STEAM TURBINE FOR POWER PLANT UNIT WITH SUPERCRITICAL STEAM PARAMETERS

P.Ravindra Kumar[1] , Hussain shaik[2] , P.Aruna[3], J.Subba Reddy[4]

[1 4] *Associate professor, Department of Mechanical Engineering, LakiReddy Bali Reddy College of Engineering, Andhra Pradesh, India*

[2 3] *Assistant professor, Department of Mechanical Engineering, LakiReddy Bali Reddy College of Engineering, Andhra Pradesh, India*

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Abstract - *Problems arise when changes to a new level of super critical steam parameters of power units are considered by considering the steam turbine losses and pipe losses. The dependence of the efficiency of the unit on the isentropic efficiencies of individual turbine stages i.e. High Pressure Turbine (HPT), Intermediate Pressure Turbine (IPT), and Low Pressure Turbine (LPT) stages in a layout is considered. From literatures and experience of best manufacturers the efficiency of a unit can be brought to the level of the best worthy. The computations from MS Excel show that the efficiency of the turbine plant is 49 %, that of the boiler is 93.66 %, and that of the power unit is 46%.*

Key Words: Supercritical steam parameters, Isentropic Efficiency of different Turbine Stages, Feed Water Heaters (FWH)

1. INTRODUCTION

Present work focuses on advancement of power generating units of thermal power plants (TPP) was intensified in economically developed countries in the early 1990s and developed, in particular, by transfer to supercritical steam parameters of 270 - 310 bar and 580 - 650°C. This was promoted by successes in new materials and design point of view, as well as by increased competition in the fuel market and by the tendency for reducing the cost of nature-protection technologies and emissions of CO_2. These studies resulted in the creation of coal-fired power plants with elevated efficiency (40 - 47%) in these countries in the last decade of the last century [1]. The service life of the active domestic coal-fired power generating units will be exhausted fully in the second half of the present decade. In order to reconstruct the existing plants with replacement of equipment and to build new Thermal power plants, India will require coal-fired units employing advanced engineering technology solutions that raise the efficiency of operation, reliability, and more power generation. The efficiency of such power units should be raised to 45 - 46% as in foreign countries against the 36% to 40 % in Indian Climatic Conditions at the active coal-fired units with standard supercritical steam parameters [2, 3].

Table1: Design Parameters of Supercritical Power Plant [4]

Power, MW	712
Design of turbine	HPT + IPT+ 2LPT
Design of boiler	Tower Type
Temperature of live and reheat steam, (°C)	581/593
Pressure of live steam, (bar)	265
Temperature of feed water, (°C)	300
Pressure in condenser, (bar)	0.05
Auxiliaries of the unit, (%)	6.2
Efficiency of turbine plant gross/net, (%)	52/49
Efficiency of boiler, (%)	93.0
Specific consumption of coal equivalent, g/(kW-h)	274
Net efficiency of the generating unit, (%)	45.0

Collecting the data from published International Journals that the net efficiency of power units of the new generation amounts to 43 - 46% except for several units with even higher net efficiency (49 - 53%), which operates at very low temperature of the cooling water that arrives at condensers from rivers or cooling towers at a temperature of 15ºC - 20ºC and therefore have a very low pressure in the condenser (0.05 bar) and a high moisture content in the last stage (up to 15%) [5].This requires lower reheat pressure in the case of single reheat or the use of second reheat. Most of the power units operating on solid fuel. Power unit in problem have elevated initial temperature of live steam and reheat [6]. The temperature of 580°C has become virtually standard in Europe, and 600°C is typical for Japan [8]. Most power units of the new generation have been designed for an initial pressure of 240 - 260 bar. The majority of power units in India have a capacity of 210 -500 MW, which, makes it possible to use a moderate number of Low-Pressure Turbines (LPT) ensures a high enough efficiency of the flow-through part of the turbine. Most power units imported from Japan have a capacity of 1000 MW [13]. Finally, we should mention the tendency of raising the temperature of feed water to up to 310-340°C for supercritical units as against subcritical power plant units getting 150-200°C. World is looking for the new level of steam parameters as with the advent of new technological materials, it is necessary that the efficiency of the passage cover the cost of conversion of the new equipment. Since the increase in steam parameters is commonly connected with higher cost of the equipment, if the output of the unit does not increase simultaneously, but the output of the unit consumes low specific fuel consumption (SFC) [14]. In the global power engineering a new level of parameters is usually been chosen so that the specific fuel consumption decreases by at least 4% except for the cases where the parameters are increased exceptionally out of the desire to raise the unit capacity of the generating unit, for which reason the only measure taken is an increase in the steam pressure. Since further increase in efficiency becomes more and more challenging, it can be expected that turning to a new higher level of parameters will occur with a difference in the specific consumption of heat at old and new units of much less (2 - 2.5%) than the 4% realized in turning to generating units producing 210 MW with standard Supercritical Steam Parameters. It should be noted that the maximum temperature of live steam is limited to 600°C. This is obtained by the fact that at a higher temperature of steam superheating requires the use of austenitic steels in the thick-wall parts of conduits and exhaust elements of the turbine.

2. Mathematical Modeling of Steam Turbine Losses

Mathematical equations applied for calculating the steam losses in steam turbines at supercritical conditions [15]:

1. Regulating valve Losses:

$$(\Delta h_{isentropic})_{Regulating\ valve} = (0.03 \text{ to } 0.05)\text{x Intial Pressure}$$

2. Loss of energy in Steam Turbine Nozzle:

