# Energy and Exergy Analysis of Supercritical Rankine Cycle

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**Abstract:** The present paper explains the exergy and energy analysis of supercritical Rankine Cycle. The plant consists of one boiler feed pump, one supercritical boiler, one steam turbine and a condenser. The energy and exergy analysis has been carriedfrom which energy efficiency, exergy efficiency and the irreversibility results obtained from the simulation has been calculated and plotted on the graph. The supercritical pressure is varied from 20MPa to 50MPa in a step of 5 MPa and the temperature has been changed from 550 to 800°C. The parametric study reveals the effect of these parameters on cycle energy and exergy efficiency. The analysis shows that cycle energy and exergy efficiency increases with increase in turbine inlet temperature and pressure and while calculating the fractional exergy losses and irreversibility it is found that maximum irreversibility occur in boiler followed by the turbine then in condenser and in the pump.

**Keywords:** Supercritical power plant; Irreversibility; exergy efficiency; exergy analysis

## 1. Introduction

Energy in general, and electricity in particular, plays a vital role in improving the standard of life and also power sector plays a very vital role in overall economic growth of any country.Exergy is the maximum theoretical useful work attainable from an energy carrier under theconditions imposed by an environment at given pressure P0 and temperature T0, and with given amounts ofchemical elements. The purpose of an exergy analysis is generally to identify the location, the source, and the magnitude of true thermodynamic inefficiencies in power plants. Moreover, the results from an exergy analysis constitute a unique base for exergoeconomics, an exergy-aided cost reduction method (Tsatsaronis (1998)). The information provided by an exergy analysis can be thus used as a guide for reducing the thermodynamic inefficiencies of power plants and for improving their performance. Indeed, less destruction and losses associated with a powerplant correspond to a higher yield of the powerplant.

The key variable resulting from an exergy analysis is the exergetic efficiency. It shows the percentage of the fuel exergy to be found in the product exergy of a power plant or a component. Therefore, the exergetic efficiency gives an unambiguous criterion for judging theperformance of a power plant and, more importantly, of its components, from the thermodynamic viewpoint.

## 2. Literature Review

Most recently A.K.Tiwari et al. (2013) done the theoretical exergy analysis of the combineBrayton/Rankine power cycle of NTPC Dadri India is presented. The results pinpoint that morethe exergy loses occurred in the gas turbine combustion chamber reaching 35% of the total exergyloses, while the exergy loses in other plant components are between 7% and 21% of the totalexergy loses at 1400 C turbine inlet temperature and pressure ratio 10 [1].

M. Pandey et al.(2013) in his paper describes the energy and exergy analysis of a reheat regenerative vapour power cycle [6] whereas Yongping Yang et al. (2013) conducted both conventional and advanced exergy analyses to a large-scale ultra-supercriticalcoal-fired power plant. The objectives of the conventional one are to compare theexergetic performances of different components, to identify and quantify the sites with thelargest exergy destruction and losses, and to find the fuel-savings potential by improving eachcomponent in isolation [13].V. Siva Reddy et al. (2013) perform the comparative energetic and exergetic analysis of coalfired supercritical thermal power plant (CSCTPT) and solar concentrator aided coal fired supercritical thermal power plant (SACSCTPT) [12].

K. Ravi Kumar et al. (2012) perform energy–exergy–environmental–economic (4-E) analyses of stand-alone line-focusing concentrating solar power plants for different plant capacities ranging from 1 to 50 MWe [5]. S.C. Kaushik et al. (2011) deals with the comparison of energy and exergy analyses of thermal power plants stimulated by coal and gas. This article provides a detailed review of different studies on thermal power plants over the years [11].EbrahimHajidavalloo et al. (2011) studied the Energy and exergy efficiencies of a supercritical power plant. The effect of ambient weather condition was considered on the condenser pressure. It was shown that high ambient temperature has more adverse effect on the exergy efficiency than the energy efficiency. As ambient temperature increases, the exergy efficiency of the boiler, condenser, heaters and feed water pump decrease, while the exergy efficiency of the turbine improves slightly [3]. Mehmet Kanoglu (2008) done the exergy analysis of a

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binary geothermal power plant and exergetic efficiencies of major plant components are determined in an attempt to assess their individual performances [9].

