EXERGY ANALYSIS AND PARAMETRIC STUDY OF REHEAT REGENERATIVE ULTRA SUPERCRITICAL RANKINE CYCLE WITH FEED WATER HEATERS

I. Satyanarayana¹, A.V.S.S.K.S. Gupta², K.G. Rajulu³ and K. Susheela⁴

¹Department of Mechanical Engineering, JNTUH College of Engineering, Kukatpally, Hyderabad-500 085, India
²Department of Mechanical Engineering, JNTUH College of Engineering, Kukatpally, Hyderabad-500 085, India.
³Department of Mechanical Engineering, JNTUA College of Engineering Anantapur, Andhra Pradesh-515 002, India.
⁴Megha Institute of Engineering and Technology for Women. Ghatkesar, Ranga Reddy district, Affiliated to JNTUH University, Kukatpally, Hyderabad-500 085, India.
E-mail: iaaditya2002@gmail.com; sindigibilli@gmail.com

Received 17 February 2010, Accepted 25 May 2010

ABSTRACT
Exergy analysis and parametric analysis have been carried for supercritical Rankine cycle with reheat, open and closed feed water heaters of higher power generation for modern steam power plants at supercritical, ultra supercritical and advanced ultra supercritical temperatures and pressures. The pressure range between 200 to 425 bar and the temperature range between 500°C-750°C have been studied in this cycle. The variation of reheat pressure is 0.2 to 0.4 times the initial pressure of the turbine has analyzed. The effect of terminal temperature difference between 3 to 8 has studied. The variation in condenser pressure has been studied from 0.02 to 0.1 bar. The effects of the flue gas inlet and outlet temperature of the boiler has been analyzed from 80°C-300°C and 900°C-1400°C respectively. The energy and exergy efficiencies, irreversibility and fractional exergy losses of all the components of the cycle have been estimated. The energy and exergy analyses were performed for each component in the system and the results tabulated. The variations of irreversibility, efficiency and fractional exergy loss with several parameters were graphically investigated. The results show that a clearer definition of the real losses in the system is given by the exergy method of analysis. The results are tabulated and shown in the graphs.

Keywords: Energy, Exergy, Irreversibility, Ultra supercritical Rankine cycle

NOMENCLATURE
\[ \Sigma = \text{Sum} \]
\[ B = \text{Boiler} \]
\[ C = \text{Condenser} \]
\[ CFWH = \text{Closed feed water heater} \]
\[ CW = \text{Cooling water} \]
\[ EX_f = \text{Exergy in the flue gas at the entering the boiler} \]
\[ EX_r = \text{Exergy in the flue gas at the exiting from the boiler} \]
\[ FG = \text{Flue gas in} \]
\[ FGO = \text{Flue Gas out} \]
\[ H_A = \text{Enthalpy of flue gases entering the boiler} \]
\[ H_B = \text{Enthalpy of flue gases leaving the boiler} \]

HS = Heat supplied (kJ/kg)
I = Irreversibility (kW)
MC = Mixing chamber
mcw = Mass of cooling water (kg/s)
ng = Number of moles of the flue gas (kg-mol)
ms = Mass of steam (kg/sec)
OFWH = Open feed water heater
P = Pressure (bar)
P1 = Pump 1
P2 = Pump 2
P3 = Pump 3
P4 = Pump 4
RH = Reheater
s = Entropy (kJ/kg.K)
ST = Steam turbine
T = Temperature (K)
To = Absolute temperature (K)
y, z = Mass fractions

Subscripts:
boi = Boiler
con = Condenser
tur = Turbine
rh = Reheat
cw = Cooling water
wi = Water inlet
i = Inlet
o = State of surroundings
s = Isentropic

