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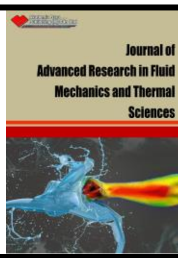

Parametric Assessment of The Thermal Performance of Coal-Fired Power Plant

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ABSTRACT

Exergy analysis has been found to be a useful method for improving the conversion efficiency of energy resources, since it helps to identify locations, types and true magnitudes of wastes and losses. The aim of this research is to analyze the effect of the operation conditions on the performance of coal-fired power plants. As well, this study focuses on the effect of different feedwater heaters' numbers that lead to the highest exergy destruction of the coal-fired power plants. For different values of the superheated steam temperature and the pressure, a parametric study was conducted to determine the efficiency of the coal-fired power plant. The results show that, when the pressures and temperature of the superheated steam increases the evaporator temperature will increases too. Increasing the temperature of evaporator rises the average maximum temperature of the cycle, which improves the thermal efficiency of the cycle as well as the powerplant efficiency. The results show that, at higher boiler pressures and temperatures, the temperature difference between the water/steam and hot gases of the boiler is reduced which means the irreversibility associated with the heat transfer process decreases. Therefore, by increasing the pressure and temperature of the superheater, the exergy efficiency of the thermal cycle is improved. It was observed that operating the coal-fired power plant at high superheated pressure and temperatures produce lead to reduce the exergy losses.

1. Introduction

The first law of thermodynamics remains the most utilized method for analyzing the energy conversion of a system [1-7]. However, this first law-based energy analysis cannot provide the actual losses of the system in terms of efficiency and thermodynamic losses [1, 8-18]. Therefore, it is important that the energy conversion efficiency of a system be analyzed by considering both the first and second laws of thermodynamics [19-24]. Second law-based exergy analysis provides a better view of the energy losses to the environment and the process-related internal irreversibility. It can

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evaluate both process and devices performances as it considers the exergy at different stages during the process of energy conversion [1, 25-38]. The concept of exergy analysis was first detailed by Carnot in 1824 and furthered by Clasisus in 1865. Normally, the first law of thermodynamics (referred to as the first law of analysis) is used to determine the efficiency of a process or a system while the second law of analysis (exergy analysis) relates to entropy production and the concept of concept of irreversibility. The first law of analysis has often been used to calculate energy losses in coal-fired power plants based on the enthalpy balance [39, 40]. However, exergy analysis has gained importance recently over the first law of analysis due to the failure of the first law of analysis to provide enough results that will aid studying the performance of power plants. Exergy analysis primarily aims at identifying the locations of exergy losses, as well as study power plants in terms of their quality [1, 41-52]. Exergy analysis remains a way of locating process or system inefficiencies which first law of analysis cannot do. Exergy analysis also helps in locating the irreversibility within a system and assesses the individual system components in terms of their efficiencies [53]. The outcomes of exergy analysis also help process practitioners in refining and developing coal-fired power plants with optimal performance [54-65]. The exergy of any thermodynamic process reflects its efficiency or inefficiency and provides a better knowledge of the processes for qualifying energy. Therefore, the location, quality, and quantity of energy destruction within a system should better be established based on exergy analysis.

In this study, second law analysis is a methodology for assessment of the performance of the component and involves examining the exergy at different feedwater heaters in a series of energy-conversion steps. This research attempts the development of an integrated strategy for the analysis and improvement of the overall performance of the coal-fired power plants based on the effect of operations conditions.

2. Methodology

In the present study, Unit 1 of Manjung Coal Fired power plant located in Malaysia is considered for investigation. The process flow diagram (PFD) of this power plant is illustrated in Figure 1. This figure does not show the boiler and economizer sections. To analyze the complete cycle of the powerplant, the continuity, energy and exergy equations governing various components of the cycle are developed and resolved using Engineering Equation Solver (EES) software. The continuity equations are invoked to find the distribution of feed water and steam throughout the cycle.

