.396
S36	546331.49	79.05	45.56	1.28049	431.57	65495.100	49.80	7557.048
S37	546331.49	90.64	71.84	1.27090	434.08	65876.160	55.17	8372.036
S38	80048.22	90.40	22.00	7.66666	2563.47	57000.346	277.65	6173.820
S39	80048.22	51.60	33.30	0.35727	111.14	2471.187	4.62	102.639
S40	43712.70	94.70	83.00	7.05874	2563.58	31128.113	459.02	5573.645
S41	546331.49	114.01	163.66	1.27298	442.12	67095.572	62.58	9497.095
S42	546331.49	132.72	343.36	1.29640	453.24	68783.885	66.72	10125.755
S43	22552.66	118.00	126.00	6.97942	2605.54	16322.709	524.62	3286.556
S44	653902.00	163.00	667.00	1.60584	583.83	106047.207	105.05	19081.525
S45	653902.00	165.47	12600.00	1.61598	601.43	109243.884	119.63	21729.033
S46	20197.62	165.47	12600.00	1.61598	601.43	3374.308	119.63	671.163
S47	81042.95	171.50	821.20	1.68986	621.00	13980.001	117.17	2637.797
S48	633704.38	165.47	12600.00	1.61598	601.43	105869.576	119.63	21057.870

Stream	Mass Flow (kg/h)	T (℃)	P(kPa)	ΔS (kJ/kg.℃)	ΔH (kJ/kg)	Energy Rate (kW)	Specific Physical Exergy (kJ/kg)	Physical Exergy Rate (kW)
S49	633704.38	196.59	1447	1.95832	744.77	131100.913	160.90	28322.290
S50	50805.09	212.17	1992	2.18621	855.27	12070.052	203.45	2871.230
S51	633704.38	236.60	3152	2.30585	917.51	161507.978	230.02	40490.179

Stream	Mass Flow (kg/h)	T(℃)	P(kPa)	ΔS (kJ/kg.℃)	ΔH (kJ/kg)	Energy Rate (kW)	Physical Exergy (kJ/kg)	Chemical Exergy (kJ/kg)	Total Exergy Rate (kW)
Air	1330356	30	865	-0.60682	3.02	1116.4	183.95	4.45	69621.772
Natural gas	50190	27	243	-0.40161	2.62	36.508	122.36	49853.61	696748.335
Combustion product	1380537	1432.172	243	2.28995	1888.807	724324.2	1206.058	44.81	277782846.5
Flue gas	1380537	387.87	241	1.09377	551.85	211624.8	225.74	44.81	103752.054

For the fuel and air streams, the specific exergy possessed with respect to the dead state is higher than the specific enthalpy. This is due to the negative difference between their entropy at the boiler inlet state and the dead state. The fuel has chemical exergy far greater than its physical exergy. This is because as the main energy source for the process, it has a potential heating value that can be harnessed through combustion. Based on the composition of the natural gas, the higher heating value of the fuel was$52876.31\ kJ/kg$ while its lower heating value was $48084.94\ kJ/kg$as returned by HYSYS. Unlike natural gas, the air which supplies the oxygen for the combustion process does not possess any heating value. Consequently, the chemical exergy due to its composition (nitrogen and oxygen) was about $9/10000$ththat of natural gas. Physical exergy dominates the total exergy in the air because it has no heating value, and its inlet temperature is higher than the dead state temperature. The air-fuel ratio used for the combustion process was 26.5064. The total exergy rate of air into the boiler was approximately $1/10$th that of natural gas. Relative to the fuel, the higher physical exergy and mass flow rate of air account for the marked change in statistics between the chemical and total exergy of air. Notwithstanding this change in statistics, the contribution of air as exergy source of the system can be deemed negligible compared to natural gas.

The combustion process produced combustion gas at a temperature of 1432.17 °Cwith specific physical exergy almost ten times that of the natural gas but with much lower chemical exergy. The combustion gas leaves as flue gas with roughly $1/5$th of the specific physical exergy of the combustion gas having been used to raise the temperature and enthalpy of feed water and cold reheat steam. During the heat transfer by the combustion gas, $653902\ kg/h$ superheated steam (at 541 °C&$12921\ kPa$) with a specific enthalpy of $3342.76\ kJ/kg$ and $585452.27\ kg/h$ hot reheat steam (at 541 °C&$3122\ kPa$) with a specific enthalpy of $3442.81\ kJ/kg$ was produced from the feed water (at $234\ 68$ °C&$3152\ kPa$) with a specific enthalpy of $907.74\ kJ/kg$ and cold reheat steam (at 351.58°C&3336 kPa) with a specific enthalpy of $3009.89\ kJ/kg$ respectively. The superheated steam and hot reheat steam passes through attenuators. The superheated steam then enters into the high-pressure turbine at 538°C and $12500\ kPa$ with specific enthalpy of $3339.50\ kJ/kg$. The reheat steam, on the other hand, enters the intermediate pressure turbine at 538 °C and $3064\ kPa$ with specific enthalpy $3436.59\ kJ/kg$.

The component-wise and overall plant performance based on the thermodynamic data presented in Table 2 is shown in Figure 3. The component-wise comparison of the energy and exergy efficiency show that the exergy efficiency of the turbines is higher than the energy efficiency. In contrast, energy efficiency was higher than the exergy efficiency in other components of the plant. This is so because turbines are work producing devices. In the case of turbines, the exergy efficiency is related to the ideal or maximum work derivable from the device based on the inlet and exhaust conditions of the steam. On the other hand, the isentropic efficiency relates to the actual work extracted from the turbine based on the steam inlet and outlet conditions. The cumulative work produced by the turbine was 225.71MW. The net work required by the condensate extraction pump and the boiler feedwater pump was approximately 3.2 MW. This brings the net power generated in the cycle to 222.52 MW.The boiler has an energy efficiency of up to 76.48 % and exergy efficiency of 38.61 %. The condenser has the least energy efficiency while the least exergy efficiency was observed in a low-pressure heater, LPH 1. Although LPH 1 had the least exergy efficiency, the value of irreversibility in the heater was 3.17MW while irreversibility was highest in the boiler with a value of 427.7 MW. The overall plant exergy efficiency for the plant is observed to be lower than energy efficiency. The overall energy efficiency of the plant was found to be 33.19% while the overall exergy efficiency was 31.94%.

5. Conclusion

In this study, approximately 222.52 MW of electricity was generated by harnessing the heating value of 50190 kg/hnatural gas (nearly 2514.08 million BTU/hof natural gas) through the combustion process and the steam cycle in a major thermal power plant in Nigeria. Based on the component-wise irreversibility, the significant contribution to the overall inefficiency is from the boiler. Proper maintenance of the boiler and improvement of its operation will enhance the overall efficiency of the plant. The component-wise efficiencies also indicate that there are rooms for thermodynamic improvement for optimum energy transformation into electricity. Understanding the impact of various operating parameters and the effect of load variation on the plant's performance will help in setting and maintaining these parameters at the optimum for improved performance.

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