1. INTRODUCTION
Today many modern steam power plants operate at supercritical pressures (P > 220.6 bar) and have thermal efficiencies higher than plants operating with sub-critical pressures. 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. The efficiency of the supercritical Rankine cycle can be improved by varying cycle parameters such as turbine inlet pressure, inlet temperature, reheat pressure, reheat temperature, extraction pressure and the condenser pressure with...
respect to the optimum value. The efficiency of the Rankine cycle is determined by two major parameters: the pressure and temperature of the steam at the inlet of the turbine, and the condenser pressure. Increasing the efficiency of the Rankine cycle by increasing the boiler pressure is often facing the problem that the moisture content of the steam at the turbine outlet increases. But materials sustainability at higher temperatures and pressures needs lot of research for developing the ultra-supercritical and advanced ultra-supercritical ranges. Exergy is the maximum amount of obtainable work from a steam in reference to the environment. The exergy method of analysis has two advantages over the energy method for design and performance analysis of energy-related systems. First, it provides a more accurate measurement of the actual inefficiencies in the system and the true location of these inefficiencies. It accomplishes this for any system, whether simple or complex. Exergy analyses also provide a true measure of the system efficiency for complex combined cycle or open systems, where the energy method gives an erroneous efficiency value.

Steam is the most common working fluid used in vapor power cycles because of its many desirable characteristics, such as low cost, availability, and high enthalpy of vaporization. Steam reheating is an important feature in steam-power plants. The main objective of reheating is to increase the power output and, under certain conditions, the thermal efficiency of the plant, thus improving plant performance. In a steam power plant, heat water heaters allow the feed water to be brought up to the saturation temperature very gradually. This minimizes the inevitable irreversibilities associated with heat transfer to the working fluid.

Energy and exergy analyses were performed for each component in the system and the results tabulated. The variations of availability difference, irreversibility and efficiency with several parameters were graphically investigated. The results show that a clearer definition of the real losses in the system is given by the exergy method of analysis (Ibrahim, 1997). Thermodynamic optimization of reheating regenerative thermal-power plants in subcritical ranges have carried out for below supercritical pressure range (Habib et al., 1999). They analyzed the second-law efficiency of the steam generator; turbine cycle and plant are evaluated and optimized. The irreversibilities in the different components of the steam generator turbine cycle sections are evaluated and discussed in this paper. Further, thermodynamic analysis of a Rankine cycle reheating steam power plant is conducted, in terms of the first law of thermodynamic analysis (i.e. energy analysis) and the second law analysis (i.e. exergy analysis), using a spreadsheet calculation technique (Dincer and Muslim, 2001; Hasanuzzaman et al, 2011). The temperature and pressure values are selected in the range between 400 and 590 degrees C, and 100 and 150 bar, being consistent with the actual values. A second law analysis showed that by using a binary fluid, the Kalina cycle reduced irreversibility in the boiler, resulting in improved efficiency of the cycle by Nag and Gupta. The exergy output depends on the degree of irreversibility of the cycle (Nag and Gupta, 1998). The details of mass, energy, exergy balances for a reference steam power plant and investigated the effect of the most important process parameters on the exergetic efficiency (Kotas, 1995). Generalized Thermodynamic Analysis of Steam Power Cycle with ‘n’ number of feed water heaters was analysed in subcritical range thermodynamic analysis of Supercritical Rankine cycle (Srinivas et al., 2007). They concluded that the energy and exergy efficiencies are increased with increase in steam turbine inlet pressure and inlet temperature. Thermodynamic analysis of Supercritical Rankine were carried out and found that the energy and exergy efficiency of the supercritical Rankine increases with an increase if boiler flue gas inlet temperature and inlet pressure (Satyanarayana et al., 2004). A Thermodynamics property of steam program has been developed by using ISI steam tables in this analysis.

In the present work, an attempt has been made to analyze the steam power cycle with reheat, open and closed feed water heaters from an exergy point of view. The schematic of the supercritical Rankine cycle with reheat, open and closed feed water heaters and the T-s diagram of the cycle are given in Figure 1.

2.0 FORMULATION

2.1 Thermodynamic properties of steam:
This whole region is divided into six sub regions, numbered 1 to 6. The total region has been divided into six regions according to the range of pressures and temperatures. The related equations for the calculation of properties in different regions are presented in ISI steam tables. A computer code has been developed for the calculation of steam tables in different regions using ISI steam tables. The calculated properties from the program are tested with standard data available in steam tables. The accuracy of the developed steam values with the standard data is 99.5 percent.

