Mehran University Research Journal of Engineering & Technology
Vol. 37, No. 4, 611-624 October 2018
p-ISSN: 0254-7821, e-ISSN: 2413-7219
DOI: 10.22581/muet1982.1804.13

Energy and Exergy Analysis of a Coal Fired Power Plant

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RECEIVED ON 20.07.2017 ACCEPTED ON 12.02.2018

ABSTRACT

In this paper, energy and exergy analysis has been conducted on a subcritical coal fired power plant of Wisconsin Power and Light Company, USA to investigate the steam cycle energy and exergy efficiency. The cycle is analyzed by developing a mathematical model using its operating and design parameters. The analysis is performed using EES (Engineering Equation Solver). The energy analysis shows that major share of energy loss occurs in condenser i.e. 72% of total cycle energy loss, whereas, exergy analysis shows that 83.09% total exergy destruction of cycle occurs in boiler.Furthermore, the simulation results are compared with actual with an absolute error of 3.1%. Additionally, the parametric study is performed to examine the effects of various operating parameters such as main steam pressure and temperature, condenser pressure, terminal and drain cooler temperature difference on net power output, energy andexergy efficiency of cycle. The parametric study shows that the plant has maximum energy and exergy efficiencies at steam pressure of 2500psi, condenser pressure of 1.0psi and main steam temperature of 1100^{o}F. Furthermore, these parameters do not seem to change energy and exergy efficiencies significantly.

Key Word: Energy Analysis, Exergy Analysis and Wisconsin Power and Light Company.

1. INTRODUCTION

The energy demand is escalating throughout the globe due to rapid development in technology and evaluation in human lifestyle [1-2]. Most of the energy demand is fulfilled by harnessing fossil fuels. It has worsened the ecosystem with emission of environmental pollutants. In order to subside the negative impacts of the fossil fuels many researchers diverted towards renewable energy resources [3]. But according to the current energy scenario, about 80% of the world's energy demand is fulfilled by utilizing non-renewable energy resources and it will be continue in coming decades [4]. Among different fossil fuels, coal is one of the most abundant source found on the earth and can be used to generate the low-cost electricity worldwide. Its consumption is increasing at an average rate of 0.6%/year and it willbreak the line of 6000 million tons per year by 2030 [5-6]. The electricity generation from coal is about 41% [7]. The share of electricity generated from coal in South Africa, Poland, China, USA, Germany, India and Pakistan is about 93, 92, 79, 49, 46, 66 and 0.15% of total power generation from

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coal, respectively. The major consequences of coal consumption are environmental pollutants. The environmental pollutants resulting from coal consumption can be decreased by increasing the efficiency of coal fired power plant. The efficient conversion of coal is essential for sustainable development because coal contributes a major share of electricity generation in the globe [7-8].

Thermal power plant based on different thermodynamics cycles are usually analyzed using first (energy) and second law (exergy) of thermodynamics. In energy and exergy analysis, theoverall performance of the power plant is determined by using energy loss and exergy destruction occurs in its different components/processes. Conventionally, these cycles are analyzed using energy analysis based on the first law of thermodynamics. In which, the enthalpy values are used to calculate the loss of efficiency during a process due to the loss of energy. Recently, exergy analysis is performed to analyze the performance of these cycles because it is an appropriate method for understanding the processes of these cycles. The optimal usage of existing power plants can be developed using exergy analysis [9]. Exergy analysis provides a platform to investigate the performance of an individual process considering the difference between its energy losses and internal irreversibility [10-11].

Kotas [12] has performed exergy analysis using Grossman diagrams. It is used to determine the exergy flow in system considering its own exergy. In this analysis, the economic optimizations of thermodynamic system are performed using life cycle cost analysis and its associated economics issues are addressed.Exergy analysis is an appropriate method to measure the performance of energy conversion system. It also identifies the maximum loss in the cycle and efficiency of its different components and how to reduce these losses. Wu et. al. [13] has evaluated the performance of coal fired power plant using two different exergy analysis methods. One method measures performance of individual component while other considers the optimization process using creative flow design. Rosen and Dincer [14] has performed exergy analysis for optimization of cycle. Exergy analysis plays the most important role to decide the optimal performance of cycle with available parameters.

Amir [15] determined the effect of excess air on energy and exergy efficiency of a boiler of steam power plant using energy and exergy analysis. His analysis revealed that energy and exergy efficiencies are respectively improved to 0.29 and 0.37% by reducing the fraction excess air from 0.4-0.15%. The decrement in the temperature of flue gases also increases above efficiencies to 0.84 and 2.3%, respectively. Ahmadi at. el. [16] calculated that preheating of surrounding air and increasing air to fuel ratio to the boiler decreases its exergy loss.

In the past decade Regulagadda et. al. [17], Memon et. al. [18], Rajpar et. al. [19] and Erdem et. al. [20] all have performed the energetic and exergetic analysis of different regenerative Rankine cycle power plants. According to their energetic results it was found that the maximum energy loss occurred in condenser followed by the boiler and then turbine. While observing the exergetic results, the maximum share of the exergy destruction (or exergy loss) was in the boiler followed by the turbine and the condenser. Furthermore, parametric study is also done to see the effects of various operating parameters on the cycle performance.

