"24P"

UNIVERSITI TEKNOLOGI MARA
University Publication Centre (UPENA)

Journal of Mechanical Engineering

An International Journal

Volume 7 No. 1 July 2010 ISSN 1823-5514

JOURNAL OF MECHANICAL ENGINEERING (JMechE)

Journal of Mechanical Engineering (ISSN 1823-5514) is jointly published by the Faculty of Mechanical Engineering (FKM) and University Publication Centre (UPENA), Universiti Teknologi MARA, 40450 Shah Alam, Selangor, Malaysia.

Journal of Mechanical Engineering

An International Journal

Volume 7 No. 1 **July 2010** **ISSN 1823-5514**

Journal of Mechanical Engineering *Vol. 7, No. 1, 91-107, 2010*

Exergy Analysis of Supercritical Cycle for 1000 MW Power Generation Using Without Reheat, Single and Double Reheat

I. Satyanarayana
A.V.S.S.K.S. Gupta
K. Govinda Rajulu
Department of Mechanical Engineering
JNTUH College of Engineering
Jawaharlal Nehru Technological University Hyderabad
Kukatpally, Hyderabad - 500 085, India
Corresponding author / email: sindigibilli@gmail.com

ABSTRACT

This paper presents Exergy analysis of Supercritical Rankine cycle without reheat, single reheat and double reheat for higher power generation of modern steam power plants for 1000 MW. A computer code has been developed to estimate the steam properties using ISI steam tables. The temperature and pressure inlet to the turbine and exhaust pressure from the turbine are identified as key parameters in this analysis. Cycle efficiency, exergy efficiency, work ratio, steam rate and heat rate has been studied. Total exergy loss and fractional exergy loss of all the components in the cycle have been analyzed. Effect of decreasing the discharge pressure in the condenser, effect of flue gas inlet and outlet temperature of the boiler and Effect of reheat pressure and number of reheats in the boiler has been studied. It is observed that both efficiencies increases more with temperature rise than pressure rise. The effects of these parameters on the exergy loss of each component have also been studied.

Keywords: *Supercritical cycle, cycle efficiency, exergy efficiency, fractional exergy loss*

ISSN 1823-5514

Introduction

Steam reheating is an important feature in steam-power plants. The main objective of reheat is to increase the power output and, under certain conditions, the thermal efficiency of the plant, thus improving plant performance. There is a wide range over which reheat pressures can be varied. Hence, for every set of steam conditions, an optimum value of reheat pressure exists that will yield an optimum steam turbine boiler reheat-cycle. Coal based thermal power plants are the main source of power generation in India. Energy is an important ingredient of economic development. The need today is to have low emission with high efficiency of operation; hence it is necessary that the sub critical operation limit must go for supercritical ranges, which is beyond 221.2 bar and 374.15°C steam pressure and temperature. The 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.

Exergy, a universal measure has the work potential or quality of different forms of energy in relation to a given environment. In this paper, an exergy analysis has carried out to the supercritical power plant to tell us how much useful work potential or exergy, supplied to the input to the system under consideration has been consumed by the process. The basic cycle of the supercritical cycle is conceived by Rankine cycle. Nag and Gupta [1] analyzed the exergy analysis of Kalina cycle. EI-sayed, and Tribus [2] made a Theoretical Comparison of Rankine with Kalina cycle. Kotas [3] made description of the exergy as well as Enthalpy of the flue gas inlet and outlet of the boiler and also chemical composition of the anthracite coal has been taken from Kotas. Kotas [4] explained the nomenclature of the exergy analysis. The properties of water/ steam have developed using ISI steam tables [5]. The basic principles of thermodynamics to analyses the thermodynamic cycles has explained by Bejan, Nag, Moran and Cengel [6,7,12,13]. Habib [8] has explained the importance of the reheat in the steam power generation.

Analysis of thermal power plants by means of the cycle efficiency is a prominent topic in the mechanical engineering. Exergy analysis provides a true measure of the system efficiency for the supercritical cycle, where the energy method gives 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 power plants are commonly coal plants, nuclear plants, geothermal or natural gas plants depending upon the type of fuel used to supply heat to steam, but the steam goes through the same basic cycle in all of them. Therefore, all can be analysed in the same manner.

This paper analyses a supercritical steam cycle emphasizing the exergy by it application without reheat, reheat and double reheat. The effect of decreasing the discharge pressure from the condenser, Effect of reheat pressure in the boiler has been studied. It is observed that both the efficiencies increases more in temperature rise than the pressure rise. The effects of these parameters on the exergy loss of each component have also been studied.

Supercritical Cycle Description

The schematic diagram of the Supercritical cycle without reheat, single reheat and double reheat, for which the analysis has been made, is shown in Figures 1, 2 and 3. The flue gas inlet temperature and out let temperature varies from 1000°C – 800°C and 80°C – 150°C. 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. Figure 4 shows the supercritical cycle without reheat. The high pressure and high temperature steam from the boiler enters the steam turbine at state 1; process 1-2 is the adiabatic reversible expansion through the turbine. The exhaust vapor at 2 is usually in the two-phase region. The steam is condensed (state point 2-3) and pumped (state point 4-1) back to the steam generator, completing the cycle.