2.2 Assumptions used in the present analysis
1. Capacity of the power plant = 1000 MW
2. Reheat pressure = 0.2-0.4 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 Tgi=900°C -1400°C and leaving is Tgo=80°C -300°C.
6. The pinch point temperature difference in the condenser is 6°C.
7. Condenser pressure Pc = 0.02 - 0.1 bar
8. Terminal Temperature Difference (TTD) in feed water heater = 3°C - 8°C
9. Cooling water temperature inlet to the condenser Tw=25°C
10. No heat losses and no pressure losses

\[ \sum m_i h_i = \sum m_i h_e \]  

Applying equation (9) to these devices gives,

Open FWH (q = 0, w = 0)

\[ z = (1-y)(h_8 - h_f) + y(h_{10} - h_8) \] (10)

Closed FWH (q = 0, w = 0)

\[ y(h_2 - h_{11}) = (1-y)(h_{10} - h_8) \] (11)

Mixing chamber (q = 0, w = 0)

\[ h_{12} = (1-y)h_{10} + yh_{12} \] (14)

The turbines are not isentropic. Then, the conservation of energy relation can be expressed as follows:

\[ W_{turbine} = (h_1 - h_2) + (1-y)(h_3 - h_d) + (1-y-z)(h_4 - h_3) \] (15)

\[ W_{pump} = (1-y-z) * W_{pump1} + (1-y) * W_{pump2} + y * W_{pump3} \] (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:

\[ H.S = (h_1 - h_{11}) + (1-y)*h_3 - h_2 \] (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,

\[ \text{Energy Efficiency} = (W_{turbine} - W_{pump}) / H.S \] (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. The coal used is anthracite of the chemical composition of the power plant has taken form Kotas (1995) are as:

<table>
<thead>
<tr>
<th></th>
<th>CO2</th>
<th>H2O</th>
<th>N2</th>
<th>O2</th>
<th>SO2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>x_k</td>
<td>6.51</td>
<td>1.634</td>
<td>35.32</td>
<td>9.324</td>
<td>0.047</td>
<td>57.735</td>
</tr>
</tbody>
</table>

\[ n_k[\text{kmol/100kg fuel}]: 0.1234 \quad 0.0310 \quad 0.6679 \quad 0.1768 \quad 0.0009 \quad 1.000 \]

\[ H_A = (\theta_A - \theta^0) \sum n_k \tau_{ph} \] (19)

\[ H_B = (\theta_B - \theta^0) \sum n_k \tau_{ph} \] (20)

\[ E_A = (\theta_A - \theta^0) \sum n_k c_v \] (21)

\[ E_B = (\theta_B - \theta^0) \sum n_k c_v \] (22)

Where mean isobaric heat capacity for evaluating enthalpy changes is

\[ \tau_x^p = \frac{\overline{\tau_x^p}}{T - T_0} \] (23)

and mean molar isobaric exergy capacity for evaluating changes in physical exergy is

\[ c_v^e = \frac{\overline{c_v^e}}{T - T_0} \] (24)

mass of stream generated for the given flow rate of flue gases obtained from the energy balance.

The mass of the steam is calculated from the capacity of the

Figure 1 Schematic diagram and T-s diagram of the cycle

2.3 Energy Analysis of the cycle

The reheat Pressure \( P_{1h} = (0.2 - 0.4) P_1 \) (1)

Location of the feed water heater is calculated from the Nag (2001) is

\[ T_{x} = (T_B - T_C)/(n+1) \] (2)

\[ T_{x} = T_C - T_0 \] (3)

All the nodal points (h,s) in the above T-s diagram has 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 Figure 1,

\[ T_{10} = T_{11} - \text{TTD} \] (4)

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

\[ q - w = h_0 - h_i \] (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:

\[ W_{pump1} = h_2 - h_6 \] (6)

\[ W_{pump2} = h_9 - h_8 \] (7)

\[ W_{pump3} = h_{11} - h_{12} \] (8)