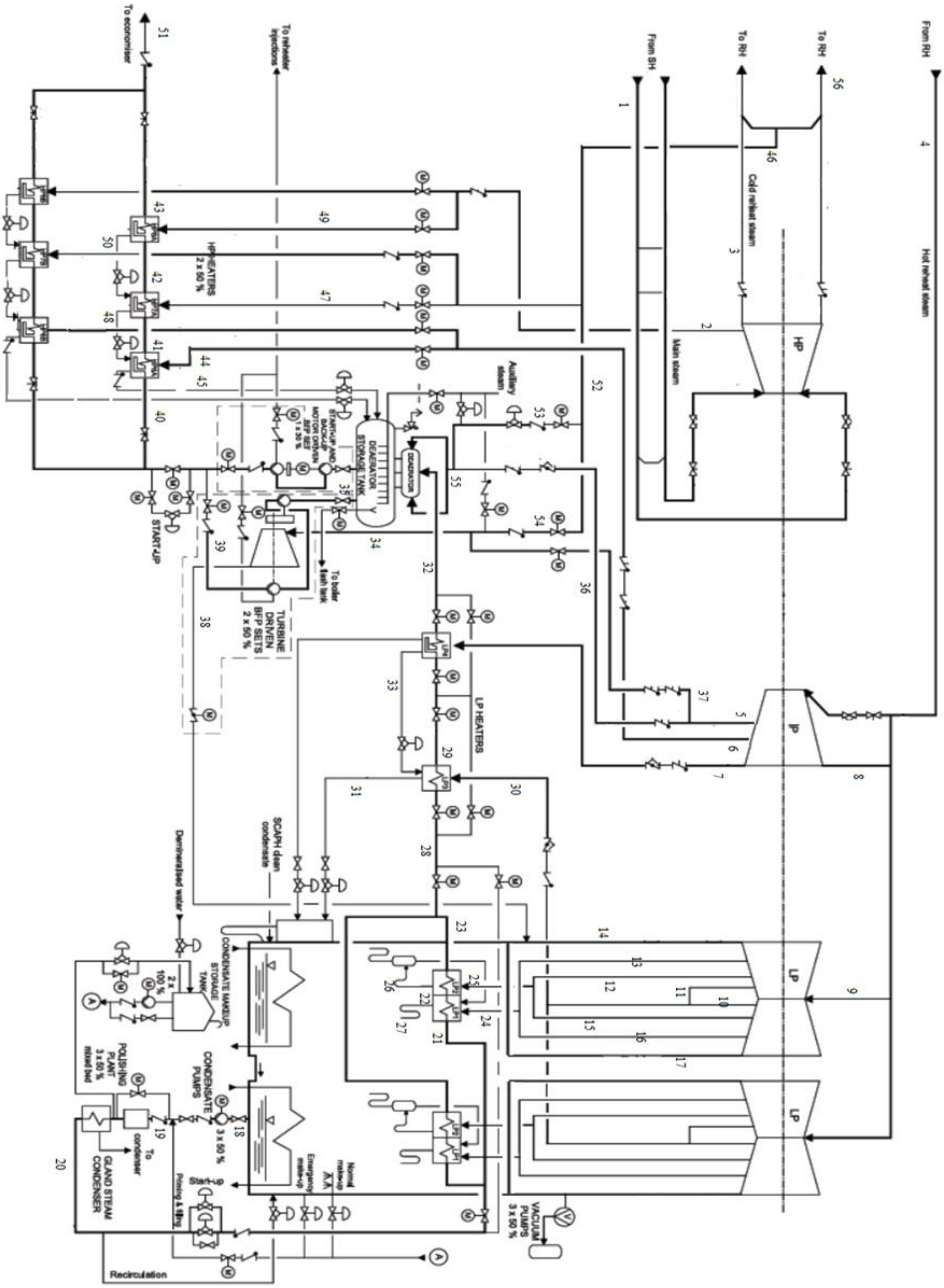

Fig. 1. Process flow diagram of the considered power plant

Table 1 summarizes the energy equations and efficiencies/effectiveness of various components of the thermal cycle; while Table 2 summarizes the exergy supply, exergy destruction and exergy efficiency of those components.

Table 1
Energy equations and efficiencies/effectiveness of various components of the thermal cycle

