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ENERGY AND EXERGY ANALYSIS OF A REHEAT REGENERATIVE VAPOR POWER CYCLE

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ABSTRACT

The present paper describes the energy and exergy analysis of a reheat regenerative vapor power cycle. The plant consists of one boiler feed pump, one supercritical boiler, two steam turbines with a reheater in between, two feed water heaters (one open and one closed) and a condenser. The energy and exergy balance study has been carried-out for each component of the plant. Energy efficiency, exergy efficiency and the irreversibility results obtained from the simulation has been presented in the form of graphs. The supercritical pressure is varied from 250 to 400 bar in a step of 50 bar and for each supercritical pressure the temperature has been changed from 500 to 800K. The parametric study reveals that the cycle energy and exergy efficiency increases with increase in pressure and temperature. This is due to reduced energetic and exergetic losses at increased pressure and temperature. While estimating the irreversibility and fractional exergy losses of various system components it is found that maximum irreversibility occurs in the boiler which accounts for 46-55% at 350 bar for the given temperature range. The fractional exergy loss in turbine, condenser, BFP, mixing chamber, OFWH and CFWH are found to 20-30%, 13-14%, 1.6 -2%, 0.37 – 0.64%, 1.6-2.8% and 1.25- 2.15% respectively at 350 bar inlet pressure and for the given temperature range. This combined study on energy and exergy analysis thus gives a better insight into the cycle operation with various system components and components requiring operational and design modifications for minimizing losses.

Keywords: Energy, Exergy, Irreversibility, Supercritical boiler, Rankine cycle.

1. INTRODUCTION

The work output is maximized when the process between the two specified states is executed in a reversible manner. A system delivers the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state. The study of thermodynamic cycles applied to power stations is of great importance due to the increasing energy consumption, the opening of electricity markets and the rising environmental restrictions, specifically in the carbon dioxide emissions issue. Power plants that use steam as their working fluid work on the basis of Rankine cycle. The first stage in designing these power plants is the thermodynamic analysis process of the Rankine cycle. Also the second law analysis of these cycles reveals where the largest irreversibilities occur and what their magnitudes are. Now for evaluating second law efficiency of a thermodynamic cycle it's very important for us to know the concept of exergy.

Exergy is the maximum useful work that could be obtained from the system at a given state in a specified environment. Steam reheating is important in the case of a vapor power plant, because it increases the quality of steam at the turbine inlet which prevents the corrosion of turbine blades, and under certain conditions the thermal efficiency of the plant also increases. The effect of reheat alone on the thermal efficiency of the cycle is very small. Regeneration or the heating up of feedwater by steam extracted from the turbine has a marked effect on cycle efficiency. That's why both are equipped in the current analysis of supercritical rankine cycle.

The optimization of reheat regenerative thermal-power plants has been analyzed [1-2] for the subcritical pressure range. The exergetic analysis and optimization has been done for the supercritical rankine cycle [3]. Generalized Thermodynamic Analysis of Steam Power Cycle with 'n' number of Feed Water Heaters has been analyzed.

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In addition to the energy analysis, a full-exergy analysis helps to identify components where high inefficiencies occur. Improvements should be done to these components to increase efficiency. The thermodynamic cycle is optimized by minimizing the irreversibilities. At full load of the cycle, the steam temperature and pressure of the boiler should be at their upper limits. However, at off-design loads, the temperature and pressure should be decreased. An exergy evaluation of a supercritical steam turbine showed that high exergy losses occur in the heat recovery steam generator and in the steam turbine. The thermal efficiency of cycle is increased to a large extent by operating the steam boiler above the critical pressure, i.e., supercritical pressure cycle [4]. However, at supercritical condition, more quantity of heat is required to generate the steam and also the metallurgical limitations of the materials. Basics, energy, exergy, chemical process, mass balance, heat balance, Analysis of various steam power cycles has been analyzed [5-8]. Thermodynamic properties of steam power plant have been developed by using ISI steam tables [9]. Mostly coal is used as the fuel in the furnace to superheat the water in the boiler.

In the present paper the energy and exergy analysis of a reheat, regenerative and supercritical cycle is done for a temperature and pressure range of 500-800°C and 250-400 bar respectively. For the purpose a MATLAB code was developed for system simulation.

