The International Journal Of Engineering And Science (IJES)
|| Volume || 4 || Issue || 9 || Pages || PP -18-29|| 2015 ||
ISSN (e): 2319 – 1813 ISSN (p): 2319 – 1805

Enhancing the Performance of 75mw Steam Power Plant with Second Law Efficiency, Condenser Pressure and Rankine Cycle

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ABSTRACT

Energy analysis, thermodynamic second law and Rankine cycle are universal method for enhancing the performance of power plant. Experimental data of 75MW power plant was obtained, mass; energy balance and thermodynamic second law were used to analysis the exergy efficiency while Rankine cycle was used to calculate the overall efficiency of the plant as well as varying the condenser pressure. Results obtained showed that an increase in the reference environment temperature increases the thermal, Rankine and exergy efficiency of the plant. Also as the condenser pressure increases, the efficiency of the plant decreases. The condenser pressure must be reduced in order to decrease the cause's irreveribilities in the system. Also, the plant should be operated above 50% of the operating capacity of the plant, to minimize wastage of energy consumption because more energy will be generated in the boiler even when not in used.

Keywords *– Energy analysis, energy efficiency, exergy analysis, exergy efficiency, power plant, rankine cycle, laws of thermodynamic.*

Date of Submission:21 August 2015 Date of Accepted: 30 September 2015

I. INTRODUCTION

Regardless of improvement in renewable energy stations like wind, sun based power, and tidal power, the energy for the world depends intensely on fossil fuels for her power generation. It is however foreseen that the world may continue to depend extensively on fossil fuels, for the generation of energy for decades. Despite the exhaustion of fossil fuel and ecological concerns, for instance, environmental change, the dependence on oil is put at 60% between independence and 2014, from which 94% is attributed to regular gas and 96% for coal Rosen, M.A [1]. The fossil fuels are predominately use in energy generation and it is ideal to quantify same in power plants as to ascertain areas where losses can be cubed. Energy and exergy are the two basic processes utilized in the analysis of steam power plant. In energy analysis first law of thermodynamics cannot be considered as a tool because there is no correlation between it and energy loss in a system. Moreover, if it is applied there wouldn't be clear separation between the quality and amount of energy inside the system, and it can't describe the irreversibility of the procedure therein. However, exergy analysis will evaluate the work capacity of a system using second law of thermodynamics and the most extreme work that can be performed by the system. In time past. exergy studies were used in the analysis of power plants, as a means to improve the energy generation and turbine power. Habib et.al .and Zubair et.al [2] used second law of thermodynamic in the analysis of regenerative Rankine power plants with reheating process. Sengupta et al. [3] led an exergy evalution of a 210 MW thermal power plant in India. Rosen et.al. [4], [5], [6], [7] performed exergy analysis of power plants that work on diverse fuels and further researched on costs analysis of thermal power plant and thermodynamic losses.

In recent time, the exergetic analysis has been found to be an important valuable tool in the design, evaluation, and optimization of thermal power plants. Consequently, exergetic analysis can give a complete and ideal state of a system. Generally, the Exergy analysis is an important tool for advancement, evaluation and improvement *of present energy utilization of power plants Vuckokic G. D et al, [8]. For complex energy systems, with broad number of parts, exergy destruction of certain segments depends on the systems qualities and the distinctive segments inefficiencies. The traditional exergy analysis demonstrates certain hindrances which significantly contribute in the overall performance in exergy analysis [9]. Mohammed Y. et al. [10] showed all the essential component in exergy analysis of a novel co-generation idea that joined LNG regasification with the generation of power.*

(Osueke C. et al, [7] studied Energy and Exergy analysis of a Sapele steam power plant. The main aim was to identify areas where energy loss are occurring and develop them for proficient and viable change in a thermal power station. Experimental data was collected from Sapele steam power plant. The mass balance, energy analysis and the efficiencies of the overall plant was determined. Likewise energy losses of all the major components on the power plant were properly identified and mathematical equations leading to the novel analysis developed. It was deduced that energy losses mainly occurred in the boiler where 105KW is lost to the environment while only 15.7 KW was lost from the condenser system. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (47%) followed by the turbine (42%), and then the condenser (7%). In addition, the exergy efficiency of the power cycle was 25%. No drastic change was noticed in the performance of major components for a moderate change in the reference environmental state temperature, and the main conclusion remained the same, meaning that the boiler is the major source of irreversibilities in the power plant while chemical reaction is the most significant source of exergy destruction