Component Name	Energy Equation	Energy Efficiency
Boiler	$Q_{Boiler} = m_1 \cdot h_1 + m_4 \cdot h_4 - m_{36} \cdot h_{36} - m_{51} \cdot h_{51}$	$\eta_{Boiler} = \dfrac{Q_{Boiler}}{m_{Coal} HHV_{Coal}}$
Condenser	$Q_{Cond} = 2m_{14} \cdot h_{14} + 2m_{17} \cdot h_{17} + 2m_{27} \cdot h_{27} + m_{31} \cdot h_{31} - m_{18} \cdot h_{18}$	NA
Deaerator	$m_{32} \cdot h_{32} + m_{34} \cdot h_{34} + 2m_{45} \cdot h_{45} - m_{35} \cdot h_{35} = 0$	NA
HPT	$W_{HPT} = m_1 \cdot h_1 - m_2 \cdot h_2 - m_3 \cdot h_3$	$\eta_{s,HP,i} = \dfrac{h_1 - h_i}{h_1 - h_{i,s}}$ $i = 2, 3$
IPT	$W_{IPT} = m_4 \cdot h_4 - m_5 \cdot h_5 - m_6 \cdot h_6 - m_7 \cdot h_7 - m_8 \cdot h_8$	$\eta_{s,IP,i} = \dfrac{h_4 - h_i}{h_4 - h_{i,s}}$ $i = 5, \dots, 8$
LPT	$W_{LPT} = 2\left(m_9 \cdot h_9 - \sum_{i=10}^{17} m_i \cdot h_i \right)$	$\eta_{s,LP,i} = \dfrac{h_9 - h_i}{h_9 - h_{i,s}}$ $i = 10, \dots, 17$
BFP	$W_{BFP} = m_{35} \cdot \left(h_{35} - h_{39} \right)$	$\eta_{s,BFP} = \dfrac{v_{35} \cdot \left(p_{39} - p_{35} \right)}{h_{39} - h_{35}}$
FWH #1	$m_{21} \cdot h_{21} + m_{24} \cdot h_{24} + m_{26} \cdot h_{26} - m_{22} \cdot h_{22} - m_{27} \cdot h_{27} = 0$	$\varepsilon = \dfrac{T_{22} - T_{21}}{T_{24} - T_{21}}$
FWH's #2, #4, #8	$m_{c,in} \cdot h_{c,in} + m_{h,in} \cdot h_{h,in} - m_{c,out} \cdot h_{c,out} - m_{h,out} \cdot h_{h,out} = 0$	$\varepsilon = \varepsilon' \cdot \dfrac{T_{c,out} - T_{c,in,b}}{T_{c,out} - T_{c,in}} + \varepsilon'' \cdot \dfrac{T_{c,in,b} - T_{c,in}}{T_{c,out} - T_{c,in}}$ $\varepsilon' = \dfrac{T_{h,in} - T_{h,sat}}{T_{h,in} - T_{c,in,b}}$, $\varepsilon'' = \dfrac{T_{c,in,b} - T_{c,in}}{T_{h,sat} - T_{c,in}}$
FWH's #3, #6, #7	$m_{h,in} \cdot h_{h,in} + m_{c,in} \cdot h_{c,in} + m_{con,dis} \cdot h_{con,dis} - m_{c,out} \cdot h_{c,out} - m_{h,out} \cdot h_{h,out} = 0$	$\varepsilon = \varepsilon' \cdot \dfrac{T_{c,out} - T_{c,in,b}}{T_{c,out} - T_{c,in}} + \varepsilon'' \cdot \dfrac{T_{c,in,b} - T_{c,in}}{T_{c,out} - T_{c,in}}$ $\varepsilon' = \dfrac{T_{h,in} - T_{h,sat}}{T_{h,in} - T_{c,in,b}}$ & $\varepsilon'' = \dfrac{T_{c,in,b} - T_{c,in}}{T_{h,sat} - T_{c,in}}$

Table 2
Exergy supply, destruction and efficiency of various components of the thermal cycle

Component Name	Exergy Supply	Exergy Destruction	Exergy Efficiency
Boiler	$Exs_{Boiler} = Ex_{Chem,Coal}$ $Ex_{Chem,Coal} = \varphi LHV$ $\varphi = 1.0437 + 0.1896(h/c) + 0.2499(o/c) + 0.0428(n/c)$	$Exd_{Boiler} = Ex_{Chem,Coal} - \begin{pmatrix} Ex_1 + Ex_4 - Ex_{51} \\ -Ex_{36} \end{pmatrix}$	$\eta_{2,Boiler} = \frac{Ex_1 + Ex_4 - Ex_{51} - Ex_{36}}{Ex_{Chem,Coal}}$
Condenser	$Exs_{cond} = Ex_{27} + 2Ex_{14} + 2Ex_{17} + Ex_{31} + Ex_{52}$	$Exd_{condenser} = Exs_{condenser} - (Ex_{53} - Ex_{18})$	$\eta_{2,condenser} = 1 - \frac{Exd_{condenser}}{Exs_{condenser}}$
Deaerator	$Exs_{deaerator} = Ex_{32} + Ex_{34} + 2Ex_{45}$	$Exd_{deaerator} = T_0 S_{gen} = T_0 \begin{pmatrix} m_{35}s_{35} - m_{32}s_{32} - \\ m_{34}s_{34} - 2m_{45}s_{45} \end{pmatrix}$	$\eta_{2,deaerator} = 1 - \frac{Exd_{deaerator}}{Exs_{deaerator}}$
HPT	$Exs_{HPT} = Ex_1 - Ex_2 - Ex_3$	$exd_{HPT} = Exs_{HPT} - W_{HPT}$	$\eta_{2s,HPT} = \frac{W_{HPT}}{Exs_{HPT}}$
IPT	$Exs_{IPT} = Ex_4 - Ex_5 - Ex_6 - Ex_7 - Ex_8$	$Exd_{IPT} = Exs_{IPT} - W_{IPT}$	$\eta_{2s,IPT} = \frac{W_{IPT}}{Exs_{IPT}}$
LPT	$Exs_{LPT} = 2Ex_9 - 2\left(\sum_{i=10}^{17} Ex_i\right)$	$Exd_{LPT} = Exs_{LPT} - W_{LPT}$	$\eta_{2s,LPT} = \frac{W_{LPT}}{Exs_{LPT}}$
BFP	$Exs_{BFP} = W_{BFP}$	$Exd_{BFP} = W_{BFP} - (Ex_{39} - Ex_{35})$	$\eta_{2s,BFP} = \frac{Ex_{39} - Ex_{35}}{W_{BFP}}$
FWH #1	$Exs_{FWH1} = Ex_{24} + Ex_{26b} - Ex_{27}$	$Exd_{FWH1} = Exs_{FWH1} - (Ex_{22} - Ex_{21})$	$\eta_{2,FWH1} = \frac{Ex_{22} - Ex_{21}}{Ex_{24} + Ex_{26b} - Ex_{27}}$
FWH's #2, #4, #8	$Exs = Ex_{h,in} - Ex_{h,out}$	$Exd = Exs - (Ex_{c,out} - Ex_{c,in})$	$\eta_2 = \frac{Ex_{c,out} - Ex_{c,in}}{Ex_{h,in} - Ex_{h,out}}$
FWH's #3, #6, #7	$Exs = Ex_{h,in} + Ex_{con,dis} - Ex_{h,out}$	$Exd = Exs - (Ex_{c,out} - Ex_{c,in})$	$\eta_2 = \frac{Ex_{c,out} - Ex_{c,in}}{Ex_{h,in} + Ex_{con,dis} - Ex_{h,out}}$