Cihan et. al. [21], Ameri et. al. [22] and Memon et. al. [23] have used the same method for evaluating the energy and exergy analysis for the CCPP's (Combined Cycle Power Plants). They all have the same kind of results and concluded that the maximum share of exergy destruction was in the combustion chamber followed by the HRSG (Heat Recovery Steam Generator) and then in the gas turbine contributing about 85% of the total exergy loss. So, these system components must be optimized to get the maximum potential.

Woudstra et. al. [24] has performed exergy analysis to evaluate alternative designs for combined cycle power plants. The evaluation results revealed that the exergy efficiency of gas power cycle varies with the heat transfer temperature. The increment in number of pressure levels in HRSG reduces temperature difference between combustion gases and steam. Thus, the number of pressure levels in HRSG reduces exergy destruction and the temperature of exhaust flue gases at the stack. The exergy flow diagram shows that the maximum exergy destruction occurs in the combustion process. In future the use of high temperature fuel in combined cycle power plants conceivably improves the performance of CCPP's. Rosen et. al. [25] has observed the effects of varying dead-state properties on the energy and exergy analysis of thermal systems. Aljundi et. al. [26] carried out similar analysis varying the reference environment and also performed the energy and exergy analysis of a steam power plant in Jordan. Dincer [27] has utilized the exergy concept to find out efficient energy and environmental policy making activities. He has also concluded that exergy analysis is useful for economics, sustainability development and a tool to distinguish the quality of the energy resources for the power plant. Roosen et. al. [28] have used a different method of Pareto for the optimization of CCPP's. The annualization factor is considered to optimize CCPP using both operation and investment costs.

In this paper, the energetic and exergetic analysis of Wisconsin Power and Light Company Columbia Energy Centre's 512MW subcritical coal-fired unit is carried out.The effect of various operating parameters i.e. inlet pressure, main steam temperature and condenser pressure on the cycle performance is determined.

2. PLANT DESCRIPTION

The plant under study, shown in Fig. 1, is owned by Madian Gas and Electric, Wisconsin Public Service Corporation and Wisconsin Power and Light Company. It is the 1st unit of the total plant with an installed capacity of 512MW and is commissioned in 1975 at a cost of $150 million. The unit 2 followed in 1978 with a capacity of 511MW, making the total installed capacity of the plant as 1023MW. The low bituminous coal, taken from the PRB (Powder River Basin), is used as the fuel having LHV of 9450 Btu/lbm [29].

The plant under study is based on the regenerative reheat Rankine cycle. It is a single reheating system consists of one reheater (RHTR). The main steam is respectively expanded through high, intermediate and low pressure turbines to generate the shaft work for electricity. After that, the steam is condensed in the condenser and then pumped forward by the condensate pump. During expanding through the turbines, the steam is bladed at different stages to preheat the feed water in the FWH (Feed Water Heaters). It consists of seven FWHs including four low pressures, one intermediate pressure, one high pressure and one open type FWH (Dearator). The first low pressure FWH is a drain forwarded type while the remaining all are cascade backward type FWHs as shown in Fig. 1. A mixture of three bladed steams is used in the SSR (Steam Seal Regulator) which regulates the pressure and distribute it again in three parts. One part is directly sent to the condenser, the other part is passed through the steam packing exhaust to the condenser and the last one is mixed with the bladed steam from the low pressure turbine and is then used in the lowest pressure feed water heater.

3. MODELING AND ASSUMPTIONS

To evaluate the performance of steam cycle power plant in terms of energy and exergy analysis the state point's parameters should be varied. Therefore, the performance of selected power plant is investigated by varying state point's paramters at different location. These values are used to assess the performance of each component of the cycle and development of an appropriate mathematical relationship between them. These relationships are based on the fundamental laws such as mass conservation, energy conservation and exergy balances. The developed mathematical model is based on after mentioned laws and simulated using EES to determine the energy and exergy efficiencies and effects of different thermodynamic parameters on the performance of the cycle. The EES software is very suitable to solve heat transfer, thermodynamics and fluid flow model equations which were essential for the study we conducted. In addition, the software is also suitable for data plotting, optimization, and regression and uncertainty analyses. The assumptions made for the analysis and values for operation of the plant are given in Table 1 [31-34].