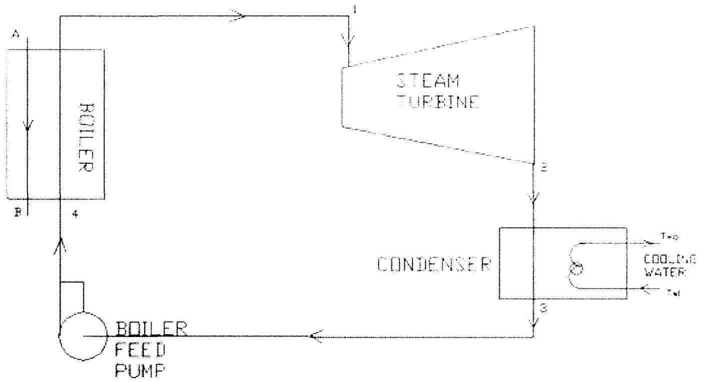

Figure 1: Schematic Diagram of the Simple Supercritical Cycle

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.

Figure 3 depicts the steam reenters the turbine twice and expands to condenser pressure. The high-pressure high temperature steam from the boiler enters the turbine at state 1 where it will be expand in the turbine. The steam expands through the turbine until state 2 is reached, then it removed and reheated up to it reaches the initial temperature at a constant pressure to state 3.

has been studied. It is observed that both the efficiencies increases more in temperature rise than the pressure rise. The effects of these parameters on the exergy loss of each component have also been studied.

Supercritical Cycle Description

The schematic diagram of the Supercritical cycle without reheat, single reheat and double reheat, for which the analysis has been made, is shown in Figures 1, 2 and 3. The flue gas inlet temperature and out let temperature varies from 1000°C – 800°C and 80°C – 150°C. 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. Figure 4 shows the supercritical cycle without reheat. The high pressure and high temperature steam from the boiler enters the steam turbine at state 1; process 1-2 is the adiabatic reversible expansion through the turbine. The exhaust vapor at 2 is usually in the two-phase region. The steam is condensed (state point 2-3) and pumped (state point 4-1) back to the steam generator, completing the cycle.

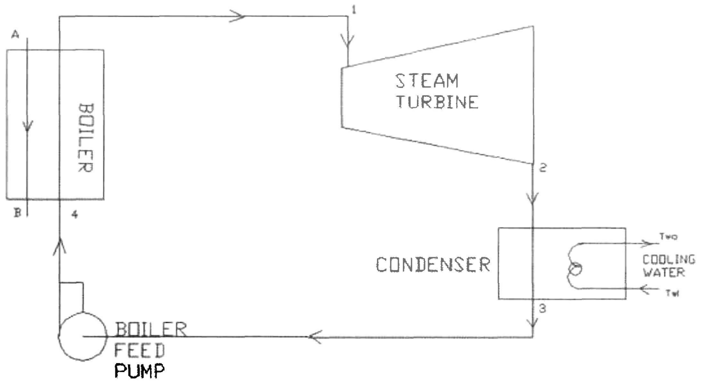

Figure 1: Schematic Diagram of the Simple Supercritical Cycle

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.

Figure 3 depicts the steam reenters the turbine twice and expands to condenser pressure. The high-pressure high temperature steam from the boiler enters the turbine at state 1 where it will be expand in the turbine. The steam expands through the turbine until state 2 is reached, then it removed and reheated up to it reaches the initial temperature at a constant pressure to state 3.

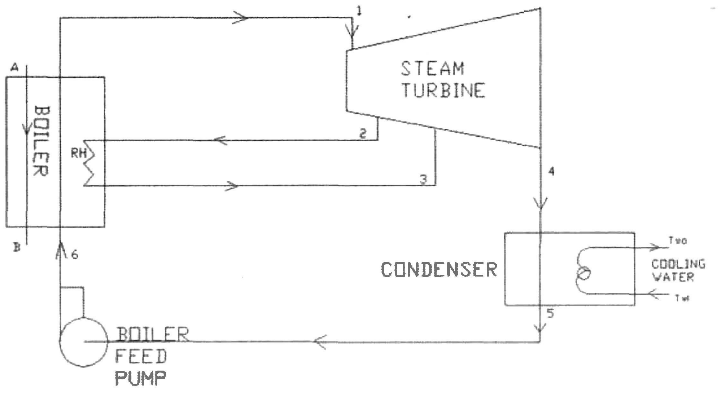

Figure 2: Schematic Diagram of the Supercritical Cycle with Single Reheat

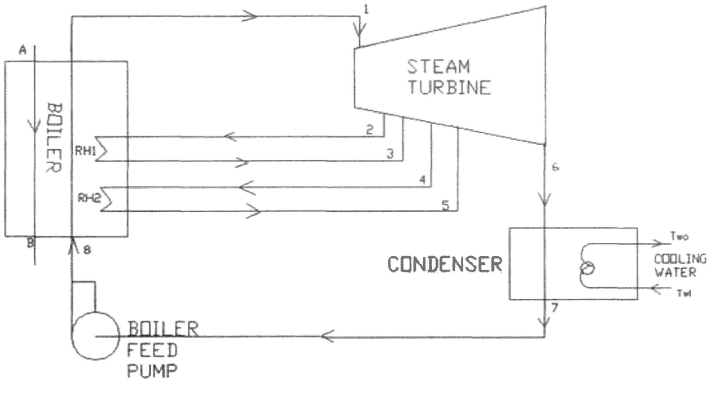

Figure 3: Schematic Diagram of the Supercritical Cycle with Double Reheat

The steam is again extracted from the turbine for further reheat at state 4. The steam expands through the turbine until state 4 is reached, then it removed and reheated at a constant pressure to state 5. The steam reenters the turbine at state 5 and expands the condenser at state 6. The steam is condensed (state point 6-7) and pumped (state point 8-1) back to the steam generator, completing the cycle.

Assumptions Used in the Analysis

1. Capacity of the power plant = 1000 MW
2. Reheat pressure = 0.20 – 0.25 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. The pinch point temperature difference for heat exchange in condenser is 5°C.
7. Condenser pressure $P_c = 0.04 - 0.07$ bar
8. Cooling water temperature inlet to the condenser $T_{wi} = 25°C$

Exergy Analysis

To calculate the cycle efficiency for the supercritical without reheat, with single reheat and double reheat, the work and heat added terms must be found.

Without Reheat:

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

$$W_{turbine} = h_1 - h_2 \qquad \text{kJ/kg} \qquad (1)$$

Boiler pump work per kg of steam supplied,

$$W_{pump} = h_4 - h_3 \qquad \text{kJ/kg} \qquad (2)$$

Heat supplied to steam boiler,

$$H.S = h_1 - h_4 \qquad \text{kJ/kg} \qquad (3)$$

Single Reheat:

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

$$W_{turbine} = ((h_1 - h_2) + (h_3 - h_4)) \qquad \text{kJ/kg} \qquad (4)$$

Boiler pump work per kg of steam supplied,

$$W_{pump} = h_6 - h_5 \qquad \text{kJ/kg} \qquad (5)$$

Heat supplied to steam boiler,

$$H.S = h_1 - h_6 \qquad \text{kJ/kg} \qquad (6)$$

Double Reheat:

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

$$W_{turbine} = ((h_1 - h_2) + (h_3 - h_4) + (h_5 - h_6)) \qquad \text{kJ/kg} \qquad (7)$$

Boiler pump work per kg of steam supplied,

$$W_{pump} = h_8 - h_7 \qquad \text{kJ/kg} \qquad (8)$$

Heat supplied to steam boiler,

$$H.S = h_1 - h_8 \qquad \text{kJ/kg} \qquad (9)$$

$$W_{net} = W_{turbine} - W_{pump} \qquad \text{kJ/kg} \qquad (10)$$

The cycle efficiency is defined as the ratio of output energy to the input energy,

$$\text{Cycle efficiency} = W_{net} / H.S \qquad (11)$$

$$\text{Steam Rate} = 3600/W_{net} \qquad \text{kg/kW-hr} \qquad (12)$$

$$\text{Heat Rate} = 3600/\text{Cycle efficiency} \qquad \text{kJ/kW-hr} \qquad (13)$$

$$\text{Work Ratio} = W_{net}/W_{turbine} \qquad (14)$$

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 calculate 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.

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

	CO_2	H_2O	N_2	O_2	SO_2	Total
n_k[kmol/100 kg fuel]	6.51	1.634	35.32	9.324	0.047	57.735
x_k	0.1234	0.0310	0.6679	0.1768	0.0009	1.000

$$H_A = (\theta_A - \theta^0)\sum_k n_k \tilde{c}^{\,h}_{pk} \qquad \text{kJ} \qquad (15)$$

$$H_B = (\theta_B - \theta^0)\sum_k n_k \tilde{c}^{\,h}_{ph} \qquad \text{kJ} \qquad (16)$$

$$E_A = (\theta_A - \theta^0)\sum_k n_k \tilde{c}^{\,\varepsilon}_{pk} \qquad \text{kJ} \qquad (17)$$

$$E_B = (\theta_B - \theta^0)\sum_k n_k \tilde{c}^{\,\varepsilon}_{pk} \qquad \text{kJ} \qquad (18)$$

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 θ_A = 1000°C to 800°C and θ_B = 150°C to 80°C and

$\theta^\circ = 25°C$ 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,

Where mean isobaric heat capacity for evaluating enthalpy changes is

$$\bar{c}_p^{\,h} = \left[\frac{\bar{h} - \bar{h}^0}{T - T_0}\right] = \frac{1}{T - T_0}\int_{T_0}^{T} \bar{c}_p dT \qquad \text{and} \tag{19}$$

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

$$\bar{c}_p^{\,\varepsilon} = \left[\frac{\bar{\varepsilon}^{\Delta T}}{T - T_0}\right] = \frac{1}{T - T_0}\left[\int_{T_0}^{T} \bar{c}_p dT - T_0 \int_{T_0}^{T} \frac{\bar{c}_p dT}{T}\right] \tag{20}$$

Where H_A = Enthalpy of flue gases entering
The boiler,

H_B = Enthalpy of flue gases leaving the boiler
E_A = Exergy in the flue gas at the entering the boiler
E_B = Exergy in the flue gas at the exiting from the boiler

The values of H_A, H_B, E_A, and E_B are common for simple supercritical cycle, single reheat cycle and double reheat.