In this investigation, Unit 1 of Manjung Coal Fired Power plant located in Malaysia is considered for analysis. This unit produces 700 MW nominally; while according to the acquired data, its actual output is around 590 MW. To have a solid ground for comparison and analysis, a baseline case is assumed. Solving the governing equations developed, different parameters of the power plant are evaluated for the baseline case

3. Results and Discussion

In this section, the impact of changing conditions at different points of the cycle on the main parameters of the cycle and power plant are presented and analyzed. Based on the results, we can conclude what would happen if this situation convey in the real power plant. In fact, in this mode, we assume that the power plant has already been designed and manufactured and is currently in operation. Therefore, different components of the power plant are fixed and cannot be varied. In the following, the effect of changing the thermal cycle conditions on the energy and exergy parameters of the power plant are depicted.

3.1 Superheater Outlet Pressure

The variations of the effectiveness of the feed water heaters with the pressure of the steam exiting the superheater and entering the high pressure turbine are illustrated in Figure 2. It is seen from this figure that heaters #7 and #8 experience remarkable changes with the variation of superheater pressure; while the effectiveness values of other heaters are approximately invariant. This is expectable as changing the superheater pressure mainly changes the steam bled from high pressure turbine and fed to heaters #7 and #8. From the figure, it is observed that increasing the superheater pressure will enhance the effectiveness of heaters #7 and #8. This can be reasoned as following. In all of the feed water heaters except heater #1, the bled steam enters in superheated condition. Therefore, the heaters are composed of two sections: desuperheating section and condensing section. The effectiveness of the condensing section is higher than that of the desuperheating section, because the convection heat transfer coefficient of the condensation is much higher than that of a single phase gas. Considering this matter, we need to note that we have increased the superheater pressure while its temperature is kept fixed. That is, the steam entering the HPT is closer to the saturated vapor curve (possessing lower entropy). Thus, in this case, the desuperheating section will mitigate resulting in higher effectiveness values.

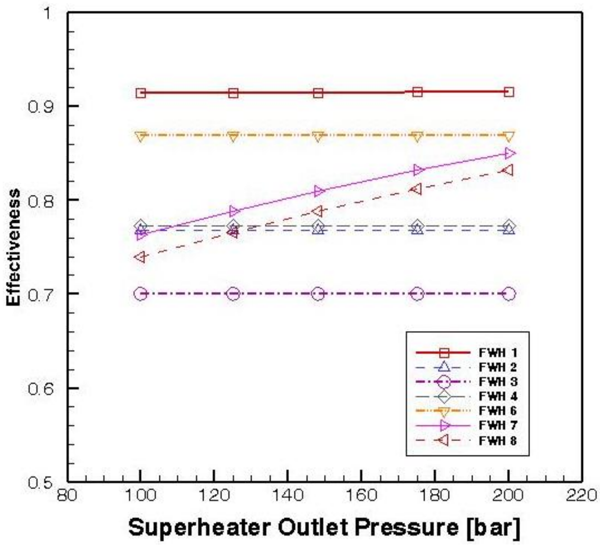

Fig. 2. Variations of FWH's effectiveness with the superheater outlet pressure

The impacts of the superheater pressure on the thermal efficiency and the powerplant efficiency are depicted in Figure 3. It is observed from this figure that higher superheater pressures lead to higher energy based efficiencies. This is due to the increment of the temperature of the evaporator which corresponds to the superheater pressure. In fact, increasing the temperature of evaporator rises the average maximum temperature of the cycle which improves the thermal efficiency of the cycle as well as the powerplant efficiency. This is totally in agreement with what we expect from thermodynamics.