The conservation of energy and mass equations reduce to
m(Wnet) = 1000 MW
where \( W_{net} = (W_{turbine} - W_{pump}) \)
Thus,
\[
m = \frac{1000x1000 \text{ kW}}{W_{turbine} - W_{pump}}
\]
Energy balance equation for obtaining the number of flue gases (\( m_{3} \)) is,
\[
m_{1}(h_{1} - h_{13}) - (y)(h_{1} - h_{3}) = m_{2}(H_{A} - H_{B})
\]
Thus
\[
m_{2}g = ms* \left( (h_{1} - h_{13}) - (y)(h_{3} - h_{2}) \right) \left( Ha - Hb \right)
\]
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 \( \theta_{1} = 1000 \degree \text{C} \) to \( 1400 \degree \text{C} \) and \( \theta_{0} = 25 \degree \text{C} \).
The composition of the flue gas has been calculated and enthalphy 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 \( E_{x} = E_{x,1} \), Entalphy of the flue gas entering \( H_{A} \), Exergy in the flue gas at the exit the boiler is \( E_{x} = E_{x,2} \) and Entalphy of the flue gas at exit of the boiler \( H_{B} \) are calculated.
Availability or Gibbs function of steam at state point 1
\[
G_{1} = E_{y} = m_{1}(h_{1} - T_{s1})
\]
Availability or Gibbs function of steam at state point 13
\[
G_{13} = E_{y,13} = m_{1}(h_{1} - T_{s13})
\]
Irreversibility in the boiler is
\[
I_{boi} = (m_{1}(E_{x,1} - E_{y}) - (m_{1}(h_{1} - h_{13}) - (y)(h_{3} - h_{2})) - T_{d}(s_{1} - s_{13}) - (1 - y)(s_{2} - s_{13}))
\]
2.4.2 Steam Turbine
The irreversibility rate in the steam turbine given by Gouy-Stodola equation is
\[
\text{Itur} = msT_{d}(s_{2} - s_{1}) + ((y)(s_{3} - s_{2}) + ((1 - y)(s_{4} - s_{3}) + ((1 - y)(s_{5} - s_{4})))
\]
2.4.3 Condenser
Mass of cooling water circulated to condense \( ms \) kg of steam is obtained from the energy balance is
\[
m_{1}(1 - y - z)(h_{3} - h_{13}) = m_{w}\ C_{pw}(T_{Wt} - T_{Wc})
\]
where \( C_{pw} \) = 4.1868 \( \text{kJ/kg.K} \)
\[
m_{w} = m_{1}(1 - y - z)(h_{3} - h_{13}) / C_{pw}(T_{Wt} - T_{Wc})
\]
Irreversibility in the condenser,
\[
I_{con} = T_{d}m_{w}C_{pw}ln(T_{Wt}/T_{Wc}) - m_{1}(1 - y - z)(s_{7} - s_{8})
\]
2.4.4 Pump1:  Irreversibility in pump1
\[
I_{p1} = msT_{d}(1 - y - z)(s_{7} - s_{8})
\]
2.4.5 Pump2:  Irreversibility in pump2
\[
I_{p2} = msT_{d}(1 - y)(s_{6} - s_{8})
\]
2.4.6 Pump3:  Irreversibility in pump3
\[
I_{p3} = msT_{d}(y)(s_{72} - s_{71})
\]
Sum of irreversibilities of all three pumps is
\[
I_{pump} = I_{p1} + I_{p2} + I_{p3}
\]
2.4.6 Open FWH:
Irreversibility in Open FWH,
\[
I_{ofwh} = msT_{d}(1 - y)(s_{3} - z)(s_{7} - (1 - y - z)s_{7})
\]
2.4.7 Closed FWH:
Irreversibility in Closed FWH,
\[
I_{cfwh} = msT_{d}(1 - y)(s_{13} - s_{2}) - (1 - y)(s_{10} - s_{9})
\]
2.4.8 Mixing chamber:  Irreversibility in mixing chamber,
\[
I_{mc} = msT_{d}(s_{13} - l - y)(s_{10} - y)(s_{12})
\]
2.4.9 Exhaust
Irreversibility or exergy loss through the exhaust
\[
I_{exhaust} = E_{x,2}
\]
Total Irreversibility is
\[
\Sigma I = I_{boi} + I_{Itur} + I_{con} + I_{pump} + I_{ofwh} + I_{cfwh} + I_{mc} + I_{exhaust}
\]
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.
\[
\text{Exergy efficiency} = \frac{(E_{x} - \Sigma I) / E_{x} \times 100}{100}
\]
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,
\[
\sum_{I} \frac{I_{boi}}{I} * 100
\]
2. Fractional exergy loss in the turbine is,
\[
\sum_{I} \frac{I_{tur}}{I} * 100
\]
3. Fractional exergy loss in the condenser is,
\[
\sum_{I} \frac{I_{con}}{I} * 100
\]
Fractional exergy loss in the open fwh is,
\[
\sum_{I} \frac{I_{ofwh}}{I} * 100
\]
4. Fractional exergy loss in the closed fwh is,
\[
\sum_{I} \frac{I_{cfwh}}{I} * 100
\]
5. Fractional exergy loss in the mixing chamber is,
\[
\sum_{I} \frac{I_{mc}}{I} * 100
\]
6. Fractional exergy loss in the Pump is,
\[
\sum_{I} \frac{I_{pump}}{I} * 100
\]
7. Fractional exergy loss in the exhaust is, 
\[ \sum I_{\text{exhaust}} = 100 \] (50)