Without Reheat:

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.

$$m_s(W_{net}) = 1000 \text{ MW}$$
$$m_s = 1000 \times 1000 \text{ kW}/W_{net} \qquad \text{kg/sec} \tag{21}$$

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

$$m_g = m_s(h_1 - h_4)/(H_A - H_B) \tag{22}$$

Availability or Gibbs function at state point 1

$$G_1 = Es_1 = m_s(h_1 - T_o s_1) \qquad \text{kW} \tag{23}$$

Availability or Gibbs function at state point 4

$$G_4 = Ew_4 = m_s(h_4 - T_o s_4) \qquad \text{kW} \tag{24}$$

Irreversibility in the boiler is

$$I_{boiler} = E_A - E_B - (Es_1 - Ew_4)$$
$$I_{boiler} = m_g(E_A - E_B) - m_s((h_1 - h_4) - T_0(s_1 - s_4)) \quad \text{kW} \tag{25}$$

Steam Turbine:

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

$$I_{turbine} = T_0.m_s\,(s_2 - s_1) \quad \text{kW} \tag{26}$$

Condenser:

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

$$m_{cw}C_{pw}(T_{wi} - T_{wo}) = m_s\,(h_2 - h_3)$$
$$m_{cw} = m_s(h_2 - h_3)/\,C_{pw}(T_{wi} - T_{wo}) \tag{27}$$

Irreversibility in the condenser,

$$I_{condenser} = T_0(mcwCpwln(T_{wi}/T_{wo}) - m_s(s_2 - s_3)) \quad \text{kW} \tag{28}$$

Pump:
Irreversibility rate in the boiler feed pump,

$$I_{pump} = m_s\,T_0(s_4 - s_3) \quad \text{kW} \tag{29}$$

Exhaust: Irreversibility of the exhaust, $I_{exhaust} = E_B$

Single Reheat:

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.

$$m_s(W_{net}) = 1000\,\text{MW}$$
$$m_s = 1000\text{x}1000\,\text{kW}/\,W_{net} \quad \text{kg/sec} \tag{30}$$

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

$$m_s((h_i - h_6) - (h_3 - h_2)) = m_g(H_A - H_B)$$
$$m_g = m_s((h_1 - h_6) - (h_3 - h_2))\,/(H_A - H_B) \tag{31}$$
$$I_{boiler} = m_g(E_A - E_B) - m_s((h_1 - h_6) - (h_3 - h_2))$$
$$- (T_0(s_i - s_6) - T_0(s_3 - s_2)) \quad \text{kW} \tag{32}$$

Steam Turbine:

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

$$I_{turbine} = T_0.m_s((s_2 - s_1) + (s_4 - s_3)) \qquad kW \qquad (33)$$

Condenser:

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

$$m_{cw} C_{pw} (T_{wi} - T_{wo}) = m_s (h_4 - h_5) \qquad (34)$$
$$m_{cw} = m_s(h_4 - h_5)/ C_{pw} (T_{wi} - T_{wo})$$

Irreversibility in the condenser,

$$I_{condenser} = T_0[m_{cw} C_{pw} ln(T_{wi}/T_{wo}) - m_s(s_4 - s_5)] \qquad kW \qquad (35)$$

Pump:

Irreversibility rate in the boiler feed pump,

$$I_{pump} = m_s T_0(s_6 - s_5) \qquad kW \qquad (36)$$

Exhaust:

Irreversibility or exergy loss through the exhaust, $I_{exhaust} = E_B$

Double Reheat:

Mass of steam 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.

$$m_s(W_{net}) = 1000 \text{ MW}$$
$$m_s = 1000 \times 1000 \text{ kW}/ W_{net} \qquad kg/sec \qquad (37)$$

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

$$m_s((h_1 - h_8) - (h_3 - h_2) - (h_5 - h_4)) = m_g(H_A - H_B)$$
$$m_g = m_s((h_1 - h_8) - (h_3 - h_2) - (h_5 - h_4))/ (H_A - H_B) \qquad (38)$$

Steam Turbine:

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

$$I_{turbine} = T_0.m_s((s_6 - s_1) + (s_3 - s_2) + (s_5 - s_4)) \qquad kW \qquad (39)$$

Condenser:

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

$$m_{cw} C_{pw} (T_{wi} - T_{wo}) = m_s (h_6 - h_7) \tag{40}$$
$$m_{cw} = m_s(h_6 - h_7)/ C_{pw} (T_{wi} - T_{wo})$$

Irreversibility in the condenser,

$$I_{condenser} = T_0[m_{cw} C_{pw} \ln(T_{wi}/T_{wo}) - m_s(s_6 - s_7)] \quad \text{kW} \tag{41}$$

Pump:

Irreversibility rate in the boiler feed pump,

$$I_{pump} = m_s T_0(s_8 - s_7) \quad \text{kW} \tag{42}$$

Exhaust:

Irreversibility or exergy loss through the exhaust, $I_{exhaust} = E_B$

Total Irreversibility is

$$\Sigma I = I_{boiler} + I_{turbine} + I_{pump} + I_{condenser} + I_{exhaust} \quad \text{kW} \tag{43}$$