Amir V. et al. [11] portrayed Carnot Cycle as the best cycle for the determination of temperature variation in any thermal system. George and Park [12] examines how to estimate the avoidable and unavoidable exergy destruction and investment costs connected with compressors, turbines, heat exchangers and burning chambers. This general procedure, though considering various subjective decisions, supports and upgrades usages of exergoeconomics. Kotas [13] clarified in this work the idea of exergy used to characterize criteria of performance of thermal plant. The dispersion of the exergy losses in a few plant components amid the constant plant running conditions has been evaluated to find the procedure irreversibility. The correlation between the energy losses and the exergy losses of the individual segments of the plant shows that the greatest energy losses of 39% happen in the condenser, while the most extreme exergy losses of 42.73% happen in the combustor. In the analysis, exergy systems notwithstanding, more routine energy analysis are utilized to assess general and components efficiencies and to distinguish and evaluate the thermodynamic losses. Datta et al. [14] studied exergy analysis of a coal-based thermally constrain plant using the setup data from a 210 MW thermal power plant under operation in India. The exergy efficiency was determined using the working data from the plant at distinctive conditions, at distinctive loads and distinctive condenser pressures. It is watched that the genuine wellspring of irreversibility in the power cycle is the boiler, which adds to exergy pulverization of the solicitation of 60%. Part stack operation builds the irreversibilities in the cycle and the impact is more professed with the diminishment of the heap. Increment in the condenser back pressure diminishes the exergy efficiency. Aljundi [15] in his work studied the energy and exergy analysis of Al-Hussein power plant in Jordan. The main objectives of this paper are to investigate the system components independently and to distinguish and evaluate the areas having biggest energy and exergy losses. Dai et al. [16] in their work analysed exergy for an ideal system, and a parameter optimization for every ideal system is accomplished by method of hereditary algorithm so as to achieve the maximum exergy efficiency. The bond production is an energy concentrated industry with energy regularly representing 50-60% of the production costs.

This research work deals with the enhancement of the Performance of 75MW steam power plant with second law method, Exergy analysis, Rankine Cycle and Condenser Pressure. It enumerated the causes of energy and exergy losses in the system as well as the effect of power output on the plant efficiencies.

XI RESEARCH METHODOLOGY

1) Plant Description

Sapele power plc, Sapele is a thermal generating station located in Nigeria's gas-rich Delta State. Sapele has an installed capacity of 1020MW. It powers six, 120MWsteam turbines which generate a daily average of 86.72MWH/H or approximately 2500GW/H annually. Sapele power plant currently operates at peak capacity of 972MW.

Sapele power plan is strategically located in Niger Delta region close to sources of both natural gas feed stock and a river for cooling its steam turbine generators. Sapele power plant includes an updated control room, a switch gear room, a staff training school and medical and recreation facilities. It began operations in 1978. Figure 3.1 displays a schematic diagram for a 70MW unit of a power plant.

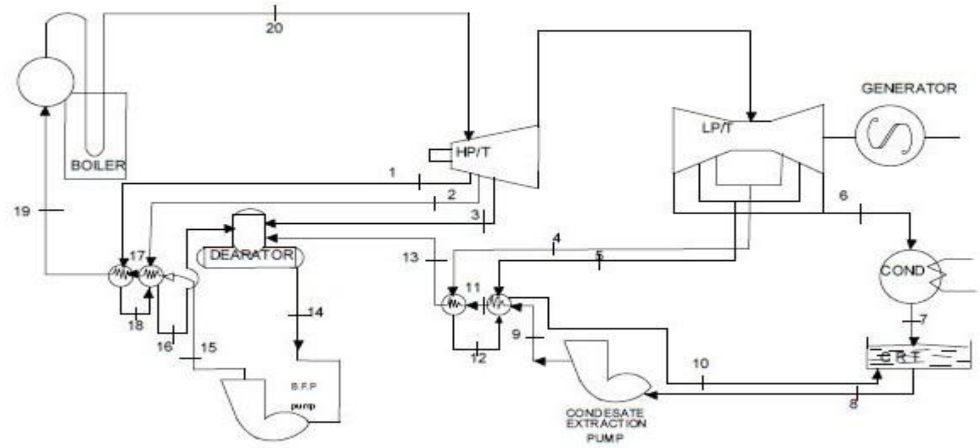

Figure 1. Schematic diagram of the power plant

TABLE 1. Operating Condition of the Power Plant

Operating condition	**Value**
Acting Power	70MW
Reacting Power @ generator	15MVAR
Frequency	50.9
Turbine Power Output	120
Feed Water Pressure	200Kg/cm^2
Extraction Steam Pressure	5Kg/cm^2
Extraction Steam mass flow rate	103.438Kg/s
Thermal Efficiency	35%

Source: Sapele 1978

TABLE 2 Properties of Heavy Oil Used In Sapele Power Plant for March 2015

Property	**Value**
Flash point	210℃
Kinematic Viscosity @ 40℃	65.69cSt
Boiling point	316℃
Specific gravity	0.87
Density @15℃	869kg/m^3
vapour pressure@20℃	0.1mmHg
Vapour density	1

Source: Sapele 1978

2) Rankine Cycle Analysis of the Superheated Steam Power Plant

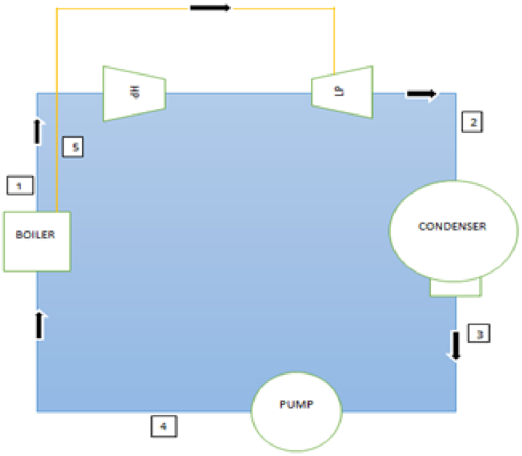

Figure 3. Schematic diagram of power plant with reheat

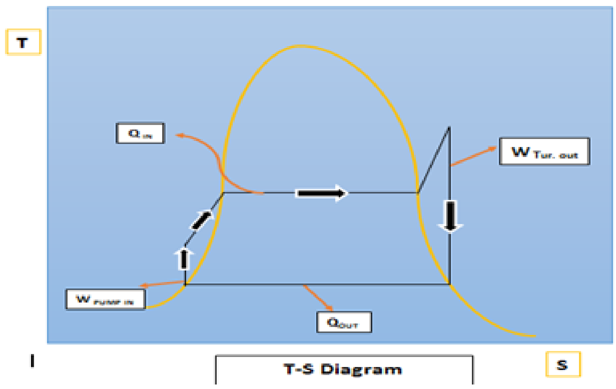

Figure 4. Schematic diagram of rankine cycle

$$Q + W = dh$$
$$Q_{451} + W = h_1 - h_4 \quad (1)$$
$$Q_{12} + W_{12} = h_2 - h_1$$

Work output, $-W_{12} = h_1 - h_2$ (2)

Condenser:

$$Q_{23} + W = h_3 - h_2$$

Since $W = 0$

$$Q_{23} = h_3 - h_2$$

Therefore

Heat rejected in condenser, $-Q_{23} = h_2 - h_3$ (3)

Pump:

$$Q_{34} + W_{34} = h_4 - h_3$$

The compression is isentropic (i.e, $s_3 = s_4$) and adiabatic(i.e, Q=0). Therefore

$$W_{34} = h_4 - h_3$$

i.e work input to pump, $W_{34} = h_4 - h_3$ (4)

This is the pump-term and as it is a small quantity in comparison with the turbine work output, $-W_{12,}$ it is usually neglected, especially when boiler pressure are low.

Net work input for the cycle $\sum W = W_{12} + W_{34}$

i.e $\sum W = (h_2 - h_1) + (h_4 - h_3)$ (5)

or Network output, $-\sum W = h_1 - h_2$ (6)

The heat supplied in the boiler, $Q_{451} = h_1 - h_2$. Then we have

Rankine efficiency, n_r = net work output/heat supplied in the boiler (7)

$$n_R = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)} \text{ or}$$

$$n_R = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_3) - (h_4 - h_3)} \quad (8)$$

If the feed-pump term, $h_4 - h_3$, is neglected

$$n_R = \frac{h_1 - h_2}{h_1 - h_3} \tag{9}$$

When the feed-pump term is to be included it is necessary to evaluate the quantity, W_{34}.