2. THERMODYNAMIC ANALYSIS

2.1 SYSTEM CONFIGURATION

The schematic of the reheat regenerative supercritical steam power cycle is shown in Fig. 1. The system consists of the supercritical boiler, the reheat steam turbine, the condenser, one open feed water heater and one closed feed water heater, three pumps and one mixing chamber. Fig. 2 shows the corresponding T-s diagram of the system layout.

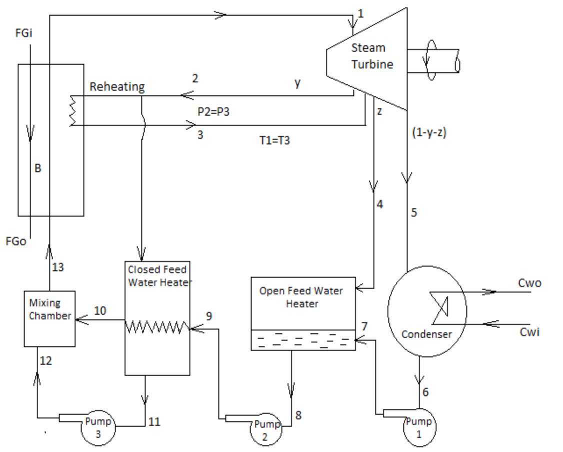

Fig. 1: The Schematic diagram of the system layout

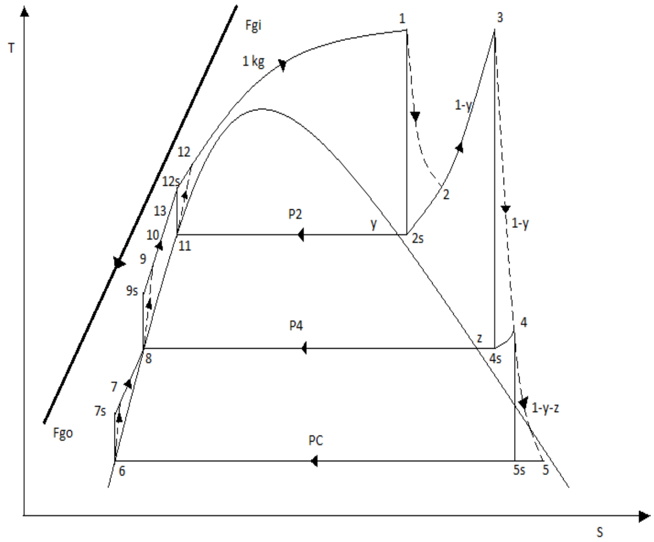

Fig. 2: T-s diagram of the cycle

In the corresponding T-s diagram, the solid lines represent the ideal processes in the boiler, steam turbine, OFWH, CFWH, condenser and pumps while the dotted lines represent the non-ideal processes in the steam turbine and pumps. In this case perfect reheating is considered, so that T1=T3.

2.2 ASSUMPTIONS USED IN THE PRESENT ANALYSIS:

1. Capacity of the power plant = $1000 MW$.
2. Reheat pressure = 0.2 times the initial pressure.
3. The isentropic efficiency of the steam turbine is 90%.
4. The pump efficiency is assumed to be 85%.
5. Flue gas entering into the boiler is $T_{gi} = 1000^o C$ and leaving is $T_{go} = 100^o C$.
6. The pinch point temperature difference in the condenser is $6^o C$.
7. Condenser pressure $P_c = 0.1$ bar.
8. Terminal Temperature Difference (TTD) in feed water heater = $3^o C$.
9. Cooling water temperature inlet to the condenser $T_{wi} = 25^o C$.
10. No heat losses and no pressure losses.

2.3 ENERGY ANALYSIS OF THE CYCLE:

Now for the energy analysis of the present supercritical rankine cycle we have to evaluate the power developed by the turbine, power consumed by the pump and the total heat transfer during the supercritical process, and for this purpose enthalpy at each and every point from 1 to 13 should be known to us.

The reheat Pressure $P_{rh} = (0.2)P1$ (1)

Location of the feed water heater is calculated from the Nag [7] is

$$T_x = (T_B - T_C)/(n+1) \quad (2)$$

Where,

T_x = Temperature rise per heater for maximum efficiency.
T_B = Saturation temperature at pressure P_1
T_C = Saturation temperature at Condenser pressure
n = number of feed water heaters
$P2$ is the saturation pressure of the temperature $T2s$

$$T_{4s} = T_{2s} - T_x \quad (3)$$

All the nodal points (h, s) in the above T-s diagram have been calculated from the basic thermodynamic principles of the actual supercritical cycle with the above assumptions.

