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Investigating The Performance of A Steam Power Plant

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ABSTRACT: *The performance analysis of Shobra El-Khima power plant in Cairo, Egypt is presented based on energy and exergy analysis to determine the causes , the sites with high exergy destruction , losses and the possibilities of improving the plant performance.*

The performance of the plant was evaluated at different loads (Full, 75% and, 50 %). The calculated thermal efficiency based on the heat added to the steam was found to be 41.9 %, 41.7 %, 43.9% , while the exergetic efficiency of the power cycle was found to be 44.8%, 45.5% and 48.8% at max, 75% and, 50 % load respectively.

The condenser was found to have the largest energy losses where (54.3%, 55.1% and 56.3% at max, 75% and, 50 % load respectively) of the added energy to the steam is lost to the environment. The maximum exergy destruction was found to be in the turbine where the percentage of the exergy destruction was found to be (42%, 59% and 46.1% at max, 75% and, 50 % load respectively). The pump was found to have the minimum exergy destruction. It was also found that the exergy destruction in feed water heaters and in the condenser together represents the maximum exergy destruction in the plant (about 52%). This means that the irreversibilities in the heat transfer devices in the plant have a significant role on the exergy destruction. So, it is thought that the improvement in the power plant will be limited due to the heat transfer devices.

Keywords : *2nd law efficiency; Dead state; Steam power plant*

I. INTRODUCTION

Efficient utilization of the energy resources is of a major interest for system designers and scientists. Thermal analysis is the way to evaluate the performance of the energy systems (power generation thermal systems). Thermal analysis includes energy and exergy analysis. Energy analysis follows the energy through the system and measures the capability of the system in converting the input energy into the desired output (thermal efficiency). Where exergy analysis gives a measure how far the system performance is from the ideality [1-3]. In energy analysis, the term thermal efficiency which is known as a conversion efficiency is used for the evaluation. Thermal efficiency is used to evaluate the performance of the overall system. For that reason; first low analysis or the conversion efficiency is not enough to evaluate the system performance. Exergy analysis or second law analysis has been used in evaluating the energy system design to optimize and improve the system [4-7]. As a result of the exergy analysis, a term second law efficiency or exergetic efficiency is used. Exergetic efficiency can be determined for the overall system as well as for the individual components of the system. Energy and exergy analysis of energy systems are used to get a complete picture of the system performance. The performance of power plants as energy systems is evaluated using energy and exergy analysis. Exergy analysis of power plants is a helpful tool helps in determining the magnitudes, causes, and locations of losses as well as improving the efficiency of the overall system and its components. Efforts were done to analyze and locate the sites of maximum losses in thermal power systems. The efforts focused also on the ways of improving the performance. Aguilar et al [8] concerned with the loss factor to be used in energy audits for the turbine. Kwak et al [9] evaluated the performance of a combined cycle plant by analyzing each component in the plant as well as the whole cycle. They introduced an economic evaluation for the plant as well Khaliq et al, [10] used the exergetic analyses to evaluate the performance of a combined gas turbine-steam power. Sciubba et al [11] highlighted the importance of the concept of exergy and its usage in evaluating and optimization of the performance of power plants.

Aljundi [12], implemented the first law and second law analysis to evaluate the performance of a steam power plant. Studied the effect of the environmental condition on the plant performance. Yamini et al [13] found that increasing the number of feedwater heaters in the Rankine cycle improved the energy and exergy efficiency of the cycle. Sinha et al studied the effect of the ship external factors on the overall energy efficiency performance of a steam power plant. [14] Lalatendu et al studied the performance of a coal-fired boiler at the design and off-design conditions. Ozdil et al [15] found that the maximum exergy destruction occurs in the fluidized bed coal combustor of the steam power plant. Navid et al [16] studied the thermodynamic and exergo-economic performance of a combined steam-organic Rankine cycle.

The objective of this work is to make a detailed analysis of a steam power plant in Shobra El-Khima, Cairo, Egypt. The plant consists of four units. The energy and exegy analysis were implemented on one unit at different loads. Causes of irreversibilities in each component will be discussed. Sites of major energy loss and exergy destruction will be determined. The possibilities of improving the exergetic efficiency are examined.