From equation (4)

Pump work $= W_{34} = h_4 - h_3$

It can be shown that for a liquid, which is assumed to be incompressible (i.e. υ =constant), the increase in enthalpy for isentropic compression is given by

$$(h_4 - h_3) = \upsilon(p_4 - p_3)$$

The proof is as follows. For a reversible adiabatic process

$$dQ = dh - \upsilon dp = 0$$

Therefore

$$dh - \upsilon dp$$

i.e. $\int_3^4 dh = \int_3^4 \upsilon dp$

for a liquid, since υ is approximately constant, we have

$$h_4 - h_3 = \upsilon\int_3^4 dp = \upsilon(p_4 - p_3)$$

i.e. $h_4 - h_3 = \upsilon(p_4 - p_3)$

therefore

Pump work input = $h_4 - h_3 = \upsilon(p_4 - p_3)$ (10)

where υ can be taken from tables for water at the pressure p_3

III . ENERGY ANALYSIS OF COMPONENT IN THE POWER PLANT

1) Steam Turbine

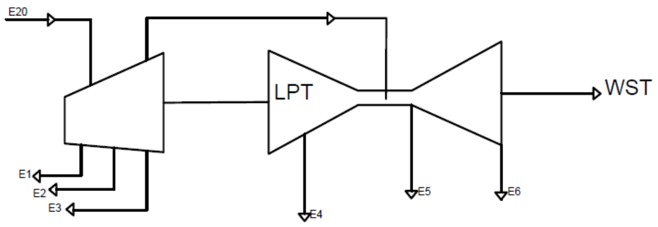

Figure 4. Schematic diagram of Turbine

1.1. Mass balance:

$$M_{20} = M_1 + M_2 + M_3 + M_4 + M_5 + M_4 \tag{11a}$$

1.2. Energy Balance:

$$M_{20}h_{20} = M_1h_1 + M_2h_2 + M_3h_3 + M_4h_4 + M_5h_5 + M_6h_6 + W_{turbine} \tag{11b}$$

1.3. Exergy Destruction**:**

$$E_{in} - E_{out} + W_{turbine} \tag{11c}$$

$$E_{in} = E_{20} \tag{11d}$$

$$E_{out} = E_1 + E_2 + E_3 + E_4 + E_5 + E_6 \tag{11e}$$

1.4. Work Output (W) = $W_{turbine}$ = 16.78KW

2) Condenser

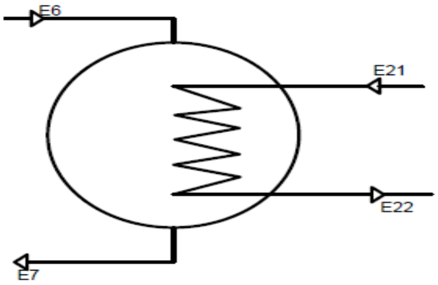

Figure 5. Schematic diagram of condenser

2.1. Mass Balance = $M_6 + M_{21} = M_7 + M_{22}$ (12a)
2.2. Energy Balance = $M_6h_6 + M_{21}h_{21} = M_7h_7 + M_{22}h_{22}$ (12b)
$E_{in} = E_6 + E_{21}$ (12c)
$E_{out} = E_7 + E_{22}$ (12d)
2.3. Exergy Destruction. = $E_{in} - E_{out} + W_{turbine}$ (12e)

3) Deaerator

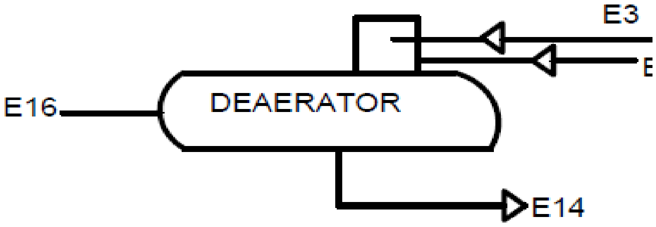

Figure 6. Schematic diagram of deaerator

3.1. Mass Balance = $M_3 + M_{13} + M_{16} = M_{14}$ (13a)
3.2. Energy Balance = $M_3h_3 + M_{13}h_{13} + M_{16}h_{16} = M_{14}h_{14}$ (13b)
3.3. Exergy Destruction = $E_{in} - E_{out} + W_{turbine}$ (13c)
$E_{in} = E_3 + E_{13} + E_{16}$ (13d)
$E_{out} = E_{14}$ (13e)