3.1 Modeling Equations

In order to perform energy and exergy analysis of thermal systems, the mass flow rate and energy transfer rate are determined using two basic laws, i.e. conservation of mass and conservation of energy. One can investigates the exergy destruction rate and exergy efficiency of thermal systems using the first and second law of thermodynamics as following [31-35]:

Continuity Equation

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

Energy Equation

$$\dot{Q}_{in} + \dot{W}_{in} + \Sigma_{in} \dot{m}h = \dot{Q}_e + \dot{W}_e + \Sigma_e \dot{m}h \quad (2)$$

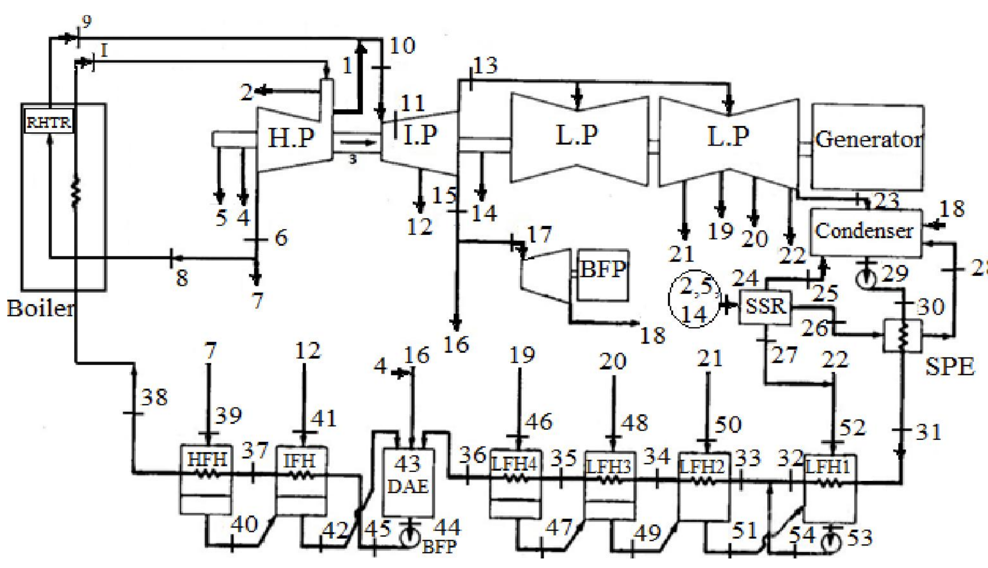

FIG. 1. SCHEMATIC DIAGRAM OF POWER PLANT [30]

Exergy Equation

$$\Sigma\left(1-\frac{T_o}{T_{in}}\right)\dot{Q}_{in}+\dot{W}_{in}+\Sigma_{in}\dot{m}=\left(h-T_o s_o\right)=\Sigma\left(1-\frac{T_o}{T_e}\right)\dot{Q}_e+\dot{W}_e+\Sigma_e\dot{m}=\left(h-T_o s_o\right) \quad (3)$$

where subscripts in and e refer to streams entering and leaving the control volume.

Also, the fuel energy required in producing the steam in boiler is given as:

$$\dot{E}x_{in.f}=\dot{m}_f.LHV \quad (4)$$

And the exergy inflow associated with the fuel flow is given by

$$\dot{E}x_{in.f}=\dot{m}_f.ex_f \quad (5)$$

In Equation (6), the specific exergy of fuel (coal) is approximated from the following expression [36]:

$$ex_f=\left(\left(LHV\right)\left(1.0438+0.0013.\frac{\alpha_H}{\alpha_C}+0.1083.\frac{\alpha_O}{\alpha_C}+0.0459.\frac{\alpha_H}{\alpha_C}\right)\right)+0.0013\alpha_S \quad (6)$$

The energy and exergy balance and their efficiency relations are summarized in Table 2 for each component of the system.

Now, the energy and exergy efficiency of the whole plant is given as:

TABLE 1. ASSUMPTIONS AND OPERATING VALUES

No.	Condition	Values
1.	Date state condition	14.7psia and 77°F
2.	Energy efficiency of the boiler	85%
3.	Isentropic efficiency of high pressure steam turbine	85%
4.	Isentropic efficiency of medium pressure steam turbine	82%
5.	Isentropic efficiency of low pressure steam turbine	80%
6.	Isentropic efficiency of pumps	85%
7.	Lower heating value of coal	9450 Btu/lbm
Note: Changes in kinetic and potential energy and exergy of the working fluid is neglected		