The temperature at nodal point 10 in T-s diagram of the Fig.2,

$$T_{10} = T_{11} - TTD \quad (4)$$

Al1 components associated with the cycle are steady flow devices, and thus, al1 processes that make up the cycle can be analyzed as steady flow processes. The kinetic and potential energy changes of the steam are usually smal1 relative to the work and heat transfer terms and, therefore, are usually neglected.

The steady flow energy equation per unit mass of steam reduces to

$$q - w = h_e - h_i \quad (5)$$

The boiler and condenser don't involve any work, and the pumps are assumed to be isentropic. Then, the conservation of energy relation for these devices can be expressed as follows:

$$W_{pump1} = h_7 - h_6 \text{ kJ/kg} \quad (6)$$

$$W_{pump2} = h_9 - h_8 \text{ kJ/kg} \quad (7)$$

$$W_{pump3} = h_{12} - h_{11} \text{ kJ/kg} \quad (8)$$

The closed feedwater heater, the open feedwater heater and the mixing chamber are assumed to be wel1 insulated. Therefore, they don't involve any work or heat transfer. Then, the conservation of energy and mass equations reduce to

$$\sum m_i h_i = \sum m_e h_e \quad (9)$$

Applying equation (9) to these devices gives, we get,

Open FWH (q = 0, w = 0)

$$z h_4 + (1 - y - z) h_7 = (1 - y) h_8 \quad (10)$$

$$z = (1 - y)(h_8 - h_7)/(h_4 - h_7) \quad (11)$$

Closed FWH (q = 0, w = 0)

$$y(h_2 - h_{11}) = (1 - y)(h_{10} - h_9) \quad (12)$$

$$y = (h_{10} - h_9)/((h_2 - h_{11}) + (h_{10} - h_9)) \quad (13)$$

Mixing chamber (q = 0, w = 0)

$$h_{13} = (1 - y)h_{10} + y h_{12} \quad (14)$$

The process of expansion of steam in the turbine is assumed to be internally irreversible with some increase in entropy. Then, the conservation of energy relation can be expressed as follows:

$$W_{turbine} = (h_1 - h_2) + (1 - y)(h_3 - h_4) + (1 - y - z)(h_4 - h_5) kJ/kg \quad (15)$$

$$W_{pump} = (1 - y - z)W_{pump1} + (1 - y)W_{pump2} + yW_{pump3} kJ/kg \quad (16)$$

The total heat added is the sum of the energy added by heat transfer during supercritical process. When expressed on the basis of a unit mass entering the turbine:

$$HS = (h_1 - h_{13}) + (1 - y)(h_3 - h_2) kJ/kg \quad (17)$$

The cycle efficiency or energy efficiency or first law in the efficiency is defined as the ratio of output energy to the input energy,

Energy Efficiency = $(W_{turbine} - W_{pump})/HS$ (18)

2.4 EXERGY ANALYSIS OF THE CYCLE:

2.4.1 Boiler:

It is a common practice to use high pressure and temperature boilers to increase the efficiency of the supercritical rankine cycle.

Mass of stream generated for the given flow rate of flue gases is obtained from the energy balance. The mass of the steam is calculated from the capacity of the

$$m_s(W_{net}) = 1000MW \tag{19}$$

where, $W_{net} = (W_{turbine} - W_{work})$

Thus,

$$m_s = 1000 \times 1000kW / (W_{turbine} - W_{pump})kg/s \tag{20}$$

Energy balance equation for obtaining $m_g C_{pg}$ is,

Heat gained by the steam = Heat lost by the flue gases.

$$m_s((h_1 - h_{13}) + (1-y)(h_3 - h_2)) = m_g C_{pg}(T_A - T_B) \tag{21}$$

Thus,

$$m_g C_{pg} = m_s((h_1 - h_{13}) + (1-y)(h_3 - h_2)) / (T_A - T_B) \tag{22}$$

The irreversibility or exergy loss in the boiler is obtained as decrease in availability function across the component.

Exergy of the flue gas entering the boiler, for the given temperature $\theta_A = 1000^o C$ and $\theta^o = 25^o C$ the composition of the flue gas has been calculated and enthalpy and exergy of the flue gas entering in to the boiler and leaving the boiler are as,

Exergy in the flue gas at the entering the boiler is $Ex_{in} = Ex_A$. Enthalpy of the flue gas entering H_A Exergy in the flue gas at the exit the boiler is $Ex_{out} = Ex_B$ and Enthalpy of the flue gas at exit of the boiler H_B are calculated.

Irreversibility in the boiler is

$$I_{boi} = \left(m_g C_{pg}(T_{g1} - T_{g2}) - m_g C_{pg} T_o \ln\left(\frac{T_{g1}}{T_{g2}}\right)\right) - \left(m_s \begin{pmatrix} ((h_1 - h_{13}) + (1-y)(h_3 - h_2)) - \\ T_o((s_1 - s_{13}) + (1-y)(s_3 - s_2)) \end{pmatrix}\right) \tag{23}$$

2.4.2 Steam Turbine:

The irreversibility rate in the steam turbine given by Gouy-Stodola equation is

$$I_{tur} = m_s T_o \begin{pmatrix} (s_2 - s_1) + (1-y)(s_4 - s_3) + \\ (1-y-z)(s_5 - s_4) \end{pmatrix} \tag{24}$$

2.4.3 Condenser:

Mass of cooling water circulated to condense m_s kg of steam is obtained from the energy balance is

$$m_s(1-y-z)(h_5 - h_6) = m_{cw} C_{pw}(T_{wi} - T_{wo}) \tag{25}$$

where, $C_{pw} = 4.1868kJ/kg.K$

$$m_{cw} = m_s(1-y-z)(h_5 - h_6) / C_{pw}(T_{wi} - T_{wo}) \tag{26}$$

Irreversibility in the condenser,

$$I_{con} = T_o\left(m_{cw} C_{pw} \ln\frac{T_{wo}}{T_{wi}}\right) - m_s(1-y-z)(s_5 - s_6) \tag{27}$$

2.5.1 Pump1: Irreversibility in pump1,

$$I_{p1} = m_s T_o(1-y-z)(s_7 - s_6) \tag{28}$$

2.5.2 Pump2: Irreversibility in pump2,

$$I_{p2} = m_s T_o(1-y)(s_9 - s_8) \tag{29}$$

2.5.3 Pump3*:* Irreversibility in pump3,

$$I_{p3} = m_s T_o(y)(s_{12} - s_{11}) \tag{30}$$

Sum of irreversibilities of all three pumps is total irreversibility in the pump:

$$I_{pump} = I_{p1} + I_{p2} + I_{p3} \tag{31}$$

2.5.4 Open FWH: Irreversibility in Open FWH,

$$I_{ofwh} = m_s T_o((1-y)s_8 - zs_4 - (1-y-z)s_7) \tag{32}$$

2.5.5 Closed FWH: Irreversibility in Closed FWH,

$$I_{cfwh} = m_s T_o(y(s_{11} - s_2) + (1-y)(s_{10} - s_9)) \tag{33}$$

2.5.6 Mixing chamber: Irreversibility in mixing chamber,

$$I_{mc} = m_s T_o((S_{13} - (1-y)S_{10} - yS_{12})) \tag{34}$$

2.4.7 Exhaust: Irreversibility or exergy loss through the exhaust

$$I_{exhaust} = m_g C_{pg}(T_{g2} - T_{go}) - m_g C_{pg} T_o \ln\frac{T_{g2}}{T_o} \tag{35}$$

Total Irreversibility is

$$\sum I = I_{boi} + I_{tur} + I_{con} + I_{pump} + I_{ofwh} + I_{cfwh} + I_{mc} + I_{exhaust} \tag{36}$$

$$E_s = m_g C_{pg}\left(T_{g1} - T_{go}\right) - m_g C_{pg} T_o \ln \frac{T_{g1}}{T_o} \quad (37)$$

Exergy efficiency is defined as the ratio of exergy output to the exergy input. Exergy output depends on the degree of Irreversibility of the cycle.

$$\text{Exergy efficiency} = \left(\left(E_s - \sum I\right)/E_s\right) \times 100 \quad (38)$$

3.0 Fractional Exergy Loss:

The fractional exergy loss of the component is defining as the ratio of irreversibility of the individual component to the total irreversibility of the cycle. Its value is estimated for all the components of the cycle. It gives the information regarding the loss of useful energy in all the component has been studied with different operating variables. The Fractional exergy formulas are as follows.