4) Boiler Feed Pump

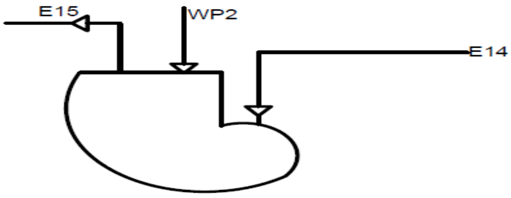

Figure 7. Schematic diagram of boiler feed pump

4.1. Mass Balance = $M_{14} = M_{15}$ (14a)
4.2. Energy Balance = $M_{14}h_{14} = M_{15}h_{15}$ (14b)
4.3. Exergy Destruction. = $E_{in} - E_{out} + W_{PZ}$ (14c)
$E_{in} = E_{14}$ (14d)
$E_{out} = E_{15}$ (14e)

5) Condensate Receive Tank (C.R.T)

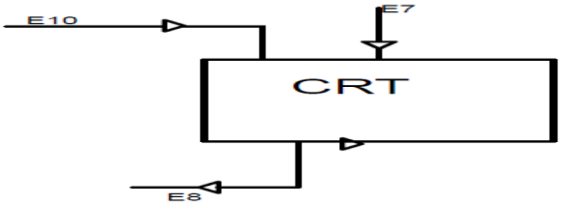

Figure 8. Schematic diagram of condensate receive tank

5.1. Mass Balance = $M_7 + M_{10} = M_8$ (15a)
5.2. Energy Balance = $M_7 h_7 + M_{10} h_{10} = M_8 h_8$ (15b)
5.3. Exergy Destruction. = $E_{in} - E_{out} + W_{PZ}$ (15c)
$E_{in} = E_7 + E_{10}$ (15d)
$E_{out} = E_8$ (15e)

6) High Pressure Heater 1

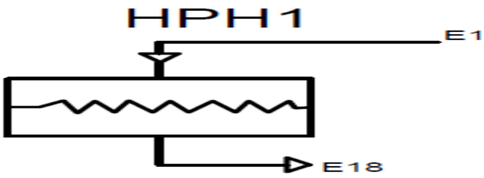

Figure 9. Schematic diagram of high pressure heater 1

6.1. Mass Balance = $M_1 + M_7 = M_{18}$ (16a)
6.2. Energy Balance = $M_1 h_1 + M_{17} h_{17} = M_{18} h_{18}$ (16b)
6.3. Exergy Destruction. = $E_{in} - E_{out}$ (16c)
$E_{in} = E_1 + E_{17}$ (16d)
$E_{out} = E_{18}$ (16e)

7) High Pressure Heater 2

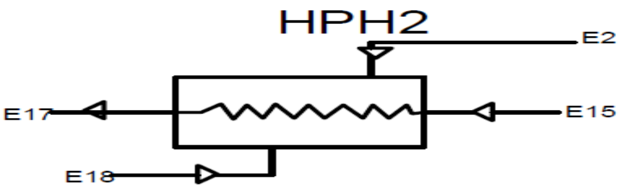

Figure 10. Schematic diagram of high pressure heater 2

7.1. Mass Balance = $M_2 + M_{15} + M_{18} = M_{17}$ (17a)
7.2. Energy Balance = $M_2 h_2 + M_{15} h_{15} + M_{18} h_{18} = M_{17} h_{17}$ (17b)
7.3. Exergy Destruction. = $E_{in} - E_{out}$ (17c)
$E_{in} = E_2 + E_{15} + E_{18}$ (17d)
$E_{out} = E_{17}$ (17e)

8) Low Pressure Heater 1

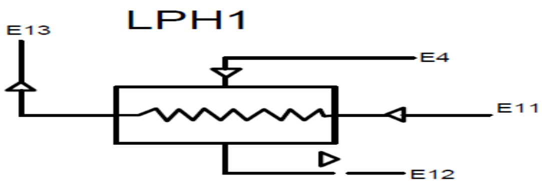

Figure 11. Schematic diagram of low pressure heater 1

8.1. Mass Balance = $M_4 + M_{11} = M_{12} + M_{13}$ (18a)
8.2. Energy Balance = $M_4 h_4 + M_{11} h_{11} = M_{12} h_{12} + M_{13} h_{13}$ (18b)
8.3. Exergy Destruction. = $E_{in} - E_{out}$ (18c)
$E_{in} = E_4 + E_{11}$ (18d)
$E_{out} = E_{12} + E_{13}$ (18e)

9) Low Pressure Heater 2

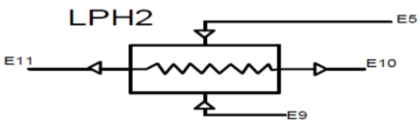

Figure 12. Schematic diagram of low pressure heater 2

9.1. Mass Balance = $M_5 + M_9 = M_{10} + M_{11}$ (19a)
9.2. Balance = $M_5 h_5 + M_9 h_9 = M_{10} h_{10} + M_{11} h_{11}$ (19b)
9.3. Exergy Destruction. = $E_{in} - E_{out}$ (19c)
$E_{in} = E_5 + E_9$ (19d)
$E_{out} = E_{10} + E_{11}$ (19e)