Table 1 Thermodynamic properties of steam at 
P=325bar, T=650°C, Pc=0.1bar, Prh=0.2P

<table>
<thead>
<tr>
<th>State</th>
<th>(h) (kJ/kg)</th>
<th>(s) (kJ/kg-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3575.601074</td>
<td>6.349768</td>
</tr>
<tr>
<td>2s</td>
<td>3123.952637</td>
<td>6.349768</td>
</tr>
<tr>
<td>2</td>
<td>3123.952637</td>
<td>6.439443</td>
</tr>
<tr>
<td>3</td>
<td>3769.962402</td>
<td>7.257148</td>
</tr>
<tr>
<td>4s</td>
<td>3085.932129</td>
<td>7.430174</td>
</tr>
<tr>
<td>4</td>
<td>3085.932129</td>
<td>7.430174</td>
</tr>
<tr>
<td>5s</td>
<td>2354.779541</td>
<td>7.430174</td>
</tr>
<tr>
<td>5</td>
<td>3123.952637</td>
<td>7.430174</td>
</tr>
<tr>
<td>6</td>
<td>191.800003</td>
<td>0.649000</td>
</tr>
<tr>
<td>7s</td>
<td>191.800003</td>
<td>0.649000</td>
</tr>
<tr>
<td>7</td>
<td>191.800003</td>
<td>0.657012</td>
</tr>
<tr>
<td>8</td>
<td>702.214172</td>
<td>2.003481</td>
</tr>
<tr>
<td>9s</td>
<td>703.003296</td>
<td>2.003481</td>
</tr>
<tr>
<td>9</td>
<td>703.142578</td>
<td>2.030327</td>
</tr>
<tr>
<td>10</td>
<td>1270.617798</td>
<td>3.127623</td>
</tr>
<tr>
<td>11</td>
<td>1270.617798</td>
<td>3.127623</td>
</tr>
<tr>
<td>12s</td>
<td>1305.039551</td>
<td>3.127623</td>
</tr>
<tr>
<td>12</td>
<td>1311.140141</td>
<td>3.163591</td>
</tr>
<tr>
<td>13</td>
<td>1280.110718</td>
<td>3.136055</td>
</tr>
</tbody>
</table>