Mohammad Ameri et al. (2008) perform the energy, exergy and exergoeconomic analysis for the Hamedan steam power plant. The results show that energy losses have mainly occurred in the condenser where 306.9 MW is lost to the environment while only 67.63 MW has been lost fromthe boiler. Nevertheless, the irreversibility rate of the boiler is higher than the irreversibility rates of the other components [10]. Marc.A.Rosen (2003) reported of energy- and exergy-based comparisons of coalfired and nuclear electrical generating stations. Overall energy and exergy efficiencies, respectively, are 37% and 36% for the coal-fired process, and 30% and 30% for the nuclear process [8].ZhaolinGu (2001) studied a supercritical power cycle with a regenerative process to reach the maximum thermal efficiency by the choice of an appropriate working fluid and optimization of the cyclic state parameters, especially the condensing temperature or pressure [14]. E Bilgen (2000) presents exergetic and engineering analyses as well as a simulation of gas turbine-based cogeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine [2].H.Jin et al. (1997) analysed a supercritical steam plant and a gas-steam turbine combined cycle by using a methodology of graphical exergy analysis (EUDs) [4].

## 3. Thermodynamic Analysis

### 3.1. System Configuration

The schematic of the reheat regenerative supercritical steam power cycle is shown in Fig. 1. The system consists of the supercritical boiler, steam turbine, the condenser and a pump and Fig. 2 shows the corresponding T-s diagram of the system layout.

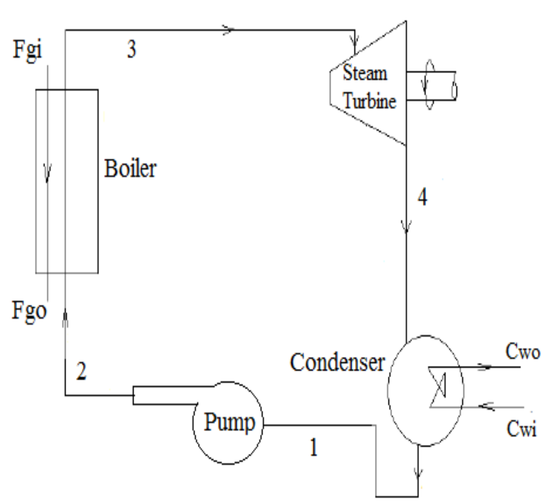


**Fig.1. Schematic diagram of system Layout**

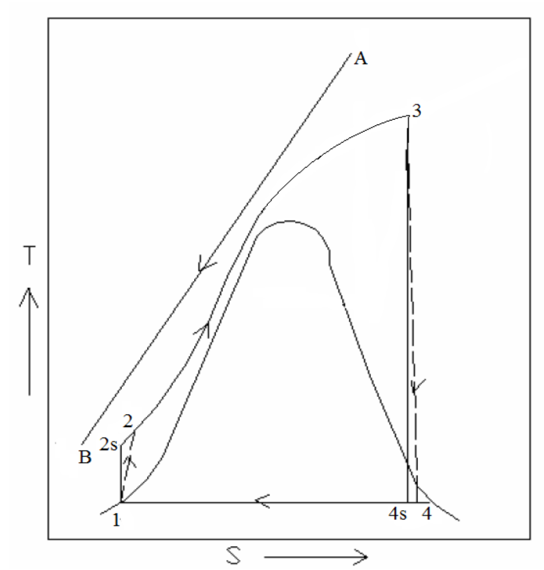


**Fig.2. T-s diagram of the cycle**

In the corresponding T-s diagram, the solid lines represent the ideal processes in the boiler, steam turbine condenser and pumps while the dotted lines represent the non-ideal processes in the steam turbine and pumps.

### 3.2. Assumptions used in this Analysis:

1. Capacity of the power plant = 500MW
2. The isentropic efficiency of the steam turbine is 85%.
3. The pump efficiency is assumed to be85%
4. Cooling water temperature inlet to the condenser is Twi = 25°C
5. Condenser pressure Pc= 10kPa
6. No heat losses and no pressure losses.