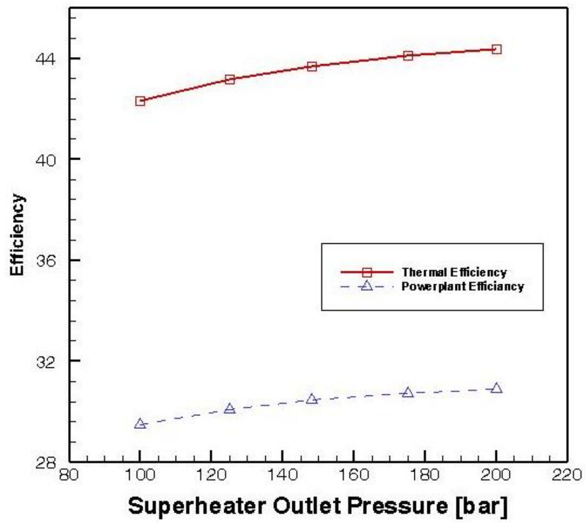

Fig. 3. The effect of superheater outlet pressure on thermal efficiencies of the cycle and power plant

Figure 4 shows the variations of exergy efficiency of the cycle versus superheater pressure. Here again, the higher superheated steam pressures cause higher exergy efficiencies. This is because, at higher boiler pressures, the temperature difference between the water/steam and hot gases of the boiler is reduced which means the irreversibility associated with the heat transfer process decreases. Therefore, by increasing the pressure of the superheater, the exergy efficiency of the thermal cycle is improved.

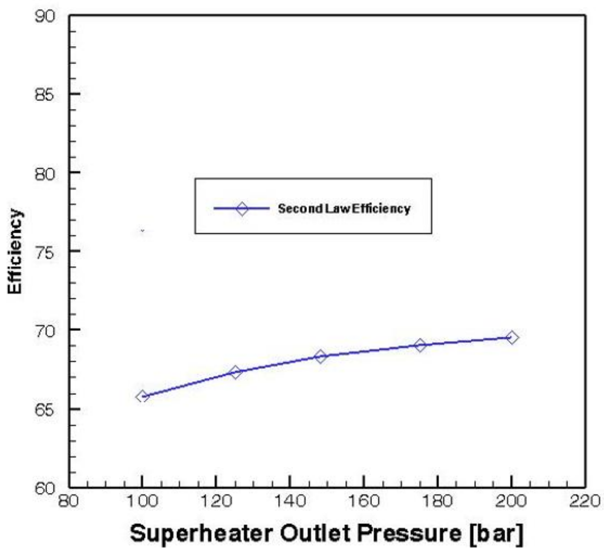

Fig. 4. The effect of superheater outlet pressure on the exergy efficiency of the cycle

The variations of the fuel consumption rate, heat rate of the cycle and the net power delivered by the cycle with the superheater pressure are given in Table 3. From this table, it is clear that increasing the superheater pressure increases the coal consumption moderately; that is, by increasing the superheater pressure by 100%, the coal consumption rate just increases up to 3%; while the net power delivered by the cycle increases more than 8%. It is because that, the coal consumption was increasing so steam mass flow rate will increases too. This shows the importance of increasing the pressure of the superheater. This issue is also reflected in the heat rate of the cycle. As is seen from the table, increasing the superheater pressure from 100 to 200 bar lowers the heat rate of the cycle from 2.364 to 2.254.

Table 3
The effect of superheater outlet pressure on the Heat Rate of the cycle, net power delivered by the cycle and mass flow rate of coal consumed in the boiler

Superheater Outlet Pressure [bar]	$\dot{m}_{Coal}$ [kg/s]	HR_{cycle}	$\dot{W}_{net}$ [MW]
100	86.32	2.364	559.330
125	87.47	2.316	578.333
148.1	88.17	2.288	590.234
175	88.70	2.266	599.472
200	88.98	2.253	604.844

Figure 5 represents the variation of exergy efficiencies of different turbines versus the superheater pressure. As changing, the pressure of superheater has no tangible effect on IPT and LPT, the exergy efficiencies of these turbines do not vary with superheater pressure, expectedly. However, the exergy efficiency of HPT decreases with the increase in the superheater pressure. Because, increasing the pressure of the steam entering HPT while its temperature is kept fixed, shifts the steam state towards the saturation vapor line. Hence, the specific volume of steam decreases and the steam enters the saturation dome sooner, which leads to higher irreversibility.