TABLE 2. MODEL EQUATIONS FOR DIFFERENT DEVICES OF THE PLANT

Components	Energy Balance	Exergy Balance	Energy Efficiency	Exergy Efficiency
Boiler	$\Sigma\,\dot{m}_{in}.h_{in}+\dot{Q}_{boiler}=\Sigma\,\dot{m}_e.h_e$	$\dot{E}x_f+\Sigma\dot{E}x_{in,B}=\Sigma\dot{E}x_{e,B}+\dot{E}x_{D,B}$	$\eta_{e,B}=\left(\frac{\dot{m}_e.h_e-\dot{m}_{in}.h_{in}}{\dot{Q}_{boiler}}\right)$	$\eta_{ex,B}=\left(\frac{\dot{E}x_{e,B}-\dot{E}x_{in,B}}{\dot{E}x_f}\right)$
Steam Turbine	$\Sigma\,\dot{m}_{in,T}.h_{in,T}=\Sigma\,\dot{m}_{e,T}.h_{e,T}+\dot{W}_{st}$	$\Sigma\,\dot{E}x_{in,T}=\Sigma\,\dot{E}x_{e,T}+\dot{W}_{st}+\dot{E}x_{D,T}$	$\eta_{e,T}=\left(\frac{W_{st}}{\dot{m}_{e,T}.h_{in,T}-\dot{m}_{e,T}.h_{e,T}}\right)$	$\eta_{ex,T}=\left(\frac{\dot{W}_{st}}{\dot{E}x_{in,T}-\dot{E}x_{e,T}}\right)$
Pump	$\dot{m}_{in,p}.h_{in,p}+\dot{W}_p\dot{m}_{e,p}.h_{e,p}$	$\dot{E}x_{in,p}+\dot{W}_p=\dot{E}x_{e,p}+\dot{E}x_{D,p}$	$\eta_{e,p}=\left(\frac{\dot{m}_{in,p}.h_{in,p}-\dot{m}_{e,p}.h_{e,p}}{\dot{W}_p}\right)$	$\eta_{ex,p}=\left(\frac{\dot{E}x_{in,p}-\dot{E}x_{e,p}}{\dot{W}_p}\right)$
Heater	$\Sigma\,\dot{m}_{in,H}.h_{in,H}=\Sigma\,\dot{m}_e.h_{e,H}.h_{e,H}$	$\Sigma\,\dot{E}x_{in,H}=\Sigma\,\dot{E}x_{e,H}+\dot{E}x_{D,H}$	$\eta_{e,H}=\frac{\dot{m}_{e,H}.h_{e,H}}{\dot{m}_{in,H}.h_{in,H}}$	$\eta_{ex,H}=\frac{\dot{E}x_{e,H}}{\dot{E}x_{in,H}}$
Condenser	$\Sigma\dot{m}_{in,cond}h_{in,cond}=\Sigma\dot{m}_{e,cond}h_{e,cond}+\dot{Q}_{rejected}$	$\Sigma\dot{E}x_{in,cond}=\Sigma\dot{E}x_{e,cond}+\dot{E}x_{D,cond}$	$\eta_{e,cond}=\frac{\dot{m}_{e,cond}.h_{e,cond}}{\dot{m}_{in,cond}.h_{in,cond}}$	$\eta_{ex,cond}=\frac{\dot{E}x_{e,cond}}{\dot{E}x_{in,cond}}$

$$\eta_{en} = \frac{\dot{W}_T}{\dot{E}n_{in.f}} \tag{7}$$

$$\eta_{ex} = \frac{\dot{W}_T}{\dot{E}x_{in.f}} \tag{8}$$

4. RESULTS AND DISCUSSION

The results and discussion section includes the energy and exergy efficiency analysis of each component of the power plant using various parameters given in Table 1 and 3. Also, the effect of operating parameters on the net power output is determined. Additionally, effectiveness of feed water heaters is examined under variation in the TTD (Terminal Temperature Difference) and DC (Drain Cooler) temperature.

4.1 Energy and Exergy Performance Analysis

In the previous section, various thermodynamic relations are developed. These equations are used to find the power consumed, power produced and power lost in different plant components. Initially the mathematical model is validated comparing the simulated values with the actual results as shown in Fig. 2. The net power output and energy efficiency of the selected power plant is obtained from relevant article [29-30] are considered as an actual results, whereas simulation results are obtained by performing simulation of developed mathematical model onEES using preliminary data regarding design and operating parameters given in Table 1 and 3. The results show that the difference in the actual and model values for the total power output and thermal efficiency was only 3.07 and 1.3% respectively.

However, in Fig. 3 energy loss and exergy destruction of the different plant components including, boiler, turbine, compressor, low and high pressure feed water heaters, is compared. Results show the maximum energy loss occurred in the condenser with a total share of 72% followed by the boiler with around 20% and then the turbine with 5.8%. Moreover, from the exergy results it was found that the maximum share of exergy destruction (i.e. 83%) is in the boiler followed by the turbine and condenser with a share of 11 and 4.5% respectively. The energy loss in condenser can be used for space heating and exergy destruction in boiler can be decreased by employing Organic Rankine Cycle to utilize energy waste in flue gases. On the other hand, Fig. 3 shows that the exergy destruction in feed water

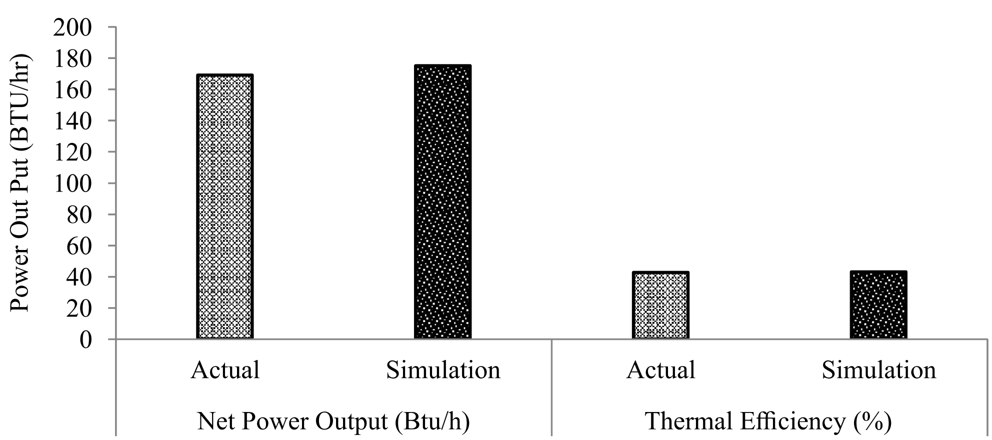

FIG. 2. COMPARISON BETWEEN SIMULATED AND ACTUAL RESULTS OF TOTAL POWER OUTPUT AND THE THERMAL EFFICIENCY

heaters was only 1 or less than 1 percentage. It is noted that exergy destruction is higher in the components involved in heat transfer with larger temperature differences such as the boiler.

4.2 Parametric Analysis

4.2.1 Effect of Condenser Pressure

Fig. 4 shows the plant performance with the variation in the condenser pressure while the boiler pressure and main steam temperature remained constant. The increment in the condenser pressure increases the enthalpy of steam that in turn reduces the expansion of steam over low pressure turbine. The reduced expansion of steam over low pressure turbine results in reduced net power output. The energy efficiency is also decreased because net power output decreases at constant heat supply to the boiler while heat rejected in the condenser is high. The exergy efficiency decreases because less useful work is produced from low pressure turbine. The exergy destruction in low pressure turbine increases than that in the condenser. However, higher condenser pressure does not affect power output of intermediate pressure turbines because steam at the exit of intermediate pressure turbine is not condensed. These changes in the power and efficiencies decrease gradually as condenser pressure is allowed to increase after 1.5psi as shown in **Fig. 4.**

4.2.2 Effect of Main Steam Temperature

The effect of steam temperature coming out of the boiler is depicted in Fig. 5 at given main steam pressure and condenser pressure i.e. 2400 and 0.5psig. It is seen in Fig. 5, the values of the main steam temperature are directly proportional to the performance of the plant. The steam at higher temperature requires higher expansion that results in increased performance and vice versa. The effect on the plant efficiencies is also same, although impact on energy efficiency is higher than exergy efficiency. The higher temperature of steam at the exit of the boiler increases enthalpy of the steam. The heat input to the boiler and net power output of the turbine increases due to higher enthalpy. The changes in the exergy efficiency with higher steam temperature are low as compare to energy efficiency because an increase in the exergy destruction in turbine is higher than the exergy destruction in the boiler.

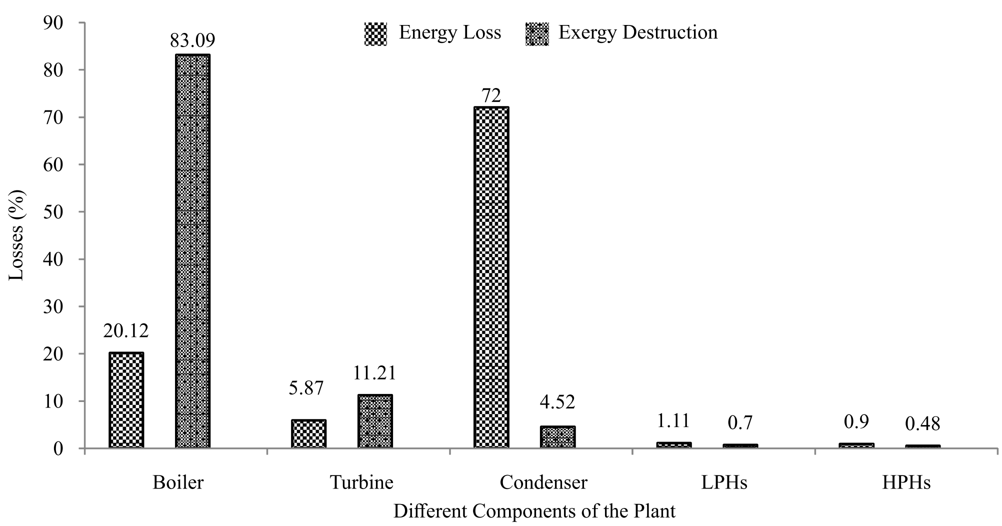

FIG. 3. ENERGY LOSS AND EXERGY DESTRUCTION IN DIFFERENT PLANT COMPONENTS

TABLE 3. STATE POINT VALUES AT DIFFERENT LOCATIONS

State Point	ṁ (klbm/h)	P (psia)	T (°F)	h (kBtu/lbm)	s (Btu/lbm.F)	x kBtu/lbm)
Inlet (I)a	3,413.62	2,415.00	1,000.00	1.46	1.53	1.34
1a	3.97	2,415.00	1,000.00	1.46	1.53	1.34
2a	0.90	2,415.00	1,000.00	1.46	1.53	1.34
3a	32.21	1,904.00	930.30	1.43	1.54	1.31
4a	7.60	556.70	624.50	1.31	1.56	1.19
5a	6.59	556.70	624.50	1.31	1.56	1.19
6a	3,362.35	556.70	624.50	1.31	1.56	1.19
7a	305.01	556.70	624.50	1.31	1.56	1.19
8a	3,057.34	556.70	624.50	1.31	1.56	1.19
9a	3,057.34	501.00	1,000.00	1.52	1.74	1.39
10a	3,061.31	503.70	1,000.00	1.52	1.74	1.39
11a	3,093.52	503.70	998.30	1.52	1.74	1.38
12a	91.23	258.70	832.50	1.44	1.75	1.30
13a	2,698.00	165.90	758.80	1.41	1.77	1.27
14a	3.78	165.90	758.80	1.41	1.77	1.27
15a	300.81	157.60	665.50	1.36	1.74	1.22
16a	194.76	157.60	665.50	1.36	1.74	1.22
17a	106.05	157.60	665.50	1.36	1.74	1.22
18b	106.05	0.49	79.01	1.07	1.99	0.92
19a	159.10	62.90	552.70	1.31	1.79	1.17
20a	83.48	22.80	308.00	1.20	1.77	1.06
21a	93.61	12.16	220.90	1.16	1.79	1.02
22a	170.54	5.76	168.30	1.11	1.79	0.97
23b	2,191.43	0.49	79.01	1.04	1.94	0.89
24a	11.27	0.49	635.40	1.35	2.37	1.17
25b	2.40	0.49	635.40	1.35	2.37	1.17
26a	2.80	0.49	635.40	1.35	2.37	1.17
27a	6.07	0.49	635.40	1.35	2.37	1.17
28b	2.80	0.49	79.01	1.07	1.99	0.91
29c	2,302.69	0.49	79.01	0.05	0.09	0.04
30d	2,302.69	157.60	80.15	0.05	0.09	0.04
31d	2,302.69	157.60	80.50	0.05	0.09	0.04
32d	2,302.69	157.60	161.10	0.13	0.23	0.11
33d	2,302.69	157.60	162.40	0.13	0.24	0.11
34d	2,302.69	157.60	195.10	0.16	0.29	0.14
35d	2,302.69	157.60	227.60	0.20	0.34	0.17
36d	2,302.69	157.60	287.50	0.26	0.42	0.22
37d	3,413.62	2,415.00	399.60	0.38	0.56	0.33
38d	3,413.62	2,415.00	475.90	0.46	0.65	0.41
39a	305.01	528.90	472.90	1.31	1.56	1.19
40c	305.01	528.90	409.60	0.39	0.58	0.34
41a	91.23	245.80	399.60	1.44	1.75	0.30
42c	396.34	245.80	379.70	0.35	0.54	0.31
43d	3,413.62	157.60	362.40	0.33	0.52	0.29
44d	3,413.62	157.60	362.40	0.33	0.52	0.29
45d	3,413.62	2,415.00	369.70	0.35	0.52	0.30
46a	159.10	59.80	292.50	1.31	1.79	1.17
47c	159.10	59.80	237.30	0.21	0.35	0.18
48a	83.48	21.79	232.20	1.20	1.77	1.06
49c	122.59	21.79	205.10	0.17	0.30	0.15
50a	93.61	11.56	200.10	1.16	1.79	1.02
51c	336.20	11.56	200.10	0.17	0.29	0.14
52a	176.61	5.48	166.10	1.12	1.81	0.97
53c	512.80	5.48	166.10	0.13	0.24	0.11
54d	512.80	157.60	168.30	0.14	0.24	0.12
Note: a= Superheated Steam, b= Saturated Steam, c= Saturated Water and d= Compressed Water						

4.2.3 Effect of Main Steam Pressure

This effect of boiler pressure on the plant performance at a given steam temperature (1000°F) is drawn in Fig. 6. It shows that the net power output and the efficiencies increase with an increment in the pressure. By increasing the pressure, the steam has to expand more through turbines at constant steam temperature and condenser pressure. The enthalpy of steam at the exit of the boiler increases with steam pressure at constant steam temperature. Thus, it increases the heat supplied to the boiler and power output from the turbine. The impact of pressure variation seems higher on exergy efficiency as compared energy efficiency. The energy efficiency increases gradually because the difference in power output and heat input decreases with mainsteam pressure. The exergy efficiency increases significantly because an increment in the steam pressure decreases the exergy destruction in the boiler and turbine while increasing useful work output from the turbine. However, to sustain high steam pressure the material cost of the components of the plant may increase which may increase the capital cost of the plant.

4.2.4 Effect of Terminal Temperature Difference

Figs. 7-9 show the effect of TTD on the effectiveness of high, intermediate and low pressure feed water heaters. It can be examined from the graphs.

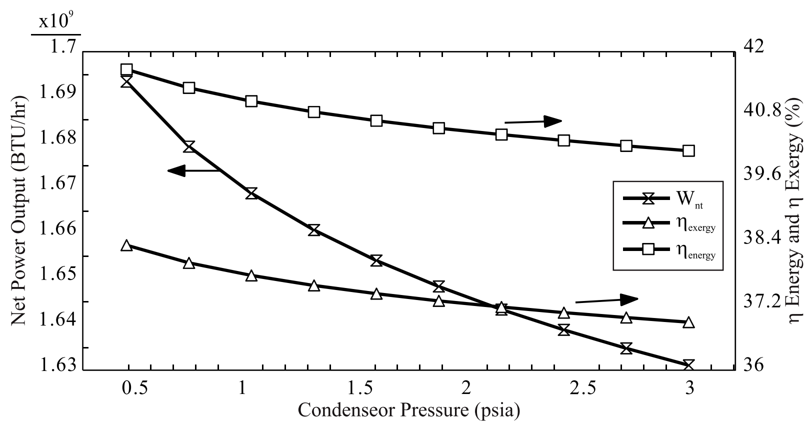

FIG. 4. IMPACT OF CONDENSER PRESSURE ON PERFORMANCE OF THE SYSTEM

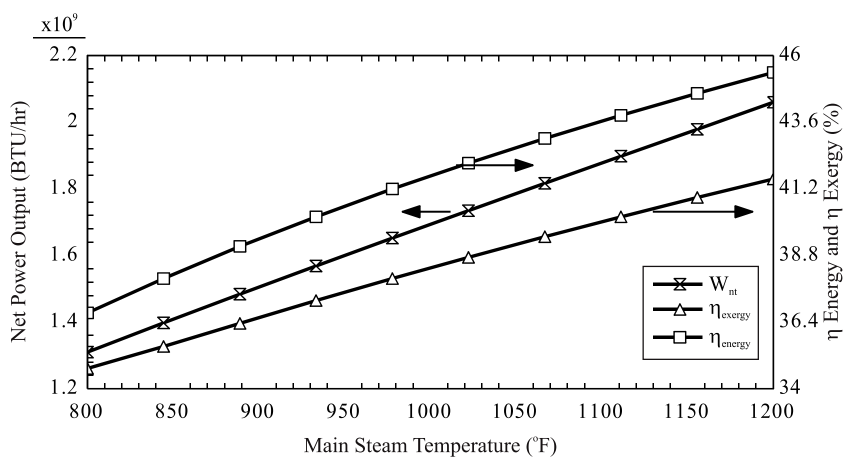

FIG. 5. EFFECT OF STEAM TEMPERATURE ON THE PERFORMANCE OF THE SYSTEM

In fact, the TTD is the difference between the saturated temperature of bled steam and the temperature of feed water coming out of the heater. Therefore, lower TTD increases the heat transfer between the bled steam and the feed water which results in high effectiveness.