1. Fractional exergy loss in the boiler is,

$$\left(I_{boi} / \sum I\right) \times 100 \quad (39)$$

2. Fractional exergy loss in the turbine is,

$$\left(I_{tur} / \sum I\right) \times 100 \quad (40)$$

3. Fractional exergy loss in the condenser is,

$$\left(I_{con} / \sum I\right) \times 100 \quad (41)$$

4. Fractional exergy loss in the open fwh is,

$$\left(I_{ofwh} / \sum I\right) \times 100 \quad (42)$$

5. Fractional exergy loss in the closed fwh is,

$$\left(I_{cfwh} / \sum I\right) \times 100 \quad (43)$$

6. Fractional exergy loss in the mixing chamber is,

$$\left(I_{mc} / \sum I\right) \times 100 \quad (44)$$

7. Fractional exergy loss in the Pump is,

$$\left(I_{pump} / \sum I\right) \times 100 \quad (45)$$

8. Fractional exergy loss in the exhaust is,

$$\left(I_{exhaust} / \sum I\right) \times 100 \quad (46)$$

4.0 Results and Graphs:

The energy and exergy analysis has been carried out for the reheat-regenerative rankine cycle using one open and one closed feed water heater. Fractional exergy losses of each component of the cycle are determined and the results are shown in the form of graphs. Figure 3 shows the variations of energy efficiency as the function of temperature for different boiler pressure. It can be seen from the graph that for a fixed turbine inlet temperature, say, 600^{o}C the efficiency increases from 46.073% to 49.936% when the boiler pressure increases from 250 bar to 400 bar. Similarly, at a fixed boiler pressure, say, 300 bar the efficiency increases from 45.528% to 52.518% as the turbine inlet temperature increases from 500^{o}C to 800^{o}C which can be seen from the Fig 4.

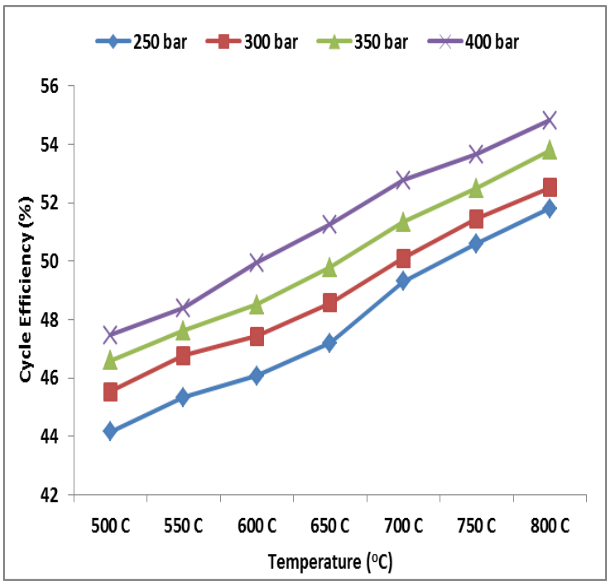

Fig. 3: Variation of Cycle efficiency at different turbine inlet temperature values with increase in pressure at Pc=0.1bar

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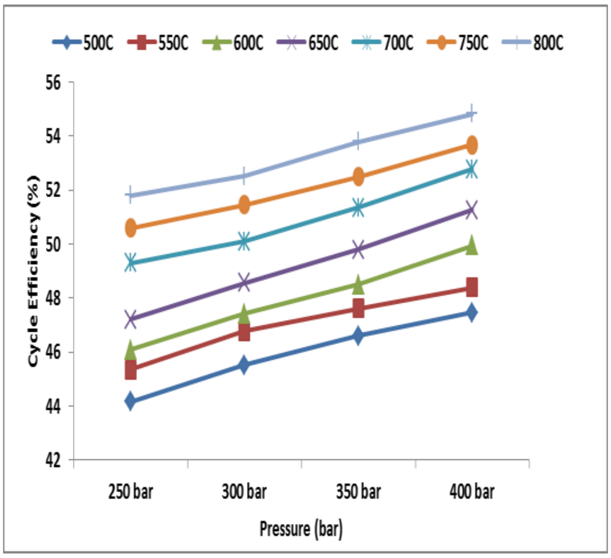

Fig. 4: Variation of cycle efficiency at different turbine inlet pressure values with increase in tempreture at Pc=0.1 bar

Actually with the increase in boiler pressure and turbine inlet temperature the enthalpy of steam at inlet to the turbine increases and condenser pressure being fixed at 0.1 bar, the gain in work output from the turbine is more compared to the amount of heat supplied in the boiler. Therefore efficiency is more at higher turbine inlet temperature and higher boiler pressure. It is found that the variation in the cycle efficiency with the increase in temperature is higher than an increase in the pressure.