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.

$$\text{Exergy efficiency, } \eta_{11} = \frac{E_A - \sum I}{\sum I} * 100 \tag{44}$$

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,

$$\frac{I_{boiler}}{\sum I} * 100 \tag{45}$$

Fractional exergy loss in the turbine is,

$$\frac{I_{turbine}}{\sum I} * 100 \tag{46}$$

Fractional exergy loss in the condenser is,

$$\frac{I_{condenser}}{\sum I} * 100 \tag{47}$$

Fractional exergy loss in the Pump is,

$$\frac{I_{pump}}{\sum I} * 100 \tag{48}$$

Fractional exergy loss in the exhaust is,

$$\frac{I_{exhaust}}{\sum I} * 100 \tag{49}$$

Results and Discussion

This complete analysis has been written using C-language for generalized program. The author tested various live steam temperatures and pressures of the supercritical power plants, which are available in the world. The analysis has been done by considering the optimum reheat pressure ratio as 0.2. It is observed from figure 5 shows that with the increase in pressure for a particular temperature cycle efficiency increases for a supercritical cycle with no reheat, single and double reheat At particular pressure at turbine inlet, the cycle efficiency of supercritical cycle without reheat, single reheat and double reheat has been analyses at various turbine inlet temperature. Also with the consideration of a double 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 and single reheat. It is also observed that the increase in cycle efficiency is very less when compared to the increase in turbine inlet pressure limits. At 325 bar / 700°C, the cycle efficiency without reheat/single reheat/double reheat is 43.44% / 46.61% / 49.39%. It is observed from Figure 6, 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 single reheat and double reheat, but the increase in efficiency is high in double reheat when compared to a supercritical cycle without reheat and single reheat. At 350 bar / 650°C, the cycle efficiency without reheat/single reheat/double reheat is 42.75% / 45.53% / 47.96%.

Fractional exergy loss in the turbine is,

$$\frac{I_{turbine}}{\sum I} * 100 \tag{46}$$

Fractional exergy loss in the condenser is,

$$\frac{I_{condenser}}{\sum I} * 100 \tag{47}$$

Fractional exergy loss in the Pump is,

$$\frac{I_{pump}}{\sum I} * 100 \tag{48}$$

Fractional exergy loss in the exhaust is,

$$\frac{I_{exhaust}}{\sum I} * 100 \tag{49}$$

Results and Discussion

This complete analysis has been written using C-language for generalized program. The author tested various live steam temperatures and pressures of the supercritical power plants, which are available in the world. The analysis has been done by considering the optimum reheat pressure ratio as 0.2. It is observed from figure 5 shows that with the increase in pressure for a particular temperature cycle efficiency increases for a supercritical cycle with no reheat, single and double reheat At particular pressure at turbine inlet, the cycle efficiency of supercritical cycle without reheat, single reheat and double reheat has been analyses at various turbine inlet temperature. Also with the consideration of a double 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 and single reheat. It is also observed that the increase in cycle efficiency is very less when compared to the increase in turbine inlet pressure limits. At 325 bar / 700°C, the cycle efficiency without reheat/single reheat/double reheat is 43.44% / 46.61% / 49.39%. It is observed from Figure 6, 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 single reheat and double reheat, but the increase in efficiency is high in double reheat when compared to a supercritical cycle without reheat and single reheat. At 350 bar / 650°C, the cycle efficiency without reheat/single reheat/double reheat is 42.75% / 45.53% / 47.96%.

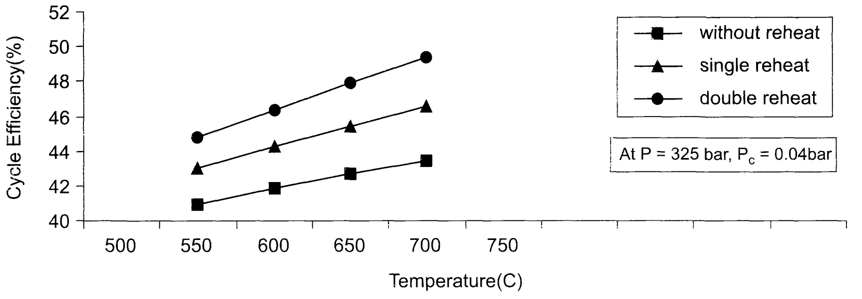

Figure 5: Effect of Cycle Efficiency on Various Turbine Inlet Temperatures

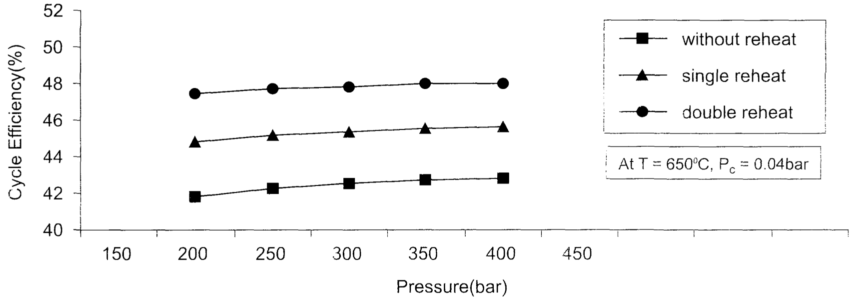

Figure 6: Effect of Cycle Efficiency on Various Turbine Inlet Pressure

Figure 7 presents the variations of exergy efficiency with increase at particular pressure at different turbine inlet temperature of non-reheat, single reheat and double reheat. It is obvious that exergy efficiency increases with increase in pressure at a particular turbine inlet temperature for a supercritical cycle without reheating, single and double reheat. The exergy efficiency also increases for a supercritical cycle with double reheat and the increase in efficiency is high when compared to a cycle without reheat and single reheat. At 325 bar / 700°C, the exergy efficiency without reheat/single reheat/double reheat is 57.72%/63.70%/ 65.90%. Figure 8 predicts 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 reheat, single and double reheat. The increase in exergy efficiency is high for cycles with double reheating when compared to a cycle without reheat and single reheat. At 350 bar / 650°C, the exergy efficiency without reheat/single reheat/double reheat is 55.46%/62.93%/65.96%.

The variation of fractional exergy loss of different components at different temperatures without reheating, single and double reheat have been shown in Figure 9. It is observed that for a boiler the decrease in fractional exergy loss with

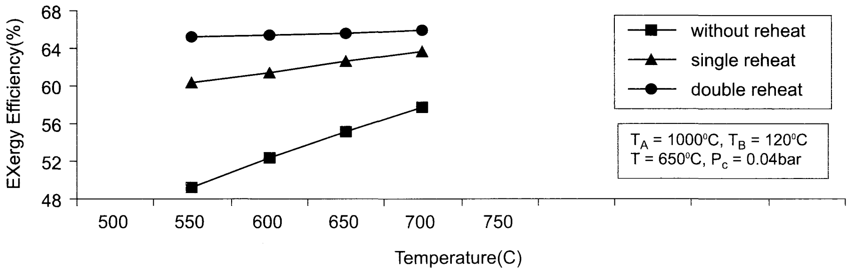

Figure 7: Effect of Exergy Efficiency of the Various Turbine Inlet Temperatures

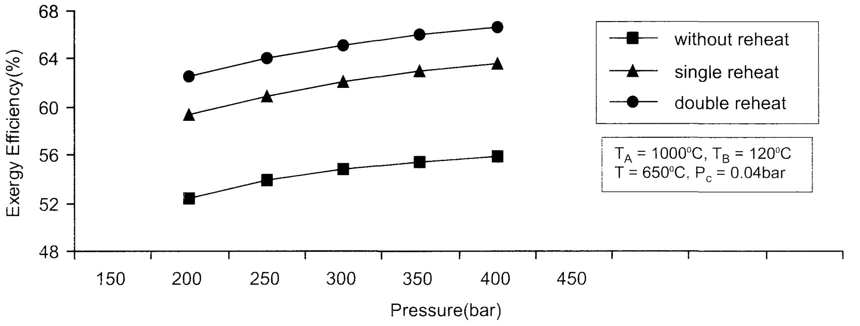

Figure 8: Effect of Exergy Efficiency of the Various Turbine Inlet Pressures

increase in temperature is high for a cycle with double reheat when compared to a cycle without reheat, single reheat and double reheat. At a Pressure/Temperature of 325 bar/650°C the fractional exergy loss in the boiler with no reheat/single reheat/double reheat is 64.25%/62.99%/62.62% and for the turbine fractional exergy loss is 25.04 %/26.46%/26.51% and for the condenser and the pump are 5.1%/4.11%/4.06% and 1.08%/1.01%/0.91%. Figure 10 shows the variation of fractional exergy loss of different components at different pressures with and without reheating. 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 without reheat, single and double reheat.

Figure 11 shows the exergy efficiency on different flue gas inlet temperature. The author analyses the flue gas temperature inlet to the boiler are 800 to 1100°C. It is observed from the result the exergy efficiency increases with increase in flue gas inlet temperature, but there will be a limited in materials to withstand the higher boiler temperature. Figure 12 depicts the exergy efficiency on different flue gas outlet temperature. The author analyses the flue gas temperature outlet to the boiler are 80 to 150°C. It is observed from the result the exergy efficiency increases with decrease in flue gas outlet temperature of double reheat than the

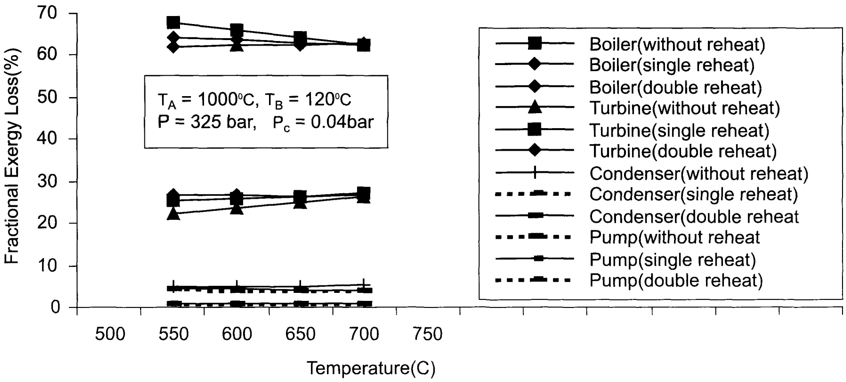

Figure 9: Effect of Turbine Inlet Temperature on Fractional Exergy Loss of Different Components

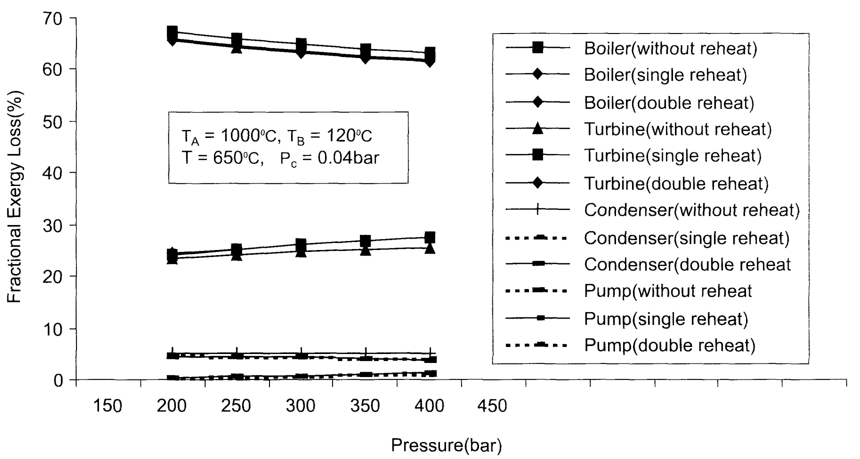

Figure 10: Effect of Turbine Inlet Pressure on Fractional Exergy Loss of Different Components

non reheat and single reheat. The variations of total exergy loss with temperature for different values of turbine inlet pressures without reheat, single reheat and double reheat are presented in Figure 13. The total exergy loss is decreasing with increase of pressure and temperatures of without reheat, single reheat and double reheat of the different components in the supercritical cycle. It is observed that total exergy loss is less with double reheat when compared to with and without reheating.

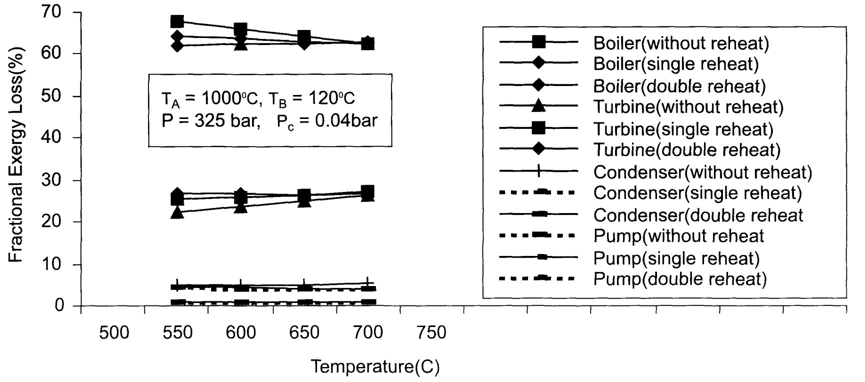

Figure 9: Effect of Turbine Inlet Temperature on Fractional Exergy Loss of Different Components

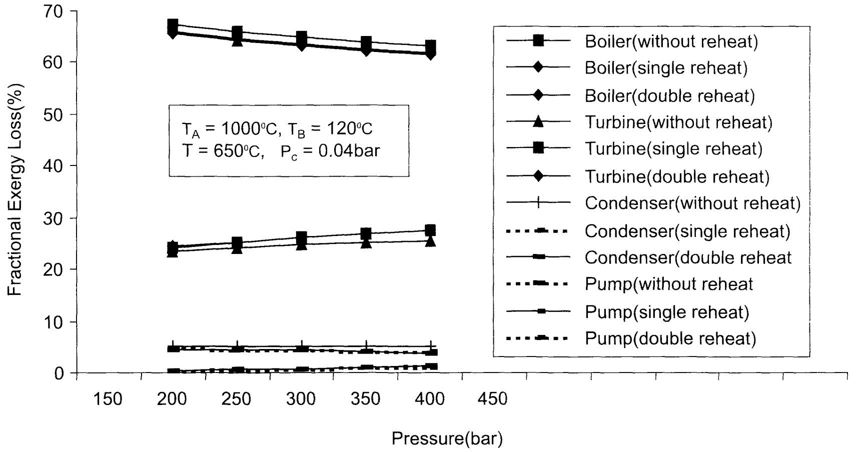

Figure 10: Effect of Turbine Inlet Pressure on Fractional Exergy Loss of Different Components

non reheat and single reheat. The variations of total exergy loss with temperature for different values of turbine inlet pressures without reheat, single reheat and double reheat are presented in Figure 13. The total exergy loss is decreasing with increase of pressure and temperatures of without reheat, single reheat and double reheat of the different components in the supercritical cycle. It is observed that total exergy loss is less with double reheat when compared to with and without reheating.

