# Second Law Analysis of Super Critical Cycle

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**Abstract**

Coal is the key fuel for power generation in the Supercritical Rankine cycle. Exergy, a universal measure has the work potential or quality of different forms of energy of energy in relation to a given environment. In this paper, an exergy analysis has carried out to the supercritical power plant tells us how much useful work potential or exergy, supplied to the input to the system under consideration has been consumed by the process. A computer code has developed for exergy to analyses the supercritical cycle without reheat as well as with single reheat. The temperature and pressure inlet to the turbine and exhaust pressure from the turbine are identified as key parameters in this analysis. Both first law efficiency and exergetical efficiency have studied at various temperature and pressure inlet to the turbine. Irreversibility as well as Fractional exergy loss of all the components has also been studied. To decrease exergy loss of supercritical power plant, effects of pump discharge pressure increases, effects of steam turbine discharge pressure decreases and effects of steam temperature increases. First law efficiency is increases with increase in temperature at a given pressure. Exergy efficiency is increases with increase in temperature and pressure. It is found that both the efficiencies increases more in temperature rise than the pressure rise in the turbine inlet. Both Irreversibility and Fractional exergy losses in the boiler is reducing with increase in temperature.

**Keywords:** Supercritical cycle, reheat, cycle efficiency, exergy efficiency and Fractional exergy loss

# 1. INTRODUCTION

Advanced Coal fired electric power plants that are cleaner, more efficient and less costly than the current fleet of coal fired power plants. The efficiency of power plants in developing countries like India and China are still around 32-35% lower heating value, modern sub critical cycles have attained efficiencies close to 40%. Further improvement in efficiency can be achieved by using supercritical steam conditions. Current supercritical coal fired power plants have efficiencies above 45%. Presently, there are more than 450 supercritical power plants are available in operation. Coal based thermal power plants are the main source of power generation in India. Energy is an important ingredient of economic development. Economic growth is directly or indirectly related to energy consumption. The total installed capacity of Indian power plants is 104,917 MW, nearly 74,420MW are accounted by thermal power generation and of this about 71% of electricity generation is coal based. The need today is to have low emission with high efficiency of operation; hence it is necessary that the sub critical operation limit have to go supercritical ranges, which is beyond 221.2bar steam pressure. Supercritical units has around 3 percent higher efficiency resulting in 8 to 10 percent savings in fuel than the sub critical units, since the fuel fired is reduced, hence the emissions are also less.

The cycle originally taken from the Rankine cycle. The basic thermodynamic principles and exergy methods are taken from the Bejan, A [1]. Nag, P.K., and Gupta are analyzed Exergy analysis of Kalina cycle [2]. El-sayed, and M.Tribus, made a Theoretical Comparison of Rankine with Kalina cycle. T.J.Kotas made description of the exergy as well as Enthalpy of the flue gas inlet and outlet of the boiler [3]. The Supercritical Rankine cycle comparison with the Rankine cycle working with same temperature limits. The area between the line of heat source and the line of representing the working fluid corresponds to the exergy loss in the process of heat transfer in HRSG. These losses are less in the supercritical cycle compared with Rankine cycle.

There are two ways usually considered for increasing the efficiency of the Supercritical Rankine Cycle utilizing variable-temperature heat sources. One is the use of a multi pressure boiler, and the other is the implementation of the so-called “supercritical cycle”. The use of a multi pressure boiler is widely accepted in the industry, but results in only moderate improvement unless the number of such boiling steps is very large. However, a significant increase in the number of boiling steps is technically and economically unfeasible and, as a result, the number of such steps does not exceed three. The use of supercritical cycle, especially with organic and exotic working fluids, can theoretically achieve a triangular shape of the cycle, and thus high efficiency, but requires extremely high pressure in the boiler, which in turn has an adverse effect on turbine performance [4]. S.L.Milora and J.W.Tester have presented a very complete study of potential of supercritical cycle [5]. As follows from their data, such a cycle has one more setback, i.e., if the working fluid is heated to a higher temperature on a turbine inlet, the temperature in the turbine outlet is relatively high, and the remaining heat cannot be properly utilized in the cycle. El-sayed, and M. Tribus, made a Theoretical Comparision of Rankine with Kalina cycle [6]. The exergy method is the best-known member of a class of techniques of thermodynamic analyses, which are, collectively referred to as second law analysis. An account of the historical development of second law analysis can be found in a paper by Haywood. Fratzscher and Beyer have given a critical examination of the developments in exergy analysis with special reference to the decade 1970-80. Kotas, Mayhew and Raichura made nomenclature for exergy analysis [7]. In this paper, complete analysis is given only for the supercritical cycle with single reheat.

# 2. REHEAT SUPECRITICAL CYCLE

Figures 1 and 2 are the schematic and T-s diagrams for the supercritical reheat power cycle. It is triangular in shape because there is no latent heat of vaporization during the boiling process. The temperature profile is well matched with flue gas temperature line; hence there would not be much of exergy loss during heat exchange process in the boiler. The steam may be reheated to a high temperature after it has partially expanded through the turbine. A significant portion of the work by the steam is accomplished when the pressure is such that the steam is saturated or nearly saturated. This is the correct place for the vapor to be re-superheated. The steam reenters the turbine and expands to condenser pressure. The steam expands through the turbine until state 2 is reached, then it removed and reheated ($P_1 = 0.2 \times P_{rh}$) at a constant pressure to state 3. The steam reenters the turbine at state 3 and expands the condenser at state 4. The steam is condensed (state point 4-5) and pumped (state point 6-1) back to the steam generator, completing the cycle.

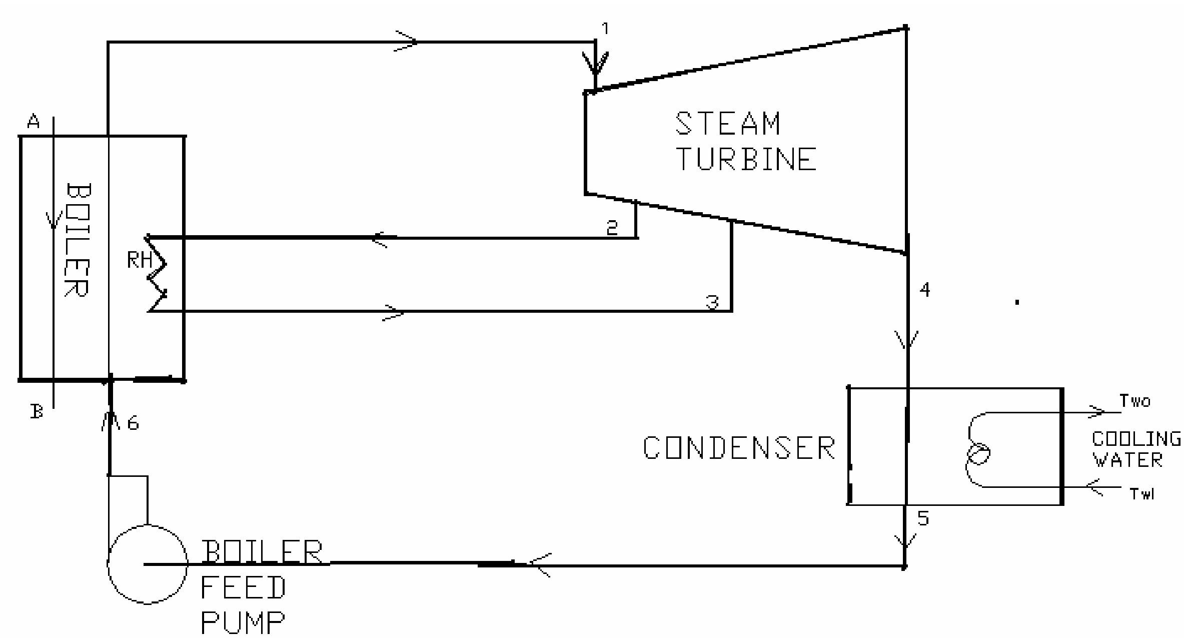


Fig1. Schematic diagram of Supercritical cycle with reheat

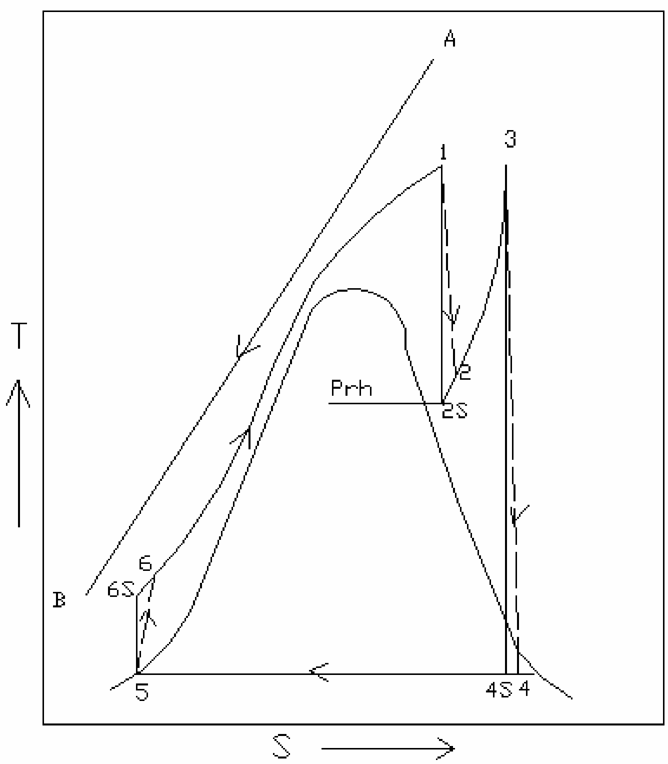


Fig 2. T-S diagram of Supercritical cycle with reheat

# 3. EXERGY ANALYSIS OF THE CYCLE:

## 3.1 ASSUMPTIONS:

1. Capacity of the power plant = 660 MW
2. Reheat pressure = 0.2 times the initial pressure
3. No heat losses and no pressure losses
4. The isentropic efficiency of the steam turbine is 90%.
5. The pump efficiency is assumed to be 80%.
6. Flue gas entering into the boiler is $970^0$C and leaving is $210^0$C.
7. The pinch point temperature difference for the heat exchanges in the condenser is $5^0$C.
8. Condenser pressure $P_c$= 0.06 bar
9. Cooling water temperature inlet to the condenser $T_{wi}$=$25^0$C

## ANALYSIS

## 3.2 CYCLE EFFICIENCY:

To calculate the cycle efficiency for the supercritical reheat cycle, the work and heat added terms must be found. An energy balance equation on the turbine yields,

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

$$W_{turbine} = ((h_1 - h_2) + (h_3 - h4)) \quad kJ/kg \tag{1}$$

Boiler pump work per kg of steam supplied,

$$W_{pump} = h_6 - h_5 \quad kJ/kg \tag{2}$$

Heat supplied to steam boiler,

$$H.S = h_1 - h_6 \quad kJ/kg \tag{3}$$

$$W_{net} = W_{turbine} - W_{pump} \tag{4}$$

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

$$\text{Cycle efficiency} = \quad (W_{turbine} - W_{pump}) \,/\, H.S \tag{5}$$

## 3.3 EXERGY EFFICIENCY:

The method of exergy analysis aims at the quantitative evaluation of the exergy destructions and losses (irreversibilities) associated with a system. Hence it is required to calculating the irreversibility in all the components of the power cycle for the estimation of exergy efficiency.

The irreversibility or exergy losses in each of the components are calculated for the specified dead state.

Let $P_{0,}$ $T_0$ are the pressure and temperature of the system when it is in the dead state.

### 3.3.1 Boiler:

The coal used is anthracite of the chemical composition of the power plant has taken form T.J.Kotas [3] are as:

| | $CO_2$ | $H_2O$ | $N_2$ | $O_2$ | $SO_2$ | Total |
|---|---|---|---|---|---|---|
| $n_k$[kmol/ 100kgfuel] | 6.51 | 1.634 | 35.32 | 9.324 | 0.047 | 57.735 |

$$H_A = \left(\theta_A - \theta^0\right)\sum_k n_k \tilde{c}_{pk}^{h} \quad \text{kJ} \qquad (6)$$

$$H_B = \left(\theta_B - \theta^0\right)\sum_k n_k \tilde{c}_{ph}^{h} \quad \text{kJ} \qquad (7)$$

$$E_A = \left(\theta_A - \theta^0\right)\sum_k n_k \tilde{c}_{pk}^{\varepsilon} \quad \text{kJ} \qquad (8)$$

$$E_B = \left(\theta_B - \theta^0\right)\sum_k n_k \tilde{c}_{pk}^{\varepsilon} \quad \text{kJ} \qquad (9)$$

Where mean isobaric heat capacity for evaluating enthalpy changes is

$$\bar{c}_p^{\,h} = \left[\frac{\bar{h} - \bar{h}^0}{T - T_0}\right] = \frac{1}{T - T_0}\int_{T_0}^{T} \bar{c}_p dT \quad \text{and} \qquad (10)$$

Mean molar isobaric exergy capacity for evaluating changes in physical exergy is

$$\bar{c}_p^{\,\varepsilon} = \left[\frac{\bar{\varepsilon}^{\Delta T}}{T - T_0}\right] = \frac{1}{T - T_0}\left[\int_{T_0}^{T} \bar{c}_p dT - T_0\int_{T_0}^{T} \frac{\bar{c}_p dT}{T}\right] \qquad (11)$$

Where $H_A$ = Enthalpy of flue gases entering The boiler,

$H_B$ = Enthalpy of flue gases leaving the boiler
$Ex_A$ = Exergy in the flue gas at the entering the boiler
$Ex_B$ = Exergy in the flue gas at the exiting from the boiler

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 power plant.

$m_s(W_{net})$ = 660 MW
$m_s$= 660x1000 kW/ $W_{net}$ kg/sec (12)

Energy balance equation for obtaining the number of flue gases ($m_g$) is,
Heat gained by the steam = Heat lost by the flue gases.

$$m_s((h_1 - h_6) - (h_3 - h_2)) = m_g(H_A - H_B)$$

$$m_g = ((h_1 - h_6) - (h_3 - h_2)) / (H_A - H_B) \qquad (13)$$

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 $\theta_A$ = 970°C and $\theta^0 = 25$ °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 $Ex_{in} = Ex_A$ = 958953.69 kJ

Enthalpy of the flue gas entering is $H_A$ =1711969.25 kJ

Exergy in the flue gas at the exit the boiler is $Ex_{out} = Ex_B$ =68474.05 kJ.

Enthalpy of the flue gas at exit of the boiler is $H_B$ = 304951.47 kJ.

Availability or Gibbs function of steam at state point 1

$$G_1 = Es_1 = m_s\,(h_1 - T_o s_1) \text{ kJ} \qquad (14)$$

Availability or Gibbs function of steam at state point 6 $G_6 = Ew_6 = m_s\,(h_6 - T_o s_6)$ kJ (15)

Irreversibility in the boiler is $I_{boiler} = Ex_A - Ex_B - (Es_1 - Ew_6)$ kJ (16)

**3.3.2 Steam Turbine:**

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

$$I_{turbine} = T_0 . m_s((s_2 - s_1) + (s_4 - s_3)) \text{ kW} \qquad (17)$$

**3.3.3 Condenser:**

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

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

$m_{cw}\, C_{pw}\,(T_f - T_d) = m_s\,(h_2 - h_3)$ (18)

$m_{cw} = m_s(h_2 - h_3)/\, C_{pw}\,(T_f - T_d)$ is 154735.2 kg.

Irreversibility in the condenser,

$I_{condenser} = T_0[m_{cw} C_{pw} \ln(T_f/T_d) - m_s(s_2 - s_3)]$ kW (19)

**3.3.4 Pump :**

Irreversibility rate in the boiler feed pump,

$I_{pump} = m_s\, T_0(s_6 - s_5)$ kW (20)

**3.3.5 Exhaust:**

Irreversibility or exergy loss through the exhaust

$I_{exhaust} = Ex_B$ =103041.92 kJ.

Total Irreversibility is

$\Sigma I = I_{boiler} + I_{turbine} + I_{pump} + I_{condenser} + I_{exhaust}$ kW (21)

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.

Exergy efficiency, $\eta_{II} = \dfrac{\left(Ex_A - \sum I\right) * 100}{Ex_A}$ (22)

## 4. FRACTIONAL EXERGY LOSS:

The definition of the fractional exergy loss of the component is 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 of each component are as follows.

Fractional exergy loss in the boiler is,

$$\frac{I_{boiler}}{\sum I} * 100 \qquad (23)$$

Fractional exergy loss in the turbine is,

$$\frac{I_{turbine}}{\sum I} * 100 \qquad (24)$$

Fractional exergy loss in the condenser is,

$$\frac{I_{condenser}}{\sum I} * 100 \qquad (25)$$

Fractional exergy loss in the Pump is,

$$\frac{I_{pump}}{\sum I} * 100 \qquad (26)$$

Fractional exergy loss in the exhaust is,

$$\frac{I_{exhaust}}{\sum I} * 100 \qquad (27)$$

## 5. RESULTS AND DISCUSSION:

All the dashed lines in the following graphs are related to single reheat where as other is without reheat. It is observed from figure 3 that with the increase in pressure for a particular temperature cycle efficiency increases for a supercritical cycle with no reheat. Also with the consideration of a single reheating, the cycle efficiency is increasing with the increase in pressure, but the increase in efficiency is high when compared to a supercritical cycle without reheating. The cycle efficiency at a pressure of 200 bar is 44.02% and for a pressure of 425 bar the efficiency is 44.94%. It is also observed that the increase in cycle efficiency is very less when compared to the increase in turbine inlet pressure limits. It is observed from figure 4, the cycle efficiency increases with the increase in turbine inlet temperature at a particular pressure for a supercritical cycle without reheating. The cycle efficiency also increases with increase in inlet temperature at a particular pressure by considering the reheating, but the increase in efficiency is high when compared to a supercritical cycle without reheating.