Fig. 5 and Fig. 6 show the variation of exergy efficiency as a function of turbine inlet temperature at different boiler pressure and as a function of boiler pressure at different turbine inlet temperature respectively.

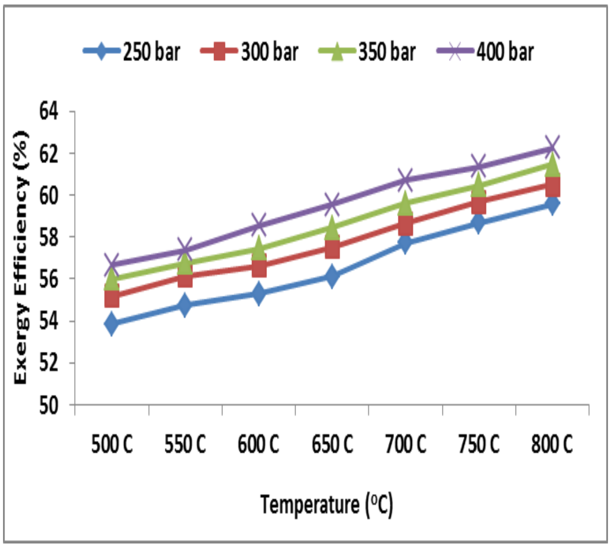

Fig. 5: Variation of exergy efficiency at different turbine inlet temperature values with increase in pressure at Pc=0.1 bar.

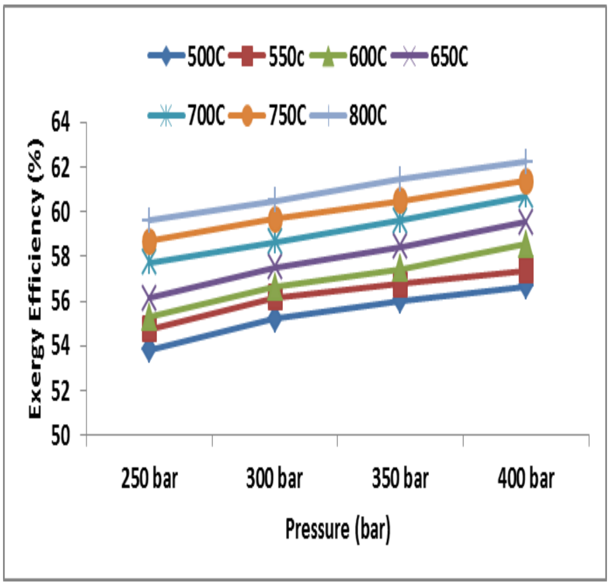

Fig. 6: Variation of exergy efficiency at different turbine inlet pressure values with increase in temperature at Pc=0.1bar

From the Fig. 5 we can see that for a fixed turbine inlet temperature of, say, 600°C the exergy efficiency for 250 bar is 55.288% and for 400 bar is 58.543%. Similarly, the exergy efficiency at 500°C is 55.206% and at 800°C is 60.498% at a fixed boiler pressure of, say, 300 bar which can be seen from the Fig. 6.

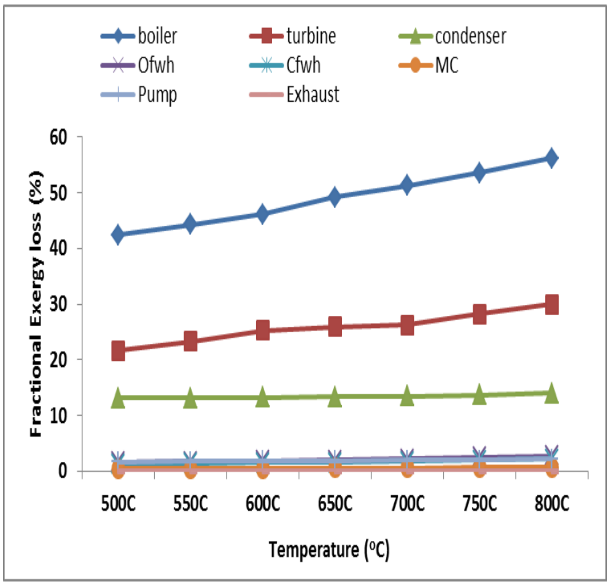

Fig. 7: Effect of turbine inlet tempreture on fractional exergy loss of different componets, T_{FGi}=1000°C, T_{FGo}=100°C, P=250 bar, P_c=0.1bar

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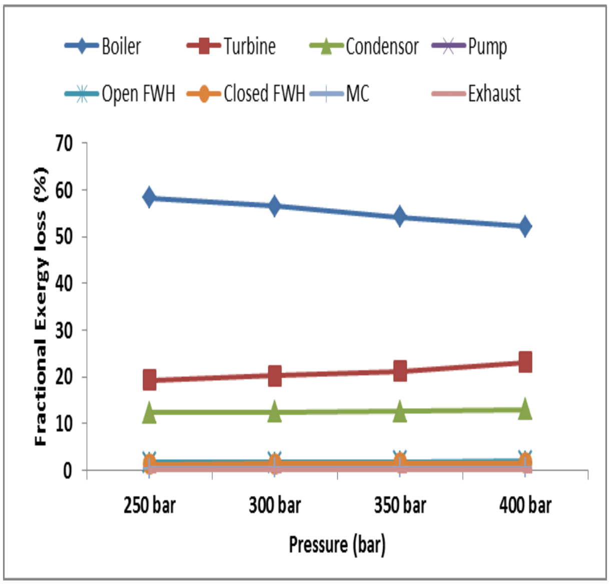

Fig. 8: Effect of turbine inlet pressure on fractional exergy loss of different components, T_{FGi}=1000^{o}C, T_{FGo}=100^{o}C, T=600^{o}C, P_c=0.1bar

Fig. 7 shows the variation of fractional exergy loss of the different components of the vapor power cycle as a function of turbine inlet temperature. It is observed that the fractional exergy loss of the boiler and the steam turbine increases with increase in turbine inlet temperature at a particular pressure whereas the fractional exergy loss of other components remains approximately constant. Fig. 8 shows the fractional exergy loss of the different components of the vapor power cycle as a function of boiler pressure. From the figure, it can be easily deduced that the fractional exergy loss of the boiler decreases with the increase in the boiler pressure at a fixed turbine inlet temperature whereas that of steam turbine increases with the increase of boiler pressure. The fractional exergy loss of other components of the cycle remains almost constant.

6.0 Conclusions:

The energy and exergy analyzes of the cycle has been performed for pressure range between 250 bar to 400 bar and temperature range 500^{o}C-800^{0}C and the results are shown in the figures 2 to 5. It is found that with the increase in boiler pressure and turbine inlet temperature, both the energy and exergy efficiency of the plant increases.

The energy efficiency of the cycle increases as a result of using regeneration, open FWH and closed FWH. From the exergy analysis it is found that the losses due to irreversibility were maximum in the boiler than in the turbine followed by the condenser. Further it was seen that the fractional irreversibility in the boiler increases with turbine inlet temperature whereas it decrease with increase in boiler pressure. In the present work a simple reheat-regenerative rankine cycle was considered for calculation of energy efficiency, exergy efficiency and the irreversible losses.

In future more complicated modeling of individual components can be considered. Therefore, the irreversibilities in the boiler, condenser and re-heater should be taken into consideration. The fractional exergy loss of the cycle has been analyzed and it is seen that the fractional exergy losses is highest in the case of boiler followed by turbine and condenser.

NOMENCLATURE

Symbol

T	Temperature	(K)
s	Entropy	(N)
P	Pressure	(Pa)
HS	Heat Supplied	(kJ/kg)
I	Irreversibility	(kW)
To	Absolute Temperature	(K)
ms	Mass of steam	(kg/s)
mg	Number of moles of the flue gas	
mcw	Mass of cooling water	(kg/s)
y, z	Mass fractions	
OFWH	Open feed water heater	
CFWH	Closed feed water heaters	
B	Boiler	
ST	Steam turbine	
C	Condenser	
P1	Pump 1	
P2	Pump 2	
P3	Pump 3	
RH	Reheater	
CW	Cooling Water	
MC	Mixing Chamber	
Fgi	Flue Gas in	
Fgo	Flue Gas out	
Σ	Sum	

Subscripts

boi	Boiler
tur	Turbine
con	Condenser
rh	Reheat
cw	Cooling water
wi	Water inlet
i	Inlet
o	Outlet
s	Isentropic

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