II. PLANT DESCRIPTION

The power capacity of the plant is (1260) MW. The fuel used is Natural gas and heavy fuel oil. It has four steam turbines with a rated power 315 MW for each one at full load. Each unit has seven stages of regenerative feed water heating systems. The bleeding water to the regenerative heaters is taken from the high-pressure turbine, the intermediate turbine, and the low-pressure turbine. The locations, type and the number of feedwater heaters are shown in Figure 1. The steam generator produces superheated steam to the high-pressure turbine at (811) K and (16.64) MPa with a mass flow rate of 269.3 kg/s at full load. The exhaust steam at 3.75 MPa from the high-pressure turbine is reheated to 811 K and enters the intermediate pressure turbine at 3.378 MPa. The steam enters the low-pressure turbine at 0.982 MPa. The condenser pressure is 8.469 kPa. At 75% load, the steam enters the high-pressure turbine at a rate of 192.5 kg/s and exhausts from the turbine at a pressure of 2.72 MPa where it is reheated to 811 K and enters the intermediate pressure turbine at 2.45 MPa. The steam enters the low-pressure turbine at 0.718 MPa. The condenser pressure is 8.36 kPa. At 50% load, the steam enters the high-pressure turbine at a rate of 168.6 kg/s and exhausts from the turbine at a pressure of 2.37MPa where it is reheated to 811 K and enters the intermediate pressure turbine at 2.13 MPa. The steam enters the low-pressure turbine at 0.822MPa. The condenser pressure is 8.469 kPa. Tab. 1 shows the operating conditions of the plant.

III. THERMODYNAMIC ANALYSIS

Exergy and availability are identical terms and they can be defined as "measure of the maximum capacity of a system to perform useful work as it proceeds to a specified final state in equilibrium with its surroundings". Exergy is not conserved where it is destructed in the system due to the irreversibilities in the system. By exergy analysis, the magnitudes, sources, and locations of thermodynamic inefficiencies can be identified.

3.1 Energy Analysis

Neglecting the potential and kinetic energy effects of the flowing streams the first law calculations at steady state operation of the plant were done.
The conservation equation of energy can be expressed as follows:

$$\overset{o}{Q} - \overset{o}{W} = \sum \overset{o}{m}_e h_e - \sum \overset{o}{m}_i h_i \qquad (1)$$

The thermal efficiency of the plant:

$$\eta_{th} = \frac{\overset{o}{W}}{\overset{o}{Q}} \qquad (2)$$

3.2 Exergy Analysis

Exergy can be transferred in the form of heat, work, and mass flow.
The general equation of exergy balance for a steady state process;

$$\overset{o}{X}_{heat} - \overset{o}{W} = \sum \overset{o}{m}_e \Psi_e - \sum \overset{o}{m}_i \Psi_i + \overset{o}{I} \qquad (3)$$