### 3.3. Range of variable parameters

1. The important variables which affect significantly the performance of theRankine cycle (energy and exergy efficiency) have been identified andlisted below.
2. Steam turbine Inlet Temperature = 550°C to 800°C
3. Steam turbine Inlet Pressure = 20MPa to 50MPa
4. Temperature of flue gas at the entry of boiler = 950°C
5. Temperature of flue gas at the entry of boiler = 400°C

### 3.4. Energy Analysis

It may be recalled that, the energy efficiency of the cycle is defined as the ratio of network done to the heat supplied in the boiler of the steam power cycle and is given by,

$$\eta = \frac{W_{net}}{\text{Heat Supplied}} \qquad (1)$$

Where Wnet= Wturbine- WpumpkJ/kg

The work done by the turbine per kg of steam supplied,

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$$W_{turbine} = (h_3 - h_4) kJ/kg \quad (2)$$

Boiler feed pump required work per kg of steam supplied,

$$W_{pump} = (h_2 - h_1) kJ/kg \quad (3)$$

Heat supplied to water/steam in the boiler per kg of steam produced is

$$H.S = (h_1 - h_4) kJ/kg \quad (4)$$

h1 = enthalpy at entry of pump or exit of condenser.
h2 = enthalpy at entry of boiler or exit of pump.
h3 = enthalpy at entry of turbine or exit of boiler.
h4 = enthalpy at entry of condenser or exit of pump.

### 3.5. Exergy Analysis

The method of exergy analysis aims at the quantitative evaluation of the exergy losses (irreversibilities) associated with a system. Hence, it is required to calculate the irreversibility in all the components of the power cycle for the estimation of exergy efficiency. The irreversibility or exergy loss in each of the components is calculatedfor the specified dead state (P0, T0).

**Boiler:**
The mass flow rate of steam (ms) required to be generated in the boiler to produce an output of 500MW power can be obtained from the energy balance as given below:

$$m_s = \frac{500 \times 1000 \text{ kW}}{W_{net}} \text{ kg/s} \quad (5)$$

Using this, the mass of the flue gas (mg) required to obtain the required mass flow rate of steam can be found by the energy balance equation i.e.,
Heat gained by the steam = Heat lost by the flue gas
$m_s(h_1 - h_4) = m_g(h_A - h_B)$

$$m_g = \frac{m_s(h_1 - h_4)}{(h_A - h_B)} \quad (6)$$

Exergy or Availability at state point 1

$$E_1 = m_s(h_1 - T_0 s_1) kW \quad (7)$$

Exergy or Availability at state point 4

$$E_4 = m_s(h_4 - T_0 s_4) kW \quad (8)$$

Irreversibility in the boiler is,

$$I_{BOILER} = (E_A - E_B) - (E_1 - E_4) kW \quad (9)$$

Substituting the E1, E4 from Eq. (7) & (8)

$$I_{BOILER} = m_g(E_A - E_B) - m_s[(h_1 - h_4) - T_0(s_1 - s_4)] kW \quad (10)$$

**Steam Turbine**:
The irreversibility in the steam turbine is given by

$$I_{TURBINE} = T_0 m_s(s_2 - s_1) kW \quad (11)$$

**Condenser:**
Mass flow rate of cooling water circulated to condense ms, kg/s, of steam is obtained from the energy balance is

$$m_{cw} c_{pw}(T_{wi} - T_{wo}) = m_s(h_2 - h_3)$$

$$m_{cw} = \frac{m_s(h_2 - h_3)}{c_{pw}(T_{wi} - T_{wo})} \quad (12)$$

Irreversibility in the condenser,

$$I_{CONDENSER} = T_0 \left[ m_s(s_2 - s_3) - (m_{cw} c_{pw} \ln(\frac{T_{wo}}{T_{wi}}) \right] kW \quad (13)$$