Mass, energy, and exergy balances for any control volume at steady state with negligible potential and kinetic energy changes can be expressed, respectively, by

$$\sum \dot{m}_i = \sum \dot{m}_e \quad (20a)$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_e \dot{h}_e - \sum \dot{m}_i \dot{h}_i \quad (20b)$$

$$X_{heat} - w = \sum \dot{m}_e \Psi_e - \sum \dot{m}_i \Psi_i + \dot{I} \quad (20c)$$

where the net exergy transfer by ($\dot{X}_{heat}$) at temperature T is given by

$$\dot{X}_{heat} = \sum \left(1 - {}^{T_o}\!/_{T} \right) Q \quad (20d)$$

and the specific exergy is given by

$$\Psi = h - h_o - T_o(s - s_o) \quad (20e)$$

Then the total exergy rate associated with a fluid stream becomes

$$\dot{X} = \dot{m}\Psi = m[h - h_o - T_o(s - s_o)] \quad (20f)$$

$$\%Ed = \frac{Ed}{E_{in}} x100\% \quad (20g)$$

Thermal efficiency and exergy efficiency of the power plant can be calculated as

$$n_{th,pp} = \frac{\dot{W}_{net}}{Q_{in}} \quad (21)$$

$$n_{ex,pp} = 1 - \frac{\dot{E}_{x,d}}{\dot{E}_{x,in}} \quad (22)$$

$$\dot{Q} = h_5 + h_3 - h_2 \quad (23)$$

10) Analysis with a full load operation condition

The entire system had been covered in analysis with full load operation condition. The power plant was analyzed using the above relation nothing that the environment reference temperature and pressure are 298K and 1.013bar respectively. The distribution of energy addition exergy losses and exergy consumption for different components has been worked out on the basis of analysis exergetic efficiency for boiler; turbine and other components have been calculated.

IV . RESULTS AND DISCUSSION

TABLE 3. Energy analysis of the power plant when T = 298.15K, P= 101.3KPa

POINT	T(K)	P(MPa)	M(ton/h)	h(kj/kg)	s(kj/kgk)	Ψ(kj/kg)	X(MW
1	628.4	3.0071	16.72	3128.6	6.7643	1117.434	5.189861
2	514.3	1.8713	13.82	2887.5	6.5614	936.7984	3.596265
3	439.9	0.4219	15.41	2789.33	7.01229	704.2632	3.014638
4	434.2	0.3131	12.73	2784.65	7.1316	664.0288	2.34808
5	375.7	0.0813	5.43	2687.68	7.7085	395.1426	0.596007
6	331.7	0.0118	198.62	2674.3	5.4638	163.089	16.998005
7	318.9	0.0118	198.62	192.202	0.7038	12.9348	0.71364
8	318.9	0.011	211.00	1462.0	5.5612	13.0358	0.76404
9	320.1	0.0112	211.00	1462.0	5.5372	18.8888	1.10709
10	314.2	0.0113	19.15	197.39	0.6747	0.925	0.00492
11	320.2	0.0319	211.00	274.67	0.899	11.3636	0.666033
12	326.2	0.0843	12.73	301.15	0.9727	15.881	0.056157
13	430.9	0.3968	211.00	418.14	1.2832	40.342	2.364489
14	438.9	1.0020	265.00	580.94	1.6800	84.8956	6.249259
15	456.5	12.5859	265.00	596.55	1.6731	102.5618	7.549688
16	445.1	0.9700	42.71	615.58	1.7465	99.7186	1.18305
17	433.1	10.223	265.00	731.49	1.9796	146.1648	10.75935
18	436.1	1.9814	15.70	748.34	2.2127	93.551	0.407986
19	484.6	9.9280	265.00	877.52	2.4625	148.2906	10.91584
20	783.5	8.7280	265.00	2608.32	6.6669	626.1794	108.09376
Output air	318.15	0.1013	23.900	444.68	3.9468	726.871	4.82561

TABLE 4. Total exergy (MW) and temperature rate at different reference environment temperature (K)