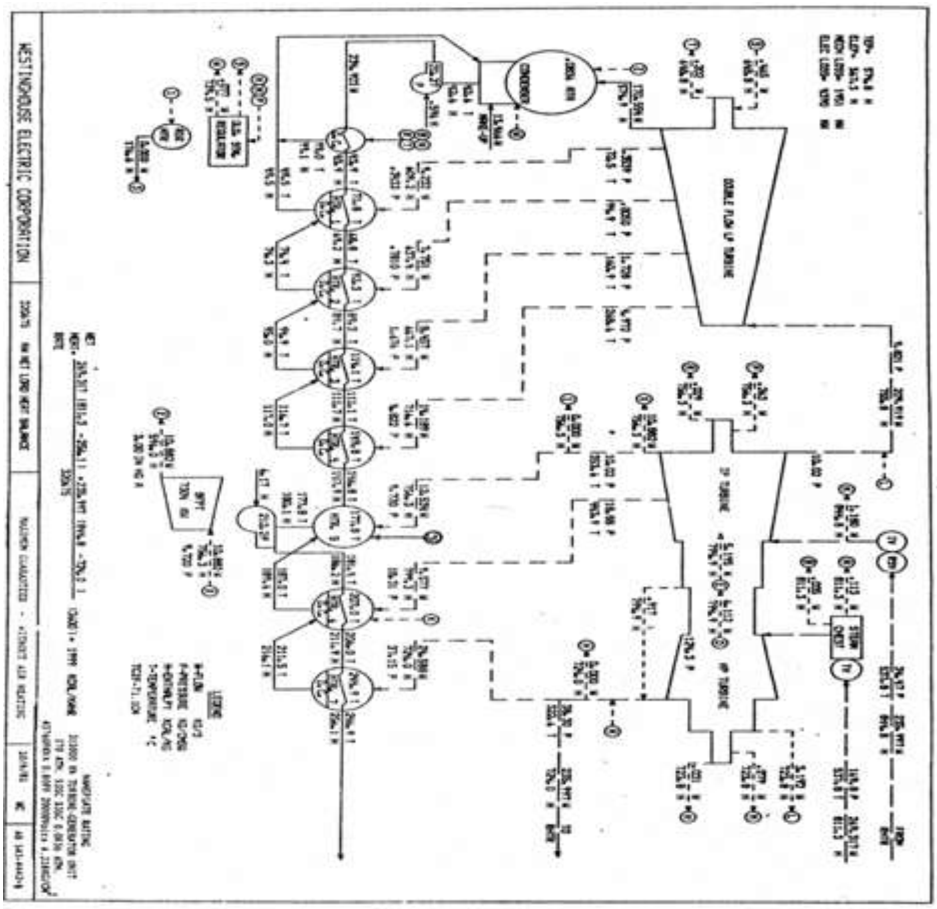

Figure 1: Cycle Scheme and operating conditions at 50 % load

Table 1 The operating operation conditions of the power plant

Operating Condition	**Value**		
	Max. Load	**75% Load**	**50% Load**
Feedwater inlet temperature to boiler	520.05 K	502.05 K	494.15 K
Steam Flow rate	969.48 ton/h	693 ton/h	607 ton/h
Steam temperature, HPT inlet	811K	811K	811K
Steam pressure, HPT inlet	166.4 bar	166.4 bar	166.4 bar
Power output	320 MW	236 MW	208 MW

The following equations represent the exergy of various forms of energy:

The net exergy transfer by heat ($\overset{o}{X}_{heat}$) at temperature T is given by

$$\overset{o}{X}_{heat} = \sum\left(1-\frac{T_o}{T}\right)\overset{o}{Q} \qquad (4)$$

And the specific exergy is given by

$$\Psi = h - h_o - T_o\,(s - s_o\,) \qquad (5)$$

Then the total exergy rate associated with a fluid stream becomes

$$\overset{o}{X} = \overset{o}{m}\Psi = \overset{o}{m}[h - h_o - T_o(s - s_o)] \qquad (6)$$

Table II shows the definition of the exergy destruction rate and the 2nd law efficiency for each component in Figure 1 in case of a steady state operation.

Table 2: Definition of the exergy destruction rate and the 2nd law

Components	Exergy Destruction Rate	2nd Law Efficiency (η_{II})
Pumps	$\overset{o}{I}_{pump} = \overset{o}{X}_{in} - \overset{o}{X}_{out} + \overset{o}{W}_{pump}$	$\eta_{II,pump} = 1 - \frac{\overset{o}{I}_{pump}}{\overset{o}{W}_{pump}}$
Heaters	$\overset{o}{I}_{heater} = \overset{o}{X}_{in} - \overset{o}{X}_{out}$	$\eta_{II,heater} = 1 - \frac{\overset{o}{I}_{heater}}{\overset{o}{X}_{in}}$
Turbine	$\overset{o}{I}_{turbine} = \overset{o}{X}_{in} - \overset{o}{X}_{out} - \overset{o}{W}_{el}$	$\eta_{II,turbine} = 1 - \frac{\overset{o}{I}_{turbine}}{\overset{o}{X}_{in} - \overset{o}{X}_{out}}$
Condenser	$\overset{o}{I}_{condenser} = \overset{o}{X}_{in} - \overset{o}{X}_{out}$	$\eta_{II,condenser} = \frac{\overset{o}{X}_{out}}{\overset{o}{X}_{in}}$
Cycle	$\overset{o}{I}_{cycle} = \sum_{allcomponent} \overset{o}{I}$	$\eta_{II,cycle} = \frac{\overset{o}{W}_{net,out}}{\overset{o}{X}_{HPTinlet}}$