**Exhaust:**
Irreversibility of the exhaust,

$$I_{EXHAUST} = E_B \quad (14)$$

**Pump:**
Irreversibility rate in the boiler feed pump,

$$I_{PUMP} = m_s T_0(s_2 - s_1) kW \quad (15)$$

**Total Irreversibility:**
Total Irreversibility is,

$$I_{TOTAL} = [I_{BOILER} + I_{TURBINE} + I_{CONDENSER} + I_{EXHAUST} + I_{PUMP}] kW \quad (16)$$

**Fractional Exergy Loss:**
The definition of the fractional exergy loss of the component is the ratioof irreversibility of the individual component to the total irreversibility ofthe cycle. Its value is estimated for all the components of the cycle. It gives the information regarding the loss of useful energy in allcomponents for different operating variables. The Fractional exergy loss formulas of each component are as follows,
Fractional exergy loss in the boiler is

$$\frac{I_{BOILER}}{\sum I} \times 100 \quad (17)$$

Fractional exergy loss in the turbine is,

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$$\frac{I_{TURBINE}}{\Sigma_I} \times 100 \qquad (18)$$

Fractional exergy loss in the condenser is,

$$\frac{I_{CONDENSER}}{\Sigma_I} \times 100 \qquad (19)$$

Fractional exergy loss in the Pump is,

$$\frac{I_{PUMP}}{\Sigma_I} \times 100 \qquad (20)$$

Fractional exergy loss in the exhaust is,

$$\frac{I_{EXHAUST}}{\Sigma_I} \times 100 \qquad (21)$$

## 4. Results and Discussions

The energy and exergy analysis has been carried out for the supercritical rankine cycle and also the fractional exergy losses of each component of the cycle are determined and the results are shown in the form of graphs.

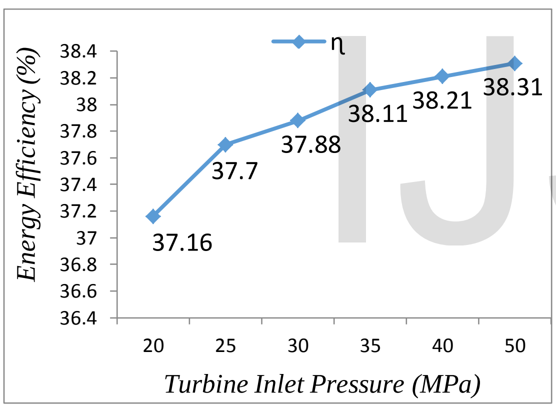


**Fig. 3. Variation of energy efficiency with different turbine inlet pressure of steam**

Figure 3 represents the variation of cycle efficiency against the various steam turbine inlet pressures at given turbine inlet temperature of the steam. From the figure we can see that the energy efficiency of the cycle at a temperature say 600°C is increases with increase in inlet pressure. The energy efficiency at turbine inlettemperature of600°C are 37.16% at 20MPa, 37.7% at 25MPa, 37.88% at 30MPa, 38.11% at 35MPa, 38.21% at 40MPa and 38.31% at 50MPa respectively.

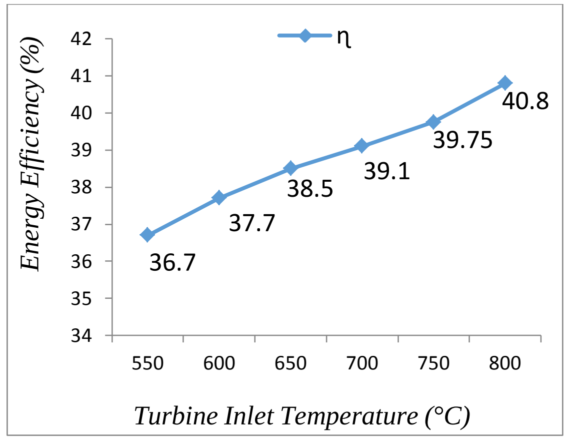


**Fig.4.Variation of energy efficiency with different turbine inlet temperature of steam**

On increasing turbine inlet temperature, turbine power output (Wt) increases, as the turbine inlet temperature increases, however power input to the pump (Wp) remains same. So, as a result of this, the network output from the turbine (Wt-Wp) increases as the turbine inlet temperature increases. But on careful observation, it may also be noted that the energy efficiency of the cycle increases significantly with turbine inlet temperature due to faster rate of increase of turbine work compared to heat supplied.From figure 4 we observe that the energy efficiency at turbine inlet temperature of 250MPa is 36.7% at 550°C, 37.7% at 600°C, 38.5% at 650°C, 39.1% at 700°C, 39.75% at 750°C and 40.8% at 800°C.