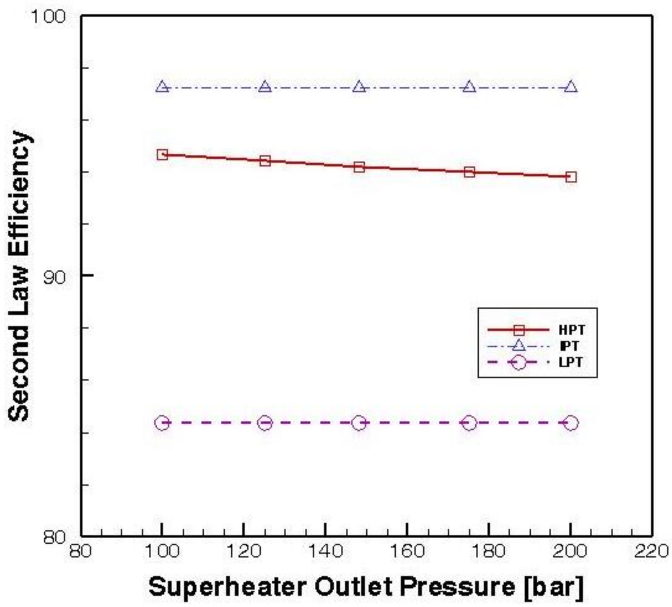

Fig. 5. Effect of the superheater outlet pressure on the second law efficiencies of different turbines

The dependency of the exergy efficiency of different feed water heaters on the superheater pressure is shown in Figure 6. Here, it is seen that the second law efficiencies of the low pressure heaters; i.e. heaters #1-#5 are not affected by the superheater pressure changes. However, the exergy efficiency of heaters #6-#8 are improved with the superheater pressure rise. Similar to what was discussed for Figure 5, higher superheater pressures lead to the shift of the states of the extracted steam from HPT towards the saturated vapor line. Thus, the temperature difference in the heaters will reduce. Therefore, the entropy generation and irreversibility associated with the heat transfer inside the heaters will mitigate.

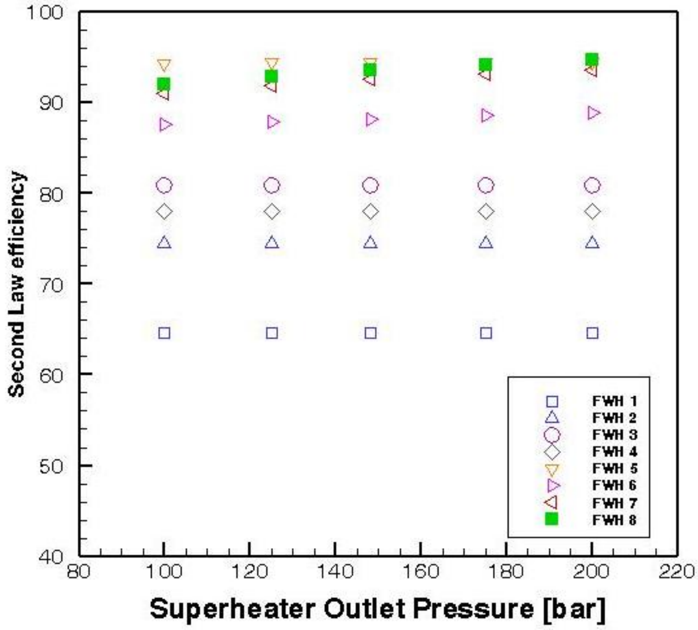

Fig. 6. Effect of the superheater outlet pressure on the exergy efficiencies of the heaters

3.2. Superheater Outlet Temperature

Graphed in Figure 7 are the variations of the effectiveness of the feed water heaters with the temperature of the steam exiting the superheater and entering the high pressure turbine. As the temperatures of the steam streams after the superheater and after the reheater are always of the same order, we assumed that this correlation conveys between these temperatures. That is, any change in the temperature of the superheated steam entering the HPT will change the temperature of the reheated steam entering IPT.

From figure 7, it is seen that the effectiveness of the high pressure heaters deteriorates with the increase of the superheater outlet temperature. The reasoning is similar to that of Figure 2; however the trend is reverse. Here, by increasing the superheater outlet temperature while the pressure is kept fixed, the bled steam deviates more from the saturated vapor line. Therefore, the desuperheating section extends which leads to lower effectiveness values. It is noteworthy that the effectiveness of the condensing section is not affected by changing the superheater outlet temperature. Moreover, the effectiveness values of the low pressure heaters are approximately invariant. This result is consistent with what was seen in Figure 2.