IV. RESULTS AND DISCUSSIONS

The performance of the power plant was analyzed with the reference ambient temperature and pressure as 298.15 K and 101.3 kPa, respectively.KATT software[17] was used to indicate the thermodynamic properties of water at the indicated state points of Figure 1 Appendix A shows a summary of the properties of the state points at different loads. Figure 2 shows that the plant has highest thermal efficiency at 50% load (≅ 44%). This efficiency was based on steam inlet conditions at HPT. Figure 3 shows that the plant has highest exergetic efficiency at 50% load (ηII ≅ 48.8%), while the lowest exergetic efficiency was found to be at full load (η_{II} ≅ 44.8%).From figure 2 and 3 it can be concluded that the plant has the best performance at off-design condition (part load conditions). The figures show that, the thermal efficiency was found to be 41.9 %, 41.7 %, 43.9% at max load, 75% load and, 50 % load respectively, while the power cycle exergetic efficiency was found to be 44.8%, 45.5% and 48.8% at max load, 75% load and, 50 % load respectively.Figure 4 shows that the maximum exergy destruction was found to be in the turbine, and the maximum exergy destruction in the turbine occurs at 75% load while operating the plant at maximum load results in lowest exergy destruction. From this; it could be concluded that the irreversibilities in the turbine are dominant over all other irreversibilities in the cycle. The percentage of exergy destruction in the turbine is (42% - 46%) of losses in the plant. Also, this indicates that focus should be on the turbine to make significant improvements [18], [19]. Figure 4 shows also that the minimum exergy destruction was found to be at the pump.

The causes of the exergy destruction include friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, non-quasi equilibrium compression or expansion [4].The pumping process of the feed water to the boiler has fewer causes of exergy destruction where the heat transfer during the process is considered to be neglected; the operating fluid is a single phase (liquid). For these reasons it is thought that the pump has the minimum exergy destruction.On the other hand, the operation of steam expansion in the turbine has many causes of exergy destruction such as high friction between steam and turbine blades due to the high velocity of steam, the deviation of the steam expansion process from the quasi-equilibrium expansion and the heat transfer of steam at a temperature and the surrounding. For these reasons it is believed that the turbine has the maximum exergy destruction.

In the following the results of each sector of the plant will be analyzed:

4.1 Turbine:

Figure 5 shows that the high-pressure turbine stage has the maximum exergy destruction. This may be because of the irreversibilities arise from the interaction between the steam flow and the turbine blades; where at this stage the steam has the highest pressure, temperature, and velocity which cause high interaction between the steam flow and the turbine blades. The figure shows also that the exergy destruction in this stage increases with decreasing the load where it has its maximum value at 50% load.Figure 5 shows that the third stage of the low-pressure turbine has the lowest exergy destruction at all loads. Also, among the three loads of operation, the 75% load has the highest exergy destruction.

4.2 Condenser:

From the first law analysis, energy losses associated with the condenser are significant because they represent about (55%) of the energy input to the plant. The exergy analysis showed that only about (20% - 28 %) of the exergy was lost in the condenser.

A large amount of heat energy is removed from the plant via the condenser. Due to the low grade of this heat energy, it is thermodynamically insignificant and from an economic point of view; it is difficult to regenerate this energy [18], [20]. Figure 4 shows that the exergy destruction in the condenser decreases with decreasing plant load. The major factor of irreversibility in the condenser is the heat transfer through a finite temperature difference.

4.3 Feed water heaters:

Figure 6 shows that the maximum exergy destruction in the feed water heaters occurs at a full load of operation. Combining the exergy destruction in feed water heaters and the condenser (figure 7) it reveals that the exergy destruction in the condenser and heaters represent the maximum exergy destruction in the plant. This means that the irreversibilities due to the heat transfer processes in the heaters and the condenser have a major influence on the exergy destruction in the plant.