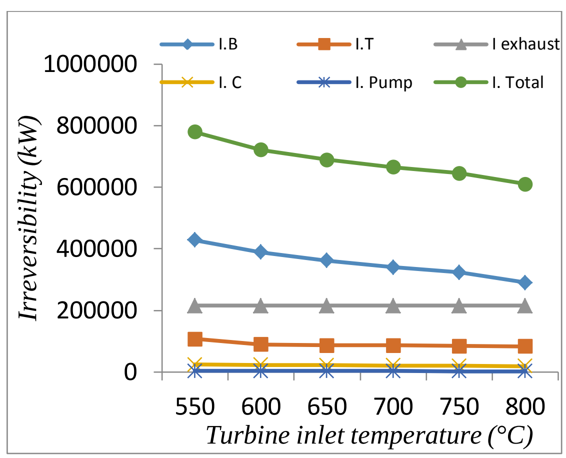


**Fig.5. Variation of Irreversibility in different components at different turbine inlet temperature.**

As shown in the graph 5 the irreversibility in all the components decreases with increase in the turbine inlet temperature but the value of irreversibility in exhaust remain constant which means that turbine inlet temperature has no influence on irreversibility in exhaust. It is significant, to point out, that the major stake of total irreversibility of the boiler is due to the irreversibility in the boiler alone.

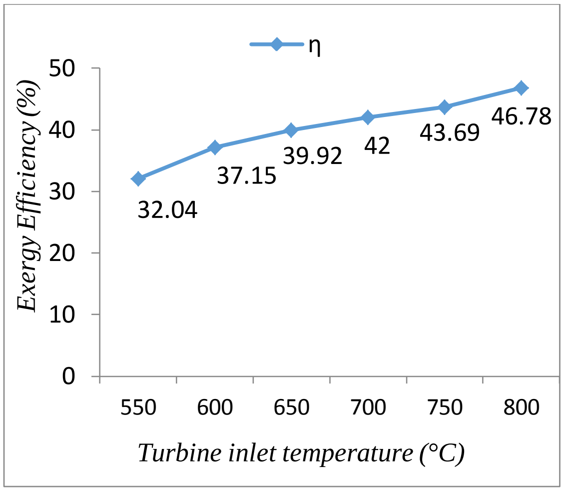


**Fig.6. Graph of Exergy efficiency with Turbine Inlet Temperature**

Exergy efficiency increases with turbine inlet temperature at a constant turbine inlet pressure as shown in figure 6. The exergy efficiency is increased from 32.04% to 46.78% from temperature 550°C to 800°C.

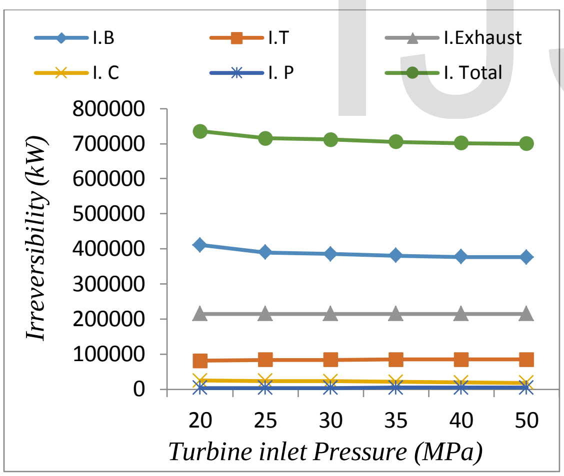


**Fig.7. Variation of Irreversibility in different components at different turbine inlet pressure**

On careful observation it may be noted that, the amount of irreversibility in the boiler, condenser and pump reduces with increase in turbine inlet pressure with a given turbine inlet temperature but the value of irreversibility of turbine increases with increase in pressure. It is also important to note that the amount of irreversibility at exhaust does not vary with turbine inlet pressure. Further it is significant, to point out,that the major stake of total irreversibility of the boiler is due to the irreversibility in the boiler alone.