Fig 5 shows the variation of exergy efficiency with increase in pressure at different turbine inlet temperature. It is obvious that exergy efficiency increases with increase in pressure at a particular turbine inlet temperature for a supercritical cycle without reheating. The exergy efficiency also increases for a supercritical cycle with reheating and the increase in efficiency is high when compared to a cycle without reheating.

Figure 6 shows the variation of exergy efficiency with temperature at different turbine inlet pressures. Exergy efficiency increases with increase in temperature at different turbine inlet pressures with and without reheating. The increase in exergy efficiency is high for a cycle with reheating when compared to a cycle without reheating.

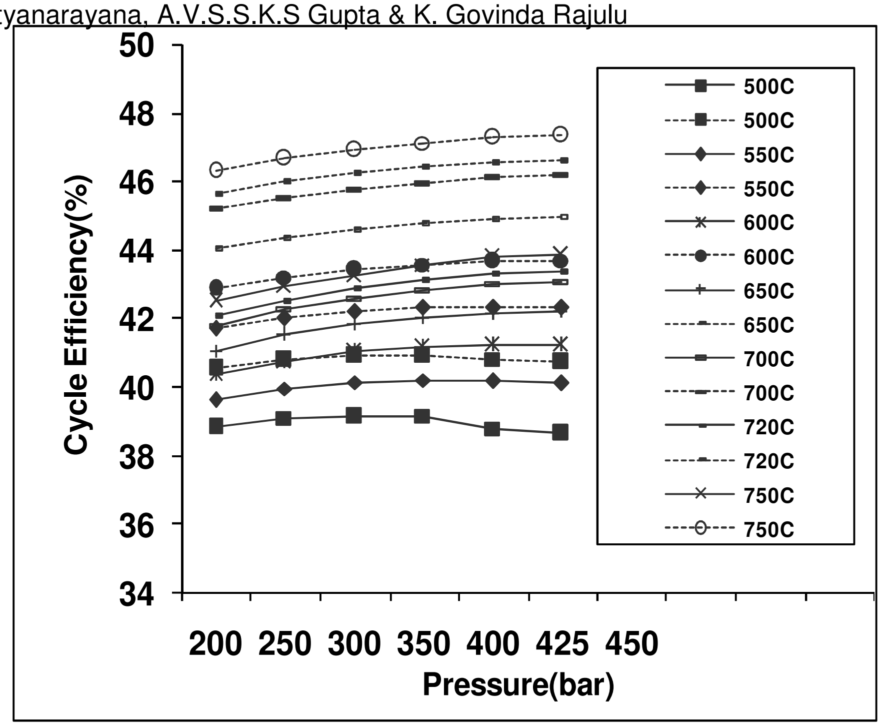


Fig.3. Variation of Cycle efficiency with P1 for Different values of T1 at $P_c$= 0.06 bar

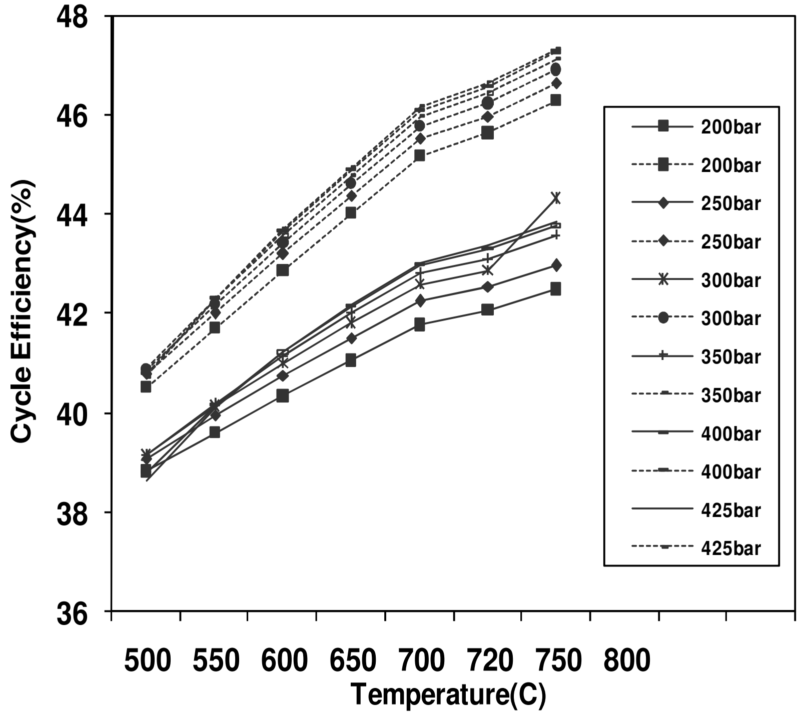


Fig.4. Variation of Cycle efficiency with T1 for Different values of P1 at $P_c$= 0.06 bar.