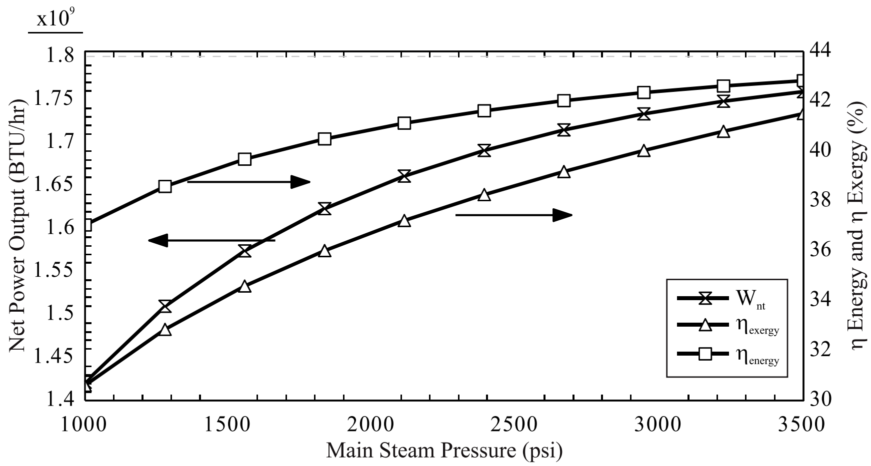

FIG. 6. EFFECT OF STEAM PRESSURE ON PERFORMANCE OF THE SYSTEM

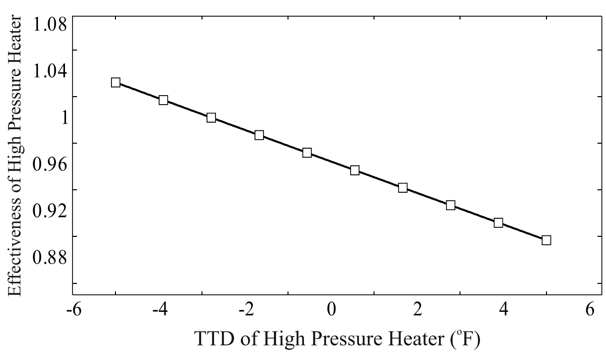

FIG. 7. EFFECT OF TTD ON THE EFFECTIVENESS OF HIGH PRESSURE FWH

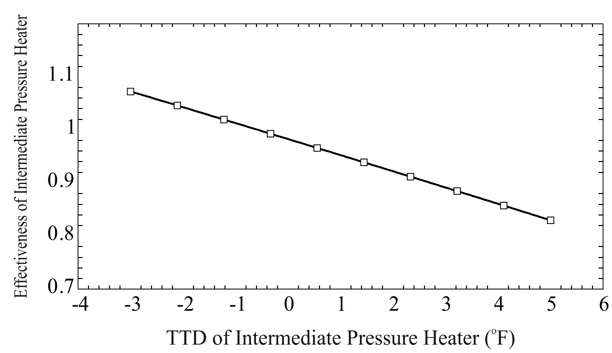

FIG. 9. EFFECT OF TTD ON THE PERFORMANCE OF LOW PRESSURE FWH

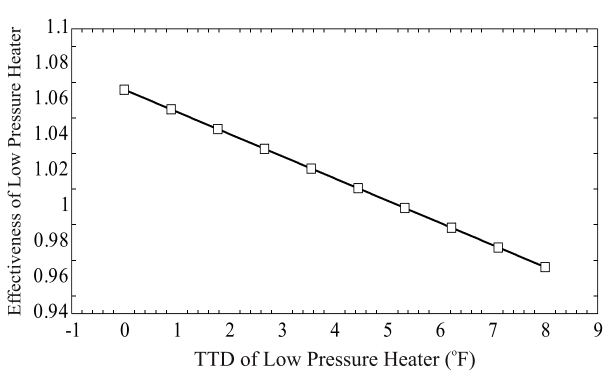

FIG. 8. EFFECT OF TTD ON THE PERFORMANCE OF INTERMEDIATE PRESSURE FWH

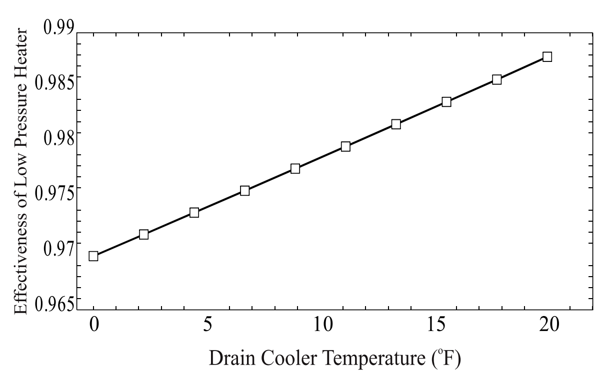

FIG. 10. EFFECT OF DC TEMPERATURE ON THE PERFORMANCE OF LOW PRESSURE FWH

4.2.5 Effect of Drain Cooler Temperature

The variation in DC temperature is only considered for the low pressure FWH in Fig. 10 as an example, although the effect will be applicable for all FWHs. It is clear from figure that the effectiveness of the FWH increases with an increment in the value of the DC temperature.

5. CONCLUSION

In this study, the performance of thermal power plant is conducted in terms of energy and exergy analysis and the effects of various parameters. The energy and exergy analysis portion gives the knowledge about the quantity and quality of the power and the working ability of any system. In this analysis results revealed that the total power output, energy efficiency and the exergy efficiency as 496.3MW, 41.67 and 41.92% respectively. Moreover, the loss in the energy and exergy in different components is determined using the thermodynamic relations. The maximum energy loss is found in the condenser i.e. 72% while the exergy destruction heavily occurs in the boiler i.e. 83%. The parametric analysis reveal that the optimum values for steam pressure,condenser pressure and main steam temperature are 2500, 1.0 psi and 100°F, respectively. It is concluded that the plant may give high efficiency and power output by operating with these values. Finally, it is noted that the performance of feed water heater decreases with TTD and increases with DC temperature.

6. NOMENCLATURE

$\dot{E}n$	Energy (Btu/hr)
Ex	Specific exergy (Btu/lbm)
$\dot{E}x$	Exergy (Btu/hr)
h	Specific enthalpy (Btu/lbm)
I	Exergy destruction rate (Btu/h)
$\dot{m}$	Mass flow rate (lbm/h)
P	Pressure (psia)
s	Specific entropy (Btu/lbm.F)
T	Temperature (F)
$\dot{Q}$	Heat flow rate (Btu)
v	Specific volume (ft3/lbm)
$\dot{W}$	Power (Btu/h)
X	Exergy rate (Btu/h)

Greek Letters

φ	Specific Exergy flow rate (Btu/lbm)
η	Efficiency

Abbreviations

LHV	Lower Heating Value
HPT	High Pressure Turbine
IPT	Intermediate Pressure Turbine
LPT	Low Pressure Turbine
SPP	Steam Power Plant
FWH	Feed Water Heater
HFH	High Pressure Feed Water Heater
IFH	Intermediate Pressure Feed Water Heater
LFH	Low Pressure Feed Water Heater
DEA	Dearator
TTD	Terminal Temperature Difference

Subscripts

b	Boiler
bfp	Boiler Feed Pump
cond	Condenser
cp	Condensate pump
DC	Drain Cooler
e	Exit
en	energy

ex	exergy
F	Fuel
j	Number of carbons
i	Inlet
k	Number of hydrogen
o	out
p	Pump
T	Turbine

ACKNOWLWDGMENT

Authors are thankful to the Department of Mechanical Engineering, Mehran University of Engineering & Technology, Jamshoro, and Department of Mechanical Engineering, Mehran University of Engineering & Technology, Shaheed Zulfiqar Ali Bhutto Campus, Khairpur Mir's, Pakistan, for providing all amenities to conduct this research work.

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