	283	288	293	298	303
1	5.643	5.490	5.339	5.189	5.042
2	3.959	3.836	3.715	3.596	3.478
3	3.448	3.302	3.157	3.015	2.873
4	2.713	2.589	2.468	2.348	2.229
5	0.765	0.707	0.652	0.596	0.541
6	15.591	13.373	11.176	8.998	6.841
7	0.343	0.486	0.690	0.714	0.796
8	0.372	0.524	0.654	0.764	0.851
9	0.701	0.858	0.993	1.108	1.199
10	0.038	0.025	0.0142	0.005	0.002
11	1.232	1.022	0.833	0.666	0.521
12	0.094	0.080	0.067	0.056	0.046
13	3.268	2.945	2.646	2.364	2.106
14	7.822	7.271	6.747	6.249	5.779
15	9.115	8.566	8.045	7.549	7.082
16	1.448	1.355	1.267	1.183	1.1033
17	12.663	12.001	11.367	10.759	10.179
18	0.536	0.492	0.449	0.408	0.368
19	13.352	12.513	11.702	10.915	10.157
20	53.173	50.786	48.427	46.049	43.788
0utput air	4.458	4.583	4.705	4.826	4.943

TABLE 5. Reference environment temperature against efficiencies

Reference Environment Temperature(K)	**Q_{in}(MW)**	**W_{net}(MW)**	**$n_{th,pp}$(%)**	**n_R(%)**	**Exergy efficiency (%)**	**Exergy Destruction (%)**
283.15	35666.6	1242.29	22.5	47.2	29.4	60.4
288.15	5972.37	1381.46	23.1	50.1	30.3	60.7
293.15	6000.82	1383.47	23.4	50.4	30.6	61.2
298.15	6003.04	1394.72	23.9	53.7	30.9	61.5

TABLE 6. Condenser pressure against thermal efficiency, exergy efficiency and cycle efficiency.

Condenser Pressure (KPa)	**Thermal efficiency (%)**	**Exergy Efficiency (%)**	**Cycle efficiency (%)**	**Heat transfer to boiler(KJ/Kg)**	**Heat rejection of condenser (KJ/Kg)**
8.5	22.4	28.5	46.3	4070.8	2501.3
9.5	21.9	28.1	45.9	3946.3	2454.2
10.5	21.4	27.6	45.3	3910.4	2137.8
11.5	21.0	27.2	45.0	3875.8	2012.9
12.5	20.7	26.8	44.5	3808.8	1988.5
13.5	20.2	26.3	44.2	3784.7	1904.1

TABLE 7. Results of power output against efficiencies

Power Output (KW)	**Thermal Efficiency (%)**	**Exergy Efficiency (%)**
22	29.5	44.8
27	29.9	45.3
32	30.2	45.9
37	30.6	46.1
42	30.9	46.6
47	31.4	46.9

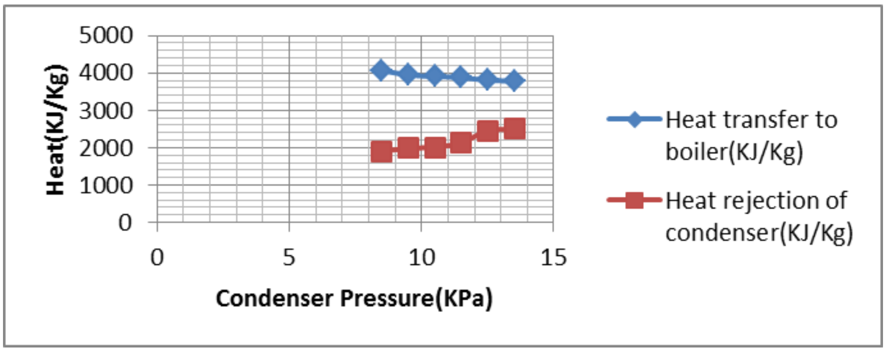

Figure 13. Graph of condenser pressure against heat transfer to boiler and heat rejection to condenser

Fig 13. shows increase in heat rejection of condenser and decrease in heat transfer to boiler as the condenser pressure increases. The heat loss leads to wastage in energy consumption, because energy is consume even when not in use. In order to eliminate heat losses in the boiler, the condenser pressure must be reduce, but must not let the water outlet from the condenser to freeze or cause water droplets at the turbine exhaust.