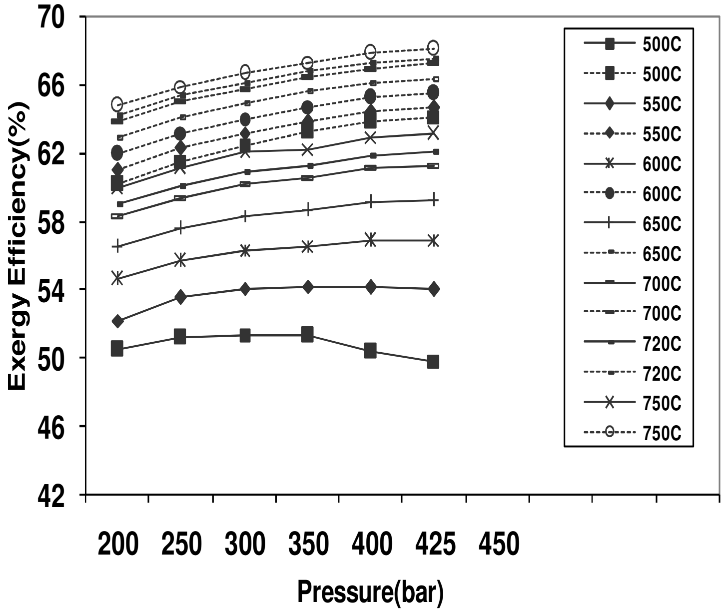


Fig5. Variation of Exergy efficiency with P1 for Different values of T1 at $P_c$= 0.06 bar.

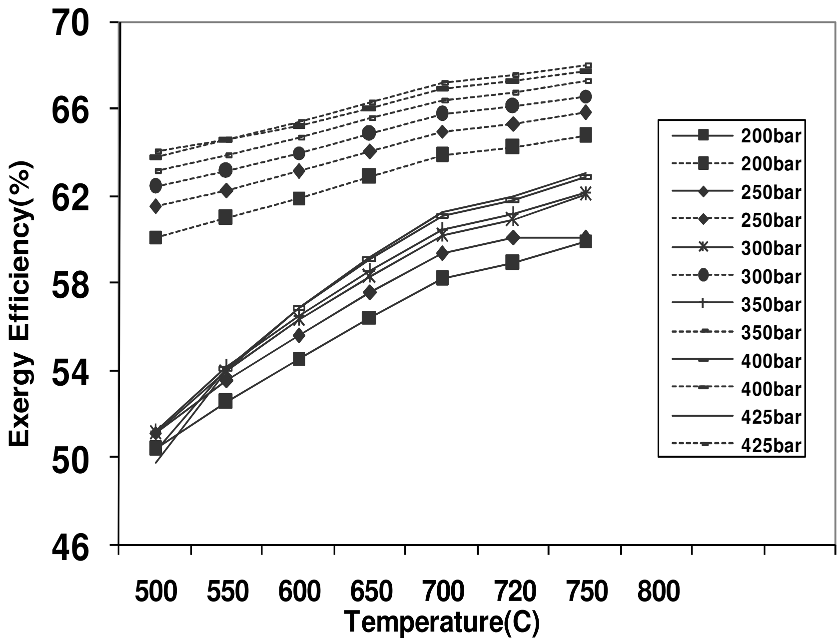


Fig 6. Variation of Exergy efficiency with T1 for Different values of P1 at $P_c$= 0.06 bar.