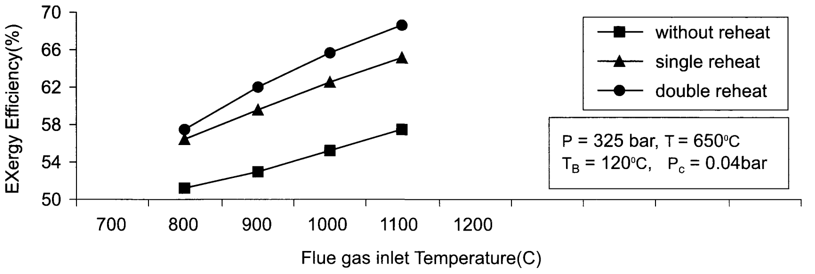

Figure 11: Effect of Exergy Efficiency on Different Flue Gas Inlet Temperature

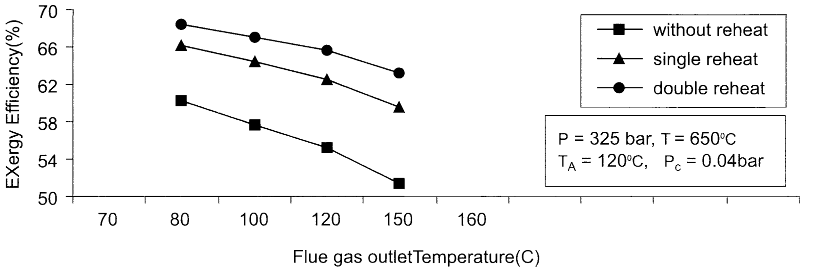

Figure 12: Effect of Exergy Efficiency on Different Flue Gas Outlet Temperature

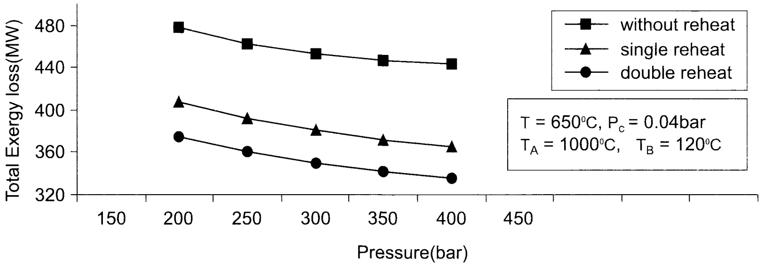

Figure 13: Effect of Total Exergy Loss on Different Turbine Inlet Pressure

Conclusion

This paper analyzes the supercritical cycle without reheat, single and double reheats from both cycle and exergitic efficiencies. The total exergy loss and fractional exergy loss are determined for the cycle without reheat, single reheat and double reheat. It is found that the cycle efficiency is high in double reheat than the non-reheat and single reheat supercritical cycle. It is also concluding that exergy efficiency is high in reheat than non-reheat supercritical cycle. The cycle efficiency and exergy gains that can typically be achieved through the use of higher steam pressures and temperatures on a non-reheat, single and double reheat supercritical cycle. The other factors, which affect the cycle efficiency and exergy efficiency, are the number of reheats, single and double, the condenser pressure, heat rate, steam rate and work ratio has been studied. The flue gas inlet and outlet temperature of the boiler has also been studied. Fractional exergy losses of all the components in the cycle is determined and compared with and without supercritical cycle.

References

[1] Nag, P.K. and Gupta, A.V.S.S.K.S. (1998). Exergy analysis of the Kalina Cycle, Applied Thermal Engineering Journal, Vol. 18, No. 6, 427-439.

[2] Y.M. El-sayed and M. Tribus (1995). A Theoretical Comparison of Rankine and 4Kalina cycles, ASME publication, Vol. 1.

[3] Kotas, T.J. (1985). The Exergy method of Thermal Power analysis, Butterworth.

[4] Kotas, T.J., et al.,"Nomenclature for exergy analysis" ASME Journal of Power and Energy, Proc. Instn. Mech. Engrs Vol. 209, 275-280.

[5] ISI steam tables in SI Units SP – 26 –1983.

[6] Bejan, A., Tsatsaronis, G. and Moran, A. (1996). Thermal Design and Optimization, Wiley, New York.

[7] Nag, P.K. (1995). Power plant engineering, 2nd Ed., Tata Mc Graw-Hill, New York.

[8] Habib, M.A., Said, S.A.M. and Al-Zaharna (1999). Thermodynamic optimization of reheat regenerative thermal-power plants Journal of Applied Energy 63, 17-34.

[9] Horlock, J.H., et al. (2000). Exergy Analysis of Modern Fossil-Fuel Power Plants, J. of Engineering for Gas Turbines and Power, ASME, Vol. 122, 1-7.

[10] Michael, J. Moran and Howard, N. Shapiro (1988). Fundamentals of Engineering Thermodynamics. New York: Wiley.

[11] Cengel, Y. A. and Boles, M.A. (1989). Thermodynamics, an Engineering Approach. Singapore: Mc-Graw Hill International Editions.

[12] Kalina, I.A. (1984). Combined cycle system with Novel Bottomming cycle, Trans. of ASME, J. of Engineering for Gas Turbine and Power", Vol. 106, 737-742.

[13] Perry, R.H. and D. Green, Perry's chemical Engineers Hand Book, 7th Ed., Mc Graw-Hill.