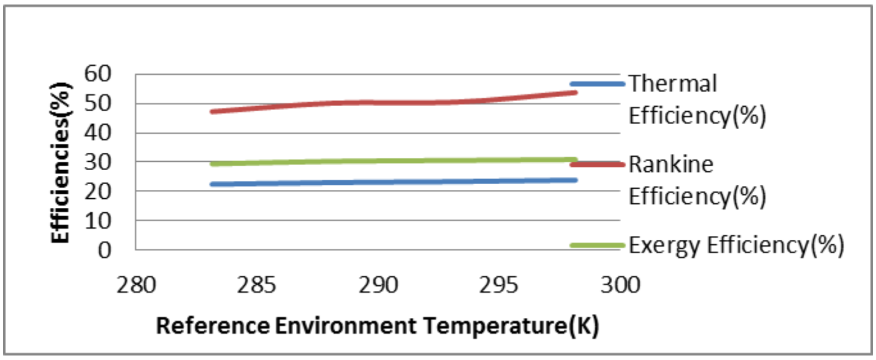

Figure 14. Graph of reference environment temperature vs. efficiencies

Fig 14. Shows a gradual increase in the efficiencies of the steam power plant as a result of increase in the reference environment temperature. It can be deduced that that the efficiency of the plant is affected by the surrounding temperature that is the inlet air temperature. To ensure maximum efficiency of the plant, the reference environment temperature must be increased.

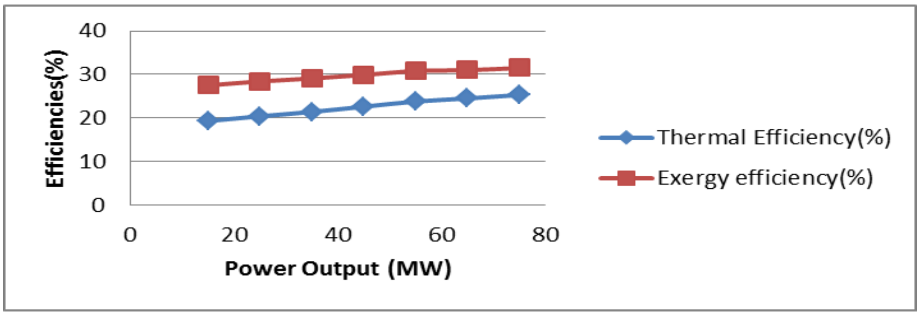

Figure 15. Graph of power output against efficiencies

Fig 15. Shows a steady increase in the thermal and exergy efficiencies of the plant as the power output increases. The operation of the power plant should not fall below 40% the designed capacity, because this will lead to an increase in the irreversibilities in the system and wastage in energy consumption. The plant should be operated in full load capacity (75MW) and not part load to enhance the efficiency of the plant.

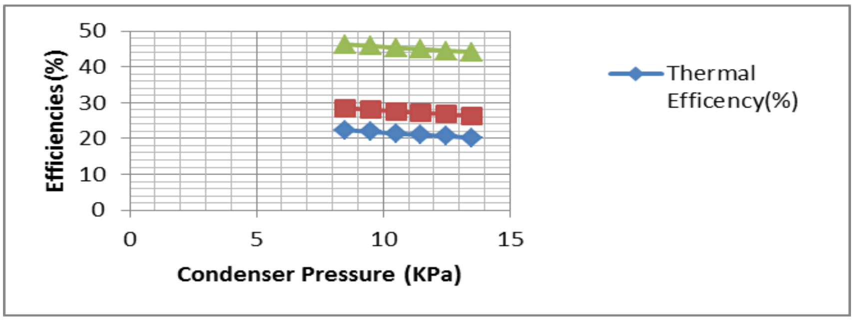

Figure 16. Graph of condenser pressure vs. efficiencies

Fig 4. shows a steady decrease in the efficiency of the plant as the condenser pressure increases. The condenser pressure must be reduced to ensure maximum utilization of the plant.

V. CONCLUSION

In this research, exergy analysis, energy balance, second law efficiency and varying the condenser pressure of the power plant has been presented. The results show that an increase in the condenser pressure is lead to gradual decrease in the efficiencies and heat transfer of the plant as well as increase the heat rejection of condenser. An increase in the reference environment temperature results in considerable increase in the efficiency of the power plant. It is observed that the plant should be operated in full capacity in order to increase its power output, thus reducing the rate of heat loss and energy consumption in the plant.

ACKNOWLEDGEMENT

The authors would like to thank the Management and staff of Landmark University, Omu-Aran, Kwara State and Afe Babalola University Ado-Ekiti, Ekiti State for providing the enabling environment to carry out this research work. Also, special thanks go to the management and staff of Sapele Power Plant, Delta State for their efforts in providing plant technical data.

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