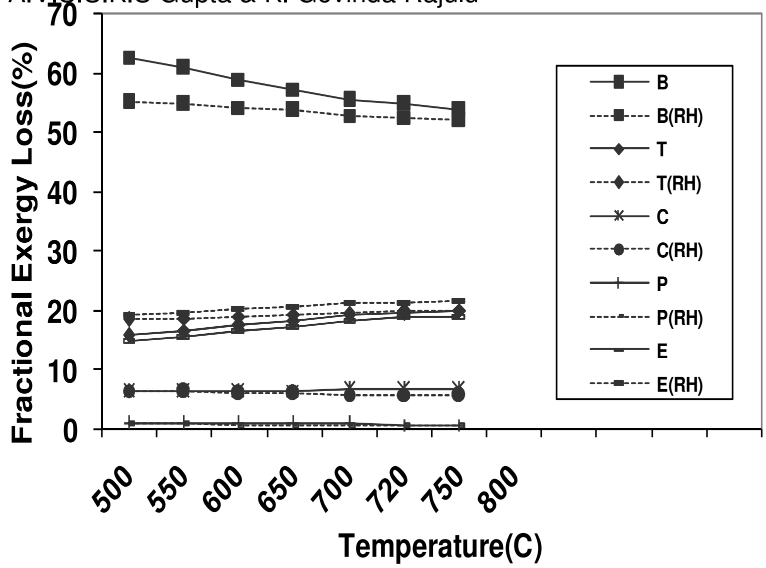


Fig 7. Effect of Temperature on Fractional Exergy loss of different components for P1=325bar

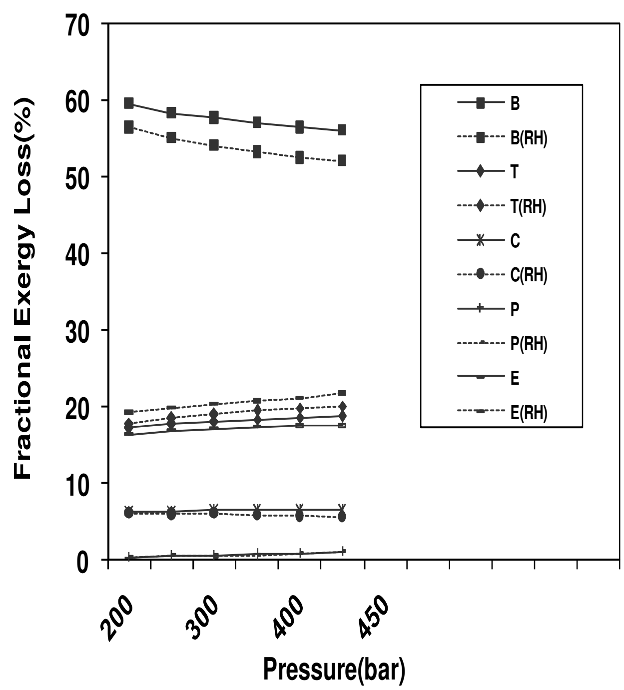


Fig 8. Effect of Pressure on Fractional Exergy loss of different components for T1=650$^0$C