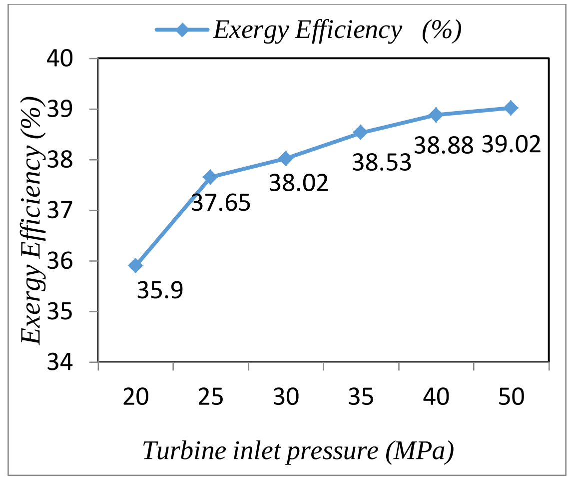


**Fig.8. Graph of Exergy efficiency with Turbine Inlet Pressure**

The total irreversibility of all the components of the cycle put together is decreasing with increase in the turbine inlet temperature. This results in the increases in the exergy efficiency with turbine inlet temperature at a constant turbine inlet pressure.

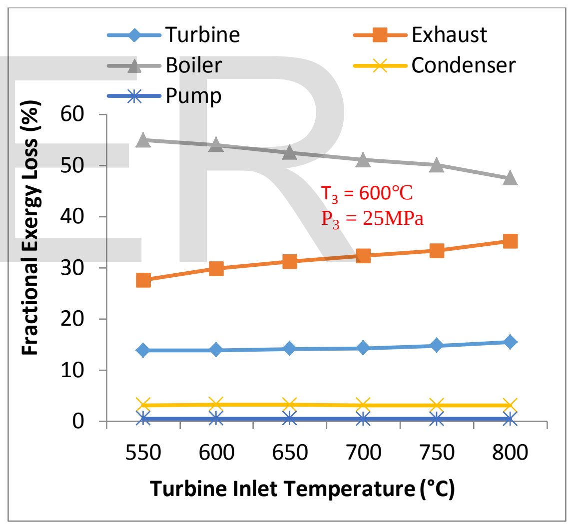


**Fig.9. Variation of fractional exergy loss with turbine inlet temperature**

Figure 9 shows the effect of turbine inlet temperature on fractional exergy loss of different components of a Supercritical rankine cycle.It may be noted from the figure that, the FEL of the boiler decreases from 54.9% to 47.52%, FEL of the turbine increases from 13.84% to 15.46% and further FEL of the condenser increases marginally from 6.24% to 7.19% with an increase of turbine inlet temperature from 500°C to 800°C. FEL of the pump is about 1% and FEL of the exhaust increases marginally from 27% to 35% with increase of turbine inlet temperature from 550°C to 800°C. In view of large magnitude of FEL in boiler, an extensive investigation is required to study the ways of

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minimizing the FEL in boiler. However, the present investigation has not focused on this point.