Table 2 Irreversibility and FEL of all components 
at FGo=1000°C, FGi =100°C

<table>
<thead>
<tr>
<th>Component</th>
<th>Irreversibility(MW)</th>
<th>Fractional Exergy Loss</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boiler</td>
<td>180.38866</td>
<td>49.45</td>
</tr>
<tr>
<td>Turbine</td>
<td>87.85409</td>
<td>24.09</td>
</tr>
<tr>
<td>Condenser</td>
<td>61.09888</td>
<td>16.75</td>
</tr>
<tr>
<td>Pump</td>
<td>7.11269</td>
<td>1.95</td>
</tr>
<tr>
<td>Open FWH</td>
<td>7.63436</td>
<td>2.09</td>
</tr>
<tr>
<td>Closed FWH</td>
<td>5.87245</td>
<td>1.61</td>
</tr>
<tr>
<td>MC</td>
<td>1.76191</td>
<td>0.48</td>
</tr>
<tr>
<td>Exhaust</td>
<td>13.07568</td>
<td>3.58</td>
</tr>
<tr>
<td>Sum</td>
<td>364.79872</td>
<td>100.00</td>
</tr>
</tbody>
</table>

4.0 RESULTS AND DISCUSSION
The equations used in the first and second law analyses are applied to the components making up the cycle, and the results are shown in Tables 1 and 2. A Thermodynamic analysis has carried out to reheat, open and closed feedwater heaters. The net output of the cycle has been energy and exergy efficiencies. The variations of energy efficiency at different turbine inlet temperatures, 500°C-800°C at pressure ranges between 170-425bar has been showing in the Figure 2. The energy efficiency has been increased continuously in increasing the turbine inlet pressure. It has been observed from the Figure 2, the cycle efficiency at advanced ultra supercritical pressure (425 bar) is 5.97 times, ultra supercritical pressure (325 bar) is 3.97 times, supercritical pressure (225 bar) is 0.91 times higher than the sub critical pressure(200 bar) at a temperature of 700°C.
Figure 4 Variation of Exergy efficiency at different Turbine inlet Temperature values with increase in Pressure at $P_c=0.1$bar

Figure 5 Variation of Exergy efficiency at different Turbine inlet Pressure values with increase in Temperature at $P_c=0.1$bar

Figure 5 shows that variation of Exergy efficiency at different Turbine inlet Pressure values with increase in Temperature at a condenser pressure $P_c=0.1$bar. The exergy efficiency increases with increase in pressure at various temperatures. The percentage in exergy efficiency increases between 1.61 – 7.83 times from 200bar to 425bar.

Figure 6 gives effect of turbine inlet temperature on fractional exergy loss of different components in the cycle. It is observed that the 45-56% fractional exergy losses(FEL) have been carried in the boiler, 21-29% fractional exergy losses in the turbine, 13.8-15.7% FEL in the condenser, 1.62-2.1% FEL in pump, 1.62-2.79% FEL in open feed water heater, 1.25-2.14 FEL in the closed feed water heater, 0.37 – 0.64 % in mixing chamber and 2.79-3.73 % FEL in the exhaust.

Figure 6 Effect of turbine inlet temperature on fractional exergy loss of different components.

Figure 7 Effect of turbine inlet pressure on fractional exergy loss of different components

Figure 7 gives effect of turbine inlet pressure on fractional exergy loss of different components in the cycle. It is observed that the 48-63% fractional exergy losses(FEL) have been carried in the boiler, 17-25.72% fractional exergy losses in the turbine, 12.9-15.35% FEL in the condenser, 1.4-2.01% FEL in pump, 1.11-2.46% FEL in open feed water heater, 0.86-1.89 FEL in the closed feed water heater, 0.86 – 0.57 % in mixing chamber and 2.94-3.47 % FEL in the exhaust. It is observed that the fractional exergy loss of the boiler increases with increase in temperature at a particular pressure where as it reduces the increase in temperature in the turbine and fractional exergy loss of the condenser is remains constant while increase in the temperature. Figure 8 and 9 represents the total exergy losses in the cycle. It shows that the when temperature increases the
total exergy loss of the cycle decreases and also decreases while increase in the pressure.

Figure 9 Effect of total exergy loss on different turbine inlet pressure

Effect of Reheat Pressure Ratio: Figures 10 -12 shows that the effect of reheat pressure ratio between 0.2 to 0.4, on cycle efficiency, exergy efficiency, Irreversibilities of all components and Fractional exergy losses of all the components have been studied and results are analyzed in the graphs.