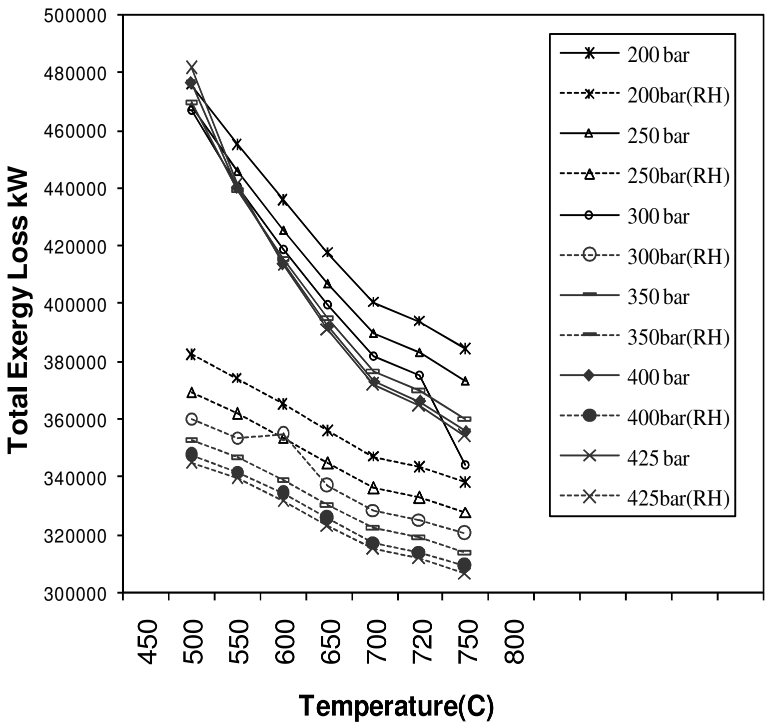


Fig 9. Variation of Total Exergy Loss with T1 for different values of P1

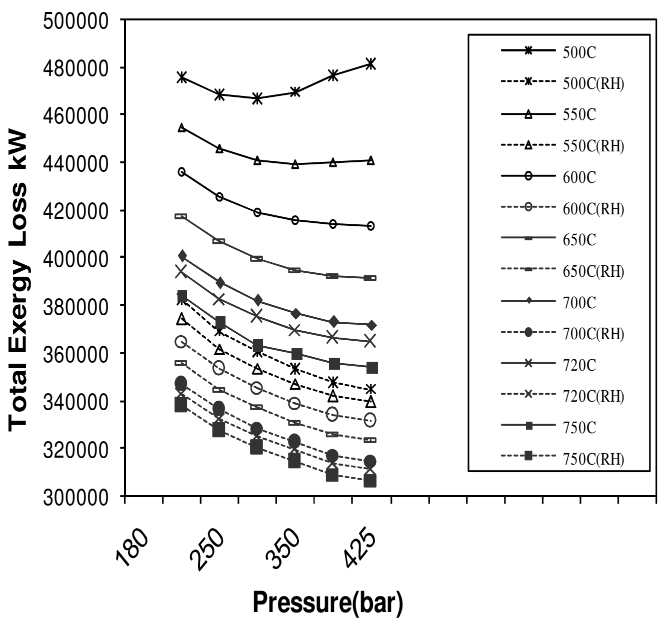


Fig 10 Variation of Total Exergy Loss with P1 for different values of T1

Figure 7shows the variation of fractional exergy loss of different components at different temperatures with and without reheating. It is observed that for a boiler the decrease in fractional exergy loss with increase in temperature is high for a cycle with reheating when compared to a cycle without reheating. At a Pressure/Temperature of 325bar/650$^0$C the decrease in fractional exergy loss in the boiler with reheating is nearly 6% when compared to a cycle without reheating. The fractional exergy loss is reduced nearly 5% in the turbine when compared with reheating.
Figure 8 shows the variation of fractional exergy loss of different components at different pressures with and without reheating. The fractional exergy loss in the boiler without reheat is 62.72% at 325bar/500$^0$C where as it is 53.73% at 325bar/750$^0$C. The fractional exergy loss with reheat at the same pressure at 500$^0$C is 54.92% and at 750$^0$C is 52.11%. The fractional exergy loss in the turbine is decreases with increase in temperature at particular pressure, but at higher temperature both fractional exergy loss is almost nearly same with and without reheating.

Figure 9 shows the variation of total exergy loss with temperature for different values of turbine inlet pressures with and without reheating. The total exergy loss is decreasing with increase of pressure and temperatures with and without reheating of the different components in the supercritical cycle. It is observed that total exergy loss is less with reheating when compared without reheating.

Fig 10 shows the variation of total exergy loss with pressure for different values of turbine inlet temperatures with and without reheating. The total exergy loss is decreasing with increase of temperature with and without reheating of the different components in the supercritical cycle. It is observed that total exergy loss is less with reheating when compared without reheating.

## 6. CONCLUSIONS:

This paper analyzes the supercritical cycle with and without reheat from both cycle efficiency, exergy efficiency. The energy and exergy analyzes of the cycle has been performed pressure range between 200bar to 425 bar and temperature range are 500$^0$C-800$^0$C. First law analysis and second law analysis has carried out throw with and withot reheat. The irreversibility and fractional exergy loss are determined for the cycle with and without reheat. It is found that the cycle efficiency is high in reheat than the non-reheat supercritical cycle. It is also conclude that exergy efficiency is high in reheat than non-reheat supercritical cycle. It is found that nearly 20-25% irreversibility is reduced by using single reheat in the boiler, where as it is 12-15% in the turbine than the without reheating. Fractional exergy losses of all the components in the cycle is determined and compared with and without supercritical cycle.

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## NOMENCLATURE:

$h$ = enthalpy (kJ/kg)

$P$ = Pressure (bar)

Ambient temperature( $^0$C)

$s$ = entropy (kJ/kg-K)

$\sum$= Sum

$m_s$ =Mass flow rate of steam( kg/s)

$m_g$ = Number of moles of the flue gas

$m_{cw}$ = Mass of cooling water (kg/s)

$I$ = Irreversibility (kW)

$T_o$ = Absolute Temperature (K)

$C_{pw}$ = Specific heat at constant pressure (kJ/kg-K)

$T_{wo}$ =Temperature of the cooling water out( $^0$C)

$T_{wi}$ =Temperature of the cooling water in($^0$C)

$W_{net}$ = Net work done (kJ/kg)

$G$ = Gibbs function (kJ)

*Suffix:*

*rh = reheat*

*cw=cooling water*

*wi = water inlet*

*wo=water outlet*

*A = flue gas inlet*

*B = flue gas outlet*

s*up= supplied*

ABRREVATIONS:

B = boiler

T = turbine

C = condenser

P = pump

E = exhaust

H.S = heat supplied