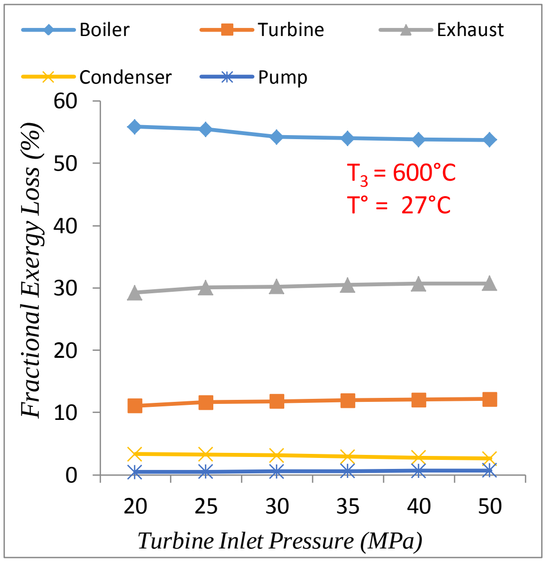


**Fig.10. Variation of fractional exergy loss with turbine inlet pressure**

It may be noted from this figure that, FEL of the boiler decreases with increase of pressure at a given inlet temperature. It is found that, FEL ofboiler at 20MPa is 55.87%, 25MPa is 54.44%, 30MPa is 54.22%, 35MPa is 53.99%, 40MPa is 53.81% and 50MPa is 53.77% respectively where as in the turbine, the FEL increases with an increase of pressure.FEL in the condenser decreases marginally from 3.32% to 2.63% and FEL of the pump is very less.

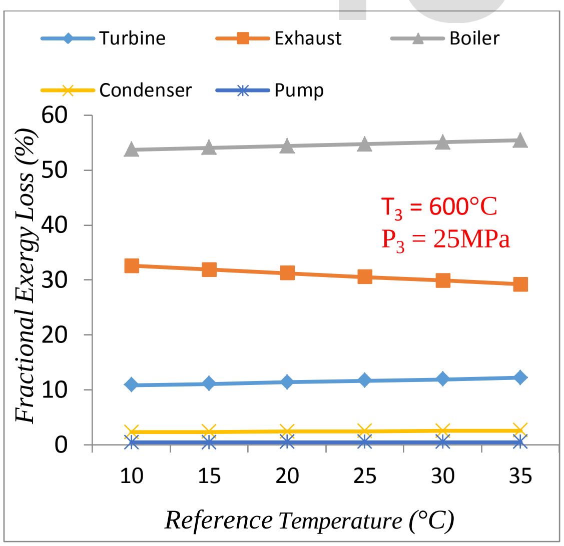


**Fig.11. Variation of fractional exergy loss with reference or atmospheric temperature.**

Reference temperature or atmospheric temperature also has a great significance on fractional exergy losses. It may be noted from the figure 11 that, FEL of the boiler increases with increase of reference temperature at a given inlet temperature or pressure. It is found that, FEL of boiler is 53.71% at 10°C, 54.38% at 20°C and 55.41% at 35°C whereas in the turbine, the FEL increases with an increase of reference temperature from 10.88% to 12.22%. FEL in the condenser increases marginally from2.34% to 2.63% with an increase of reference temperature from 10°C to 35°C. FEL of the pump is very less and it is insignificant.

## 5. Conclusions

The energy and exergy analyses of the cycle has been performed for pressure range between 20MPa to 50MPa and temperature range 550°C-800°C and the results are shown in the figures 3 to 11. It is found that with the increase in boiler pressure and turbine inlet temperature, both the energy and exergy efficiency of the plant increases. 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 and pump. Further it was observed 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 supercritical rankine cycle was considered for calculation of energy efficiency, exergy efficiency and the irreversible losses.



## Nomenclature

Cpw= Specific heat of cold water (kj/kg-K)
h = Enthalpy (kj/kg)
I = Irreversibility (kW)
mcw= Mass flow rate of cold water (kg/sec)
ms= Mass flow rate of steam (kg/sec)
P = Pressure (bar)
$P_0$ = Absolute Pressure (bar)
S = Entropy (kj/kg-K)
T = Temperature (°C)
$T_0$ = Absolute Temperature (°C)
TA= Temperature of flue gas inlet (°C)
TB = Temperature of flue gas outlet (°C)
Twi= Temperature of the cooling water inlet (°C)
Two = Temperature of the cooling water outlet (°C)
Wnet = Net work done (kj/kg)
⊚ = Summation
SUFFIX;
A = Flue gas inlet
B = Flue gas outlet
cw = cooling water
wi = water inlet
wo = water outlet

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