Figure 10 Cycle efficiency, heat input as function of reheat pressure ratio

Figure 11 Steam mass, Flue gas mass as function of reheat pressure ratio

Figure 12 Exergy efficiency, Total Irreversibility as function of reheat pressure ratio

Figure 13 Fractional exergy loss as function of reheat pressure ratio

Figure 10 shows Cycle efficiency, heat input as function of reheat pressure ratio. It is observed that the cycle efficiency has been increasing in increase of reheat pressure ratio. It is also observed that the cycle efficiency increases 1.33 times at 0.25P1, 2.46 times 0.3P1, 3.51 times 0.35P1 and 4.22 times 0.4P1 increases when compare to pressure ratio 0.2. On the other side of the axis as heat input against the reheat pressure ratio, this is a decrease when increasing the reheat pressure ratio.

Effect of Reheat Pressure Ratio: Figures 10 -12 shows that the effect of reheat pressure ratio between 0.2 to 0.4, on cycle efficiency, exergy efficiency, Irreversibilities of all components and Fractional exergy losses of all the components have been studied and results are analyzed in the graphs.

Figure 10 shows Cycle efficiency, heat input as function of reheat pressure ratio. It is observed that the cycle efficiency has been increasing in increase of reheat pressure ratio. It is also observed that the cycle efficiency increases 1.33 times at 0.25P1, 2.46 times 0.3P1, 3.51 times 0.35P1 and 4.22 times 0.4P1 increases when compare to pressure ratio 0.2. On the other side of the axis as heat input against the reheat pressure ratio, this is a decrease when increasing the reheat pressure ratio.
in the reheat pressure ratio and other components are shown in figure.

Effect of Terminal Temperature Difference (TTD):
Figure 14 shows that the efficiency and total irreversibility as a function of Terminal Temperature Difference. Exergy efficiency increases with an increase in TTD and then it decreases.

Maximum exergy efficiency has occurred at TTD is 5 i.e., 68.35%. The percentage of increase in exergy efficiency 0.33 times as compare to the TTD is 4 and 2.86% increases at TTD is 5. It is also observed that the efficiency decreases slightly after TTD increases. This graph also shows that the total irreversibility on the second y-axis line against TTD. It is observed that the total irreversibility increases with increase in temperature up to 1300°C and then total irreversibility decreases. Figure 17 shows that the irreversibility of all components as function of flue gas outlet temperature. Irreversibility is more in the boiler which is about 135 MW to 252 MW. It has been observed that the irreversibility increases rapidly from 900°C to 1300°C and it has reached the maximum value of irreversibility and then slightly decreases.

Irreversibility in the turbine is decreases slowly while increasing the flue gas temperature up to 1000°C and remains constant. Irreversibility in the condenser is also decreasing from 900°C to 1000°C and increases at 1100°C and remains steady in irreversibility when increasing the flue gas outlet temperature. Irreversibility in the open feed water heater is 1.83MW/900°C, 1.04MW/1000°C, 1.82MW/1100°C, 1.7MW/1200°C, 1.61MW/1300°C and 1.68MW/1400°C and also the irreversibility in the closed feed water heater is 1.41MW/900°C, 1.57MW/1000°C, 1.43MW/1100°C, 1.3MW/1200°C, 1.24MW/1300°C and 1.29 MW/1400°C. Irreversibility in the mixing chamber varies between 0.32 to 0.47 MW and exhaust decreases with an increase of flue gas outlet temperature. Figure 18 shows that the fractional exergy loss as function of flue gas outlet temperature of all components in the cycle. FEL in the boiler is 53.34%/52.99%/56.19%/55.83%/55.49% /55.15% at TTD increases from 3 to 8. FEL of the turbine is decreases slightly with increase in TTD where as in the condenser FEL is increases with increase in TTD.