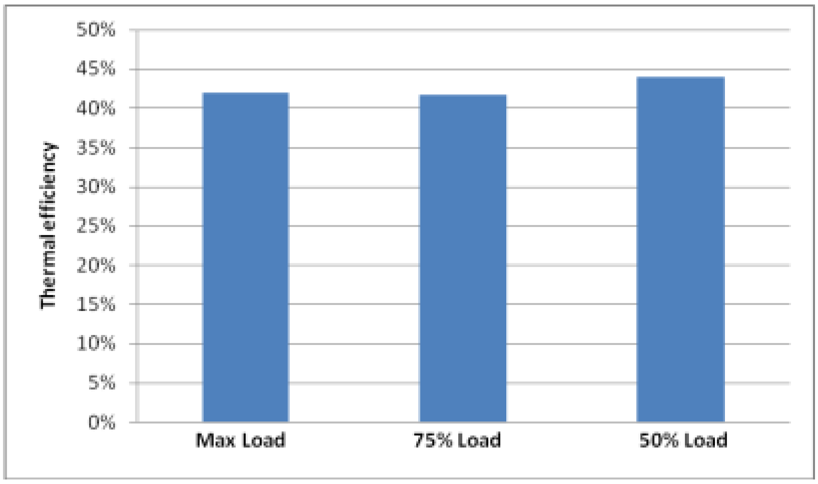

Figure 2: Thermal efficiency at different loads

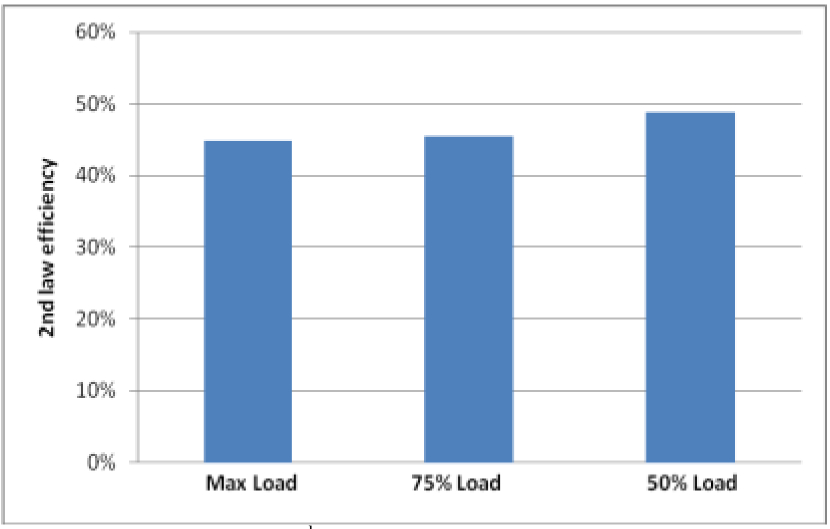

Figure 3: 2nd law efficiency at different loads

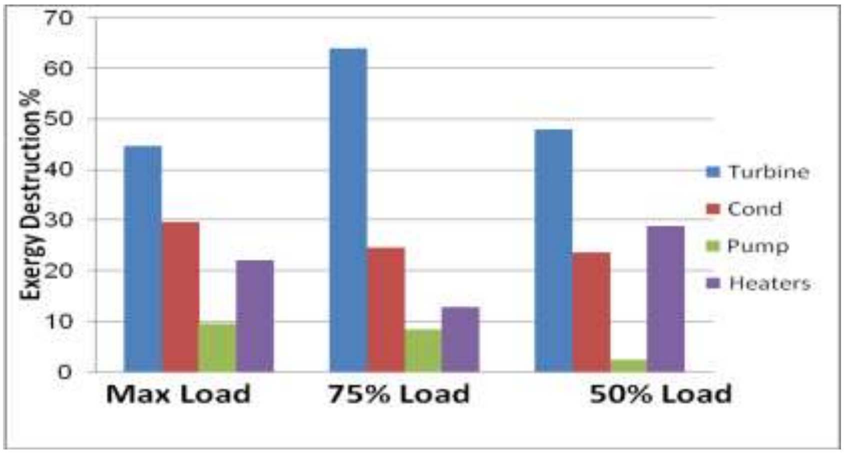

Figure 4: Percent exergy destruction at different loads

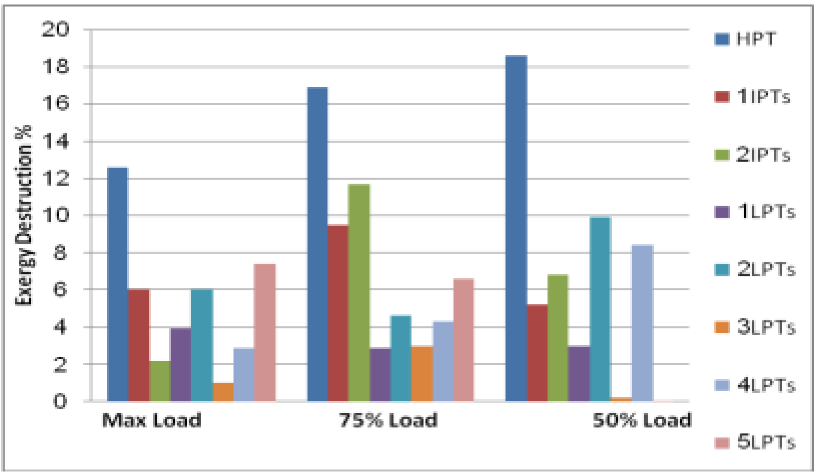

Figure 5: Percent exergy destruction in the turbine stages at different loads

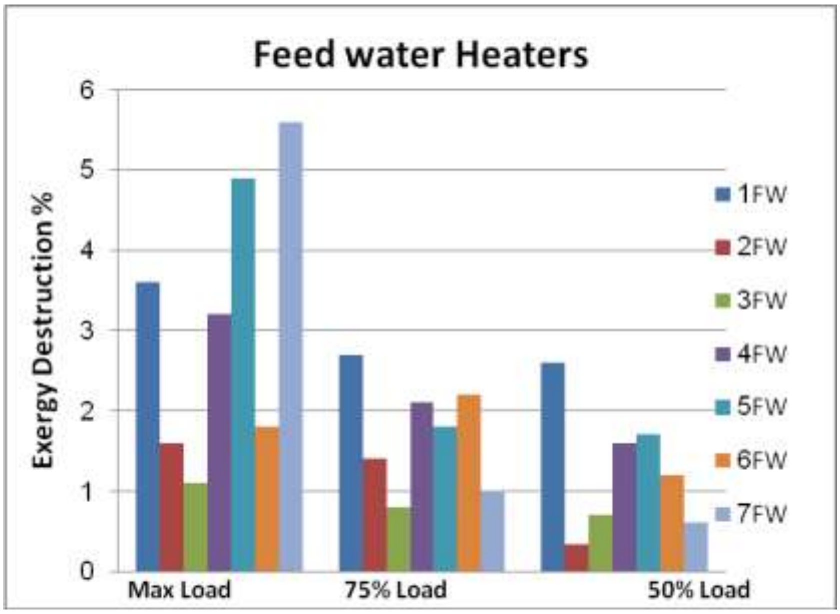

Figure 6: Percent exergy destruction in the feed water heaters at different loads

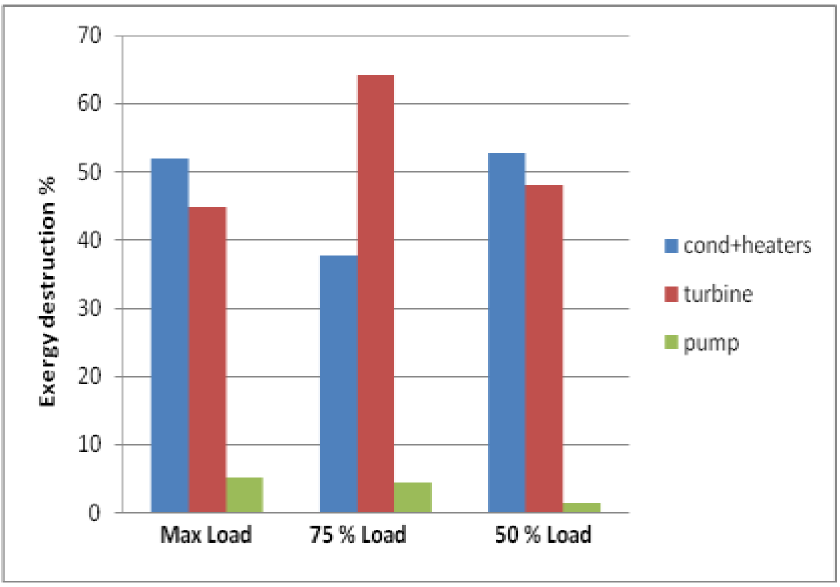

Figure 7: Comparison of % exergy destruction at different loads

V. CONCLUSION

In the present study, the first law and the second law of thermodynamics were used to evaluate the performance of a steam power plant. The main objective of this paper was to analyze the plant main components separately and to identify and quantify the sites having largest energy and exergy losses at different loads. The calculated thermal efficiency of the cycle based on specific heat input to the steam was 41.9%, 41.7% and 43.9% at 50%, 75%, and full load respectively. The calculated exergy efficiency of the power cycle is (45% - 49%), which is low. The best mode of operation of the plant was found to be at 50% load where; the plant has highest thermal and exergetic efficiencies at this mode of operation.