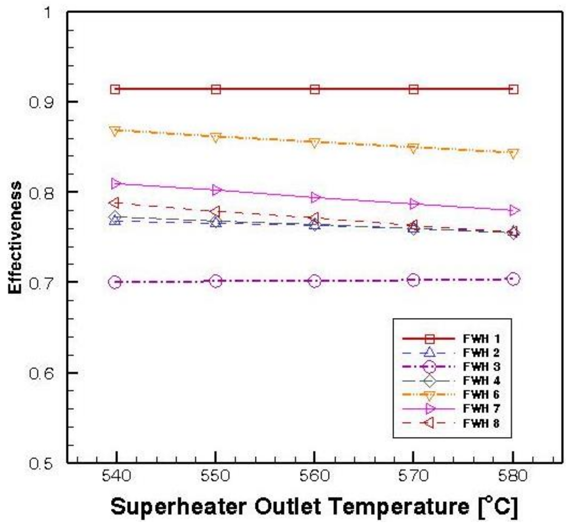

Fig. 7. Variations of FWH's effectiveness with the superheater outlet temperature

The effect of increasing the temperature of the steam exiting the superheater on the thermal efficiency and the power plant efficiency are illustrated in Figure 8. It is observed from this figure that the higher superheated steam temperatures correspond to higher thermal and power plant efficiencies. This is due to the increment of the average temperature of the steam passing through the whole boiler including economizer, evaporator and superheater sections, while the pressure and thus the temperature of the condenser are kept fixed.

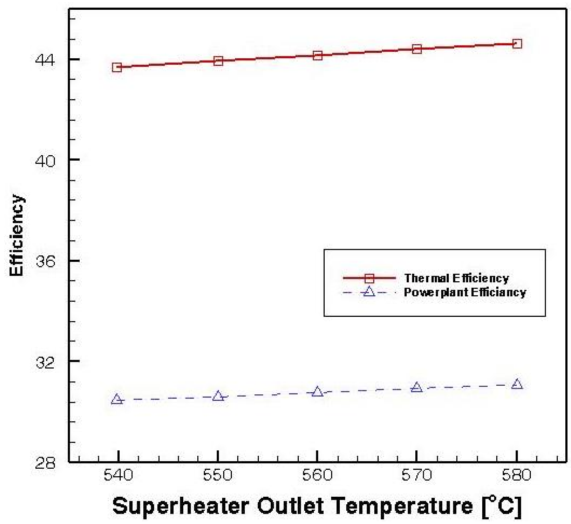

Fig. 8. The effect of superheater outlet temperature on thermal efficiencies of the cycle and powerplant

Figure 9 shows the variations of the exergy efficiency of the cycle versus the inlet temperature of the steam entering high pressure turbine. From this figure, it is evident that the higher superheated steam temperatures result in higher exergy efficiencies. Similar to what presented in Figure 3, this is because, at higher boiler temperatures, the temperature gap between the water/steam and hot gas streams is reduced leading to lower irreversibility associated with the heat transfer across finite temperature gaps. Therefore, by increasing the temperature of the superheated steam, the exergy efficiency of the thermal cycle is enhanced.

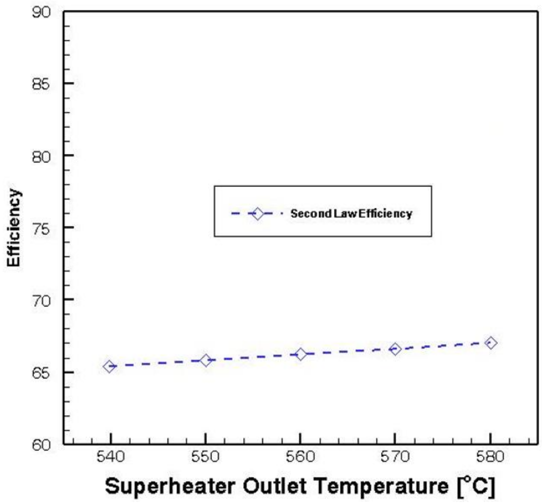

Fig. 9. The effect of superheater outlet temperature on the exergy efficiency of the cycle

Table 4 gives the variations of the fuel consumption rate, heat rate of the cycle and the net power delivered by the cycle with the temperature of the superheated steam entering the high pressure turbine. From this table, it is seen that the coal consumption is increased by the increment of the superheated steam temperature. By increasing the superheated steam temperature from 539.8 to 580°C, the coal consumption rate increases around 4%; while the net power delivered by the cycle increases more than 6%. This shows the role of increasing the temperature of the superheated steam. From the heat rate standpoint, it is seen that increasing the superheated steam temperature from 539.8 to 580°C, reduces the heat rate of the cycle from 2.288 down to 2.242.

Table 4
The effect of superheater outlet temperature on the Heat Rate of the cycle, net power delivered by the cycle and mass flow rate of coal consumed in the boiler

Superheater Outlet Temperature [°C]	$\dot{m}_{Coal}$ [kg/s]	HR_{cycle}	$\dot{W}_{net}$ [MW]
539.8	88.17	2.288	590.234
550	89.10	2.275	599.693
560	90.01	2.264	608.966
570	90.92	2.252	618.241
580	91.82	2.241	627.523

Figure 10 represents the variation of exergy efficiencies of different turbines with the superheater outlet temperature. Similar to Figure 5, as the change in the temperature of the superheated steam entering HPT has slight effect on IPT and LPT, the exergy efficiency of these turbines do not vary with superheated steam temperature, remarkably. However, the exergy efficiency of HPT is enhanced with the increase in the superheated steam temperature. Because, increasing the temperature of the steam entering HPT while its pressure is kept fixed dislocates the steam state further from the saturation vapor line. Hence, the specific volume of steam increases and the steam entrance into the saturation dome is delayed which leads to lower irreversibilities.

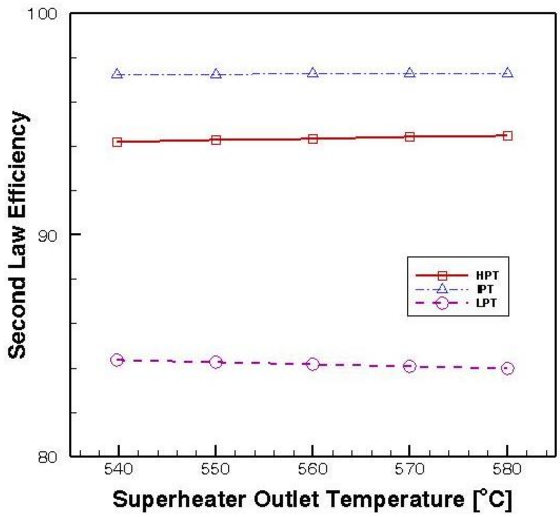

Fig. 10. Effect of the superheater outlet temperature on the second law efficiencies of different turbines

The second law efficiency of different feed water heaters varies with the superheated steam temperature as shown in Figure 11. It is seen that all the exergy efficiencies are reduced by increasing the temperature of the superheated steam, slightly. Unlike what we have seen in Figure 6, here the behavior of all the heaters is similar, because the temperature of the reheated steam is also increased. Moreover, the reason for this descending behavior lies in this fact that increasing the superheated steam temperature expands the region of desuperheating sections of the heaters which possess higher temperature gaps between the streams.

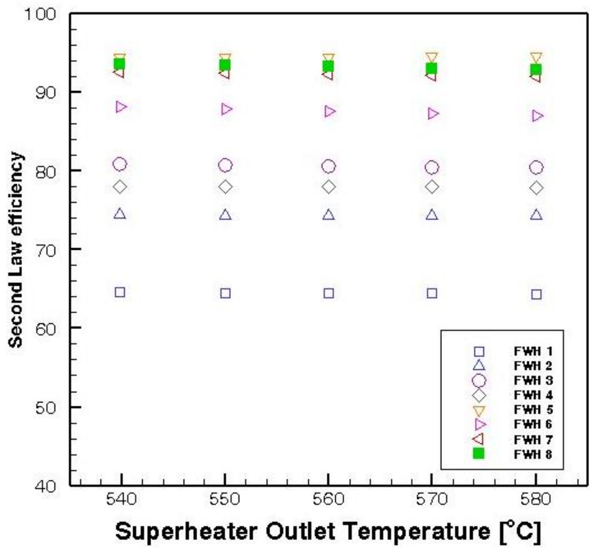

Fig. 11. Effect of the superheater outlet temperature on the exergy efficiencies of the heaters

4. Conclusion

This study is devoted to analyze a coal-fired power plant, to investigate the impact of the operational conditions as well as different configurations of the feed water heaters on the energy and exergy performances of the coal-fired power plant.

i. Higher superheater pressures lead to higher energy and exergy based efficiencies.
ii. By increasing the superheater pressure by 100%, the net power delivered by the cycle increases more than 8%.
iii. By increasing the superheated steam temperature from 539.8 to 580℃, the net power delivered by the cycle increases more than 6%.
iv. The main impact of superheated steam temperature is observed in the exergy supplies and destructions to/at FWH's #1 (3.78%), #4 (3.58%) and #6 (4.33%) which receive steam from the LPT, IPT and IPT, respectively.
v. The exergy efficiency of heaters #2 is increased by increasing the temperature of the superheated steam; while the behavior of other closed feed water heaters is reverse.

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