Irreversibility in the turbine is decreases slowly while increasing the flue gas temperature up to 1000°C and remains constant. Irreversibility in the condenser is also decreasing from 900°C to 1000°C and increases at 1100°C and remains steady in irreversibility when increasing the flue gas outlet temperature. Irreversibility in the open feed water heater is 1.83MW/900°C, 1.04MW/1000°C, 1.82MW/1100°C, 1.7MW/1200°C, 1.61MW/1300°C and 1.68MW/1400°C and also the irreversibility in the closed feed water heater is 1.41MW/900°C, 1.57MW/1000°C, 1.43MW/1100°C, 1.3MW/1200°C, 1.24MW/1300°C and 1.29 MW/1400°C. Irreversibility in the mixing chamber varies between 0.32 to 0.47 MW and exhaust decreases with an increase of flue gas outlet temperature. Figure 18 shows that the fractional exergy loss as function of flue gas outlet temperature of all components in the cycle. FEL in the boiler is 53.34%/52.99%/56.19%/55.83%/55.49% /55.15% at TTD increases from 3 to 8. FEL of the turbine is decreases slightly with increase in TTD where as in the condenser FEL is increases with increase in TTD.

Effect of Flue gas Temperature outlet in the boiler:
Figure 16 shows that the exergy efficiency, Total Irreversibility as function of Flue gas outlet temperature at a particular flue gas inlet temperature and a fixed turbine inlet pressure and temperature and also at a constant turbine exit pressure. The flue gas outlet temperature has been analyzed and studied from 900°C to 1400°C and it is observed that the exergy efficiency has increased with an increase of the flue gas temperature of 1300°C. At 900°C the exergy efficiency is 61.18% and at 1400°C, the exergy efficiency is about 75.75% that means the increase in efficiency from 900°C to 1400°C is 23.81%. It is also shown on the graph; the total irreversibility is a function of flue gas outlet temperatures. Total irreversibility has increased with increase in temperature up to 1300°C and then total irreversibility decreases. Figure 17 shows that the irreversibility of all components as function of flue gas outlet temperature. Irreversibility is more in the boiler which is about 135 MW to 252 MW. It has been observed that the irreversibility increases rapidly from 900°C to 1300°C and it has reached the maximum value of irreversibility and then slightly decreases.
Effect of Flue gas Temperature inlet in the boiler:

Figure 20 shows that the irreversibility of all components as function of flue gas inlet temperature of all components. Major irreversibility occurs in the boiler and it increases with an increase of flue gas inlet temperature and irreversibility in the turbine remainssame throughout the increase of flue gas inlet temperature. Irreversibility in the exhaust has been increases with an increase in flue gas inlet temperature, 7.23MW/80°C, 13.07MW/100°C, 33.82MW/150°C, 62.09MW/200°C, 96.56MW/250°C, 136.31MW/300°C.

Figure 20 Irreversibility of all components as function of flue gas inlet temperature

5.0 CONCLUSION

The energy and exergy analyzes of the cycle has been performed for pressure range between 170 bar to 425 bar and temperature range are 500°C-800°C and are shown in the Figures 2 to 5. Effect of reheat pressure ratio has been studied from 0.2 to 0.4. effect of terminal temperature difference have studied from 3 to 8, effect of flue gas temperature outlet in the boiler has studied from 900°C-1400°C and effect of flue gas temperature inlet in the boiler have been analyzed from 80°C-300°C. The energy analysis and exergy analysis has analyzed in this paper for single reheat, open and closed feed water heaters of Rankine cycle in sub critical, supercritical ultra supercritical and advanced ultra supercritical range. Both cycle efficiency and exergy efficiency are increases with increase in pressure and temperature. It is observed that the cycle efficiency is increases with increase in temperature than the pressure increase. The irreversibilities and fractional exergy loss of all the components have been studied and results are discussed. The largest irreversibilities in the cycle appear during the heat addition, heat rejection and regeneration processes, respectively. Therefore, the irreversibilities in the boiler, condenser and re-heater should be taken into consideration. The fractional exergy loss and total exergy loss of the cycle has analyzed. Total exergy loss decreases with increases in temperature and pressure. Exergy efficiency increases with an increase in terminal temperature difference and then it is decreases. It is observed that the total irreversibility increases with increase in TTD. Total irreversibility has increased with increase in temperature up to 1300°C and then total
irreversibility decreases. Exergy efficiency is increases with increase in flue gas temperature outlet of the boiler. It is observed that the total irreversibility increases with an increase in flue gas inlet temperature.

REFERENCES