The maximum energy loss occurs in the condenser where 56.4%, 55.2% and 54.4% of the input energy was lost to the environment at 50%, 75%, and full load respectively. But this energy is thermodynamically insignificant due to its low-grade energy. The exergy destruction in feed water heaters and in the condenser together represents the maximum exergy destruction in the plant (about 52%). This means that the irreversibilities in the heat transfer devices in the plant have a significant role in the exergy destruction in the plant. But it is thought that the opportunities to improve the exergy destruction in these heat transfer devices are low due to the nature of the heat transfer process itself which should be done at finite temperature difference (physical constraints). The second law analysis of the plant showed that the turbine is the major source of losses where 46.1%, 59.6% and 42% of the fuel exergy input to the cycle was destroyed at 50%, 75%, and full load respectively. The turbine has the maximum exergy destruction while the pump has the minimum exergy destruction. In contrast to the case of heat transfer devices; it is thought that the opportunities to improve the irreversibilities in the turbine are high. In general, part of the irreversibility in the plant can not be avoided due to physical, technological, and economic constraints.

Nomenclature			
h	Specific enthalpy (J/kg)	s	Specific entropy (J/kg.K)
I	Exergy destruction rate (W)	T	Temperature (K)
m^o	Mass flow rate (kg/s)	$\overset{o}{W}$	Work done rate or power done by the system (W)
P	Pressure (Pa)	$\overset{o}{X}$	Total exergy rate (W)
$\overset{o}{Q}$	Heat transfer rate to the system kJ/s		
Greek Symbols		**Abbreviations**	
η_{th}	First law efficiency	HPT	High-pressure turbine
η_{II}	2nd law efficiency	IPT	Intermediate pressure turbine
ψ	Specific exergy (J/kg)	LPT	Low-pressure turbine
Subscripts			
e	exit		
i	inlet		
o	Dead state condition		

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APPENDIX

The thermodynamic properties of water at indicated state points shown in Fig. 1Full Load:

node	*T* (K)	*P* (MPa)	$\overset{o}{m}$ (kg/s)	*h* (kJ/kg)	*s* (kJ/kg K)	Ψ(kJ/kg)	$\overset{o}{X}$ (MW)
1	811	16.64	269.317	3391	6.406	1484.7	399.9
2	596	3.75	236.317	3026	6.5	1093.5	256.8
3	811	3.378	234.82	3531	7.257	1264.2	296.857
4	717	1.35	226.42	3346	7.304	1174	274.716
5	627	0.982	209.919	3166	7.322	988.6	223.847
6	542	0.4873	195.73	3000	7.356	812.5	169.814
7	434	0.169	187.3	2793	7.413	588.5	115.169
8	368	0.0833	179.55	2668	7.423	460.5	82.667
9	346	0.0347	170.32	2631	7.719	423.5	72
10	315.7	0.0085	170.554	2403	7.8	80.2	13.636

75 % Load:

node	T (K)	P (MPa)	$\overset{o}{m}$ (kg/s)	h (kJ/kg)	s (kJ/kg K)	Ψ(kJ/kg)	$\overset{o}{X}$ (MW)
1	811	16.64	192.5	3391	6.406	1484.7	285.8
2	568	2.72	169.2	2990	6.575	1035	199.284
3	811	2.45	169.2	3540	7.46	1330.5	225.1
4	718	1.348	163.238	3175	7.48	1136	192
5	628	0.718	152.3	3175	7.48	950.6	154.94
6	544	0.357	142.749	3009	7.515	774	118.246
7	436	0.124	136.918	2802	7.574	549.5	78.54
8	370	0.0571	131.737	2675	7.615	410.3	56.173
9	339	0.02573	127	2619	7.822	292.6	38.540
10	369	0.0085	127.2	2429	7.9	153.99	19.402

50 % Load:

node	T (K)	P (MPa)	$\overset{o}{m}$ (kg/s)	h (kJ/kg)	s (kJ/kg K)	Ψ(kJ/kg)	$\overset{o}{X}$ (MW)
1	811	16.64	168.627	3391	6.406	1486.6	250.641
2	560	2.37	147.729	2977.8	6.612	1012	170.6
3	811	2.13	146.99	3543.4	7.464	1323.7	194.574
4	718	1.165	142.185	3372	7.531	1132	166.445
5	631	0.822	132.889	3171	7.54	928	123.354
6	545	0.3116	124.837	3012	7.582	757	99.945
7	457	0.1085	119.96	2844	7.729	545	68.06
8	360	0.0498	115.6	2657	7.732	357	42.88
9	336	0.0226	112.08	2614	7.867	274.2	31.7
10	315.7	0.0085	112.023	2442	7.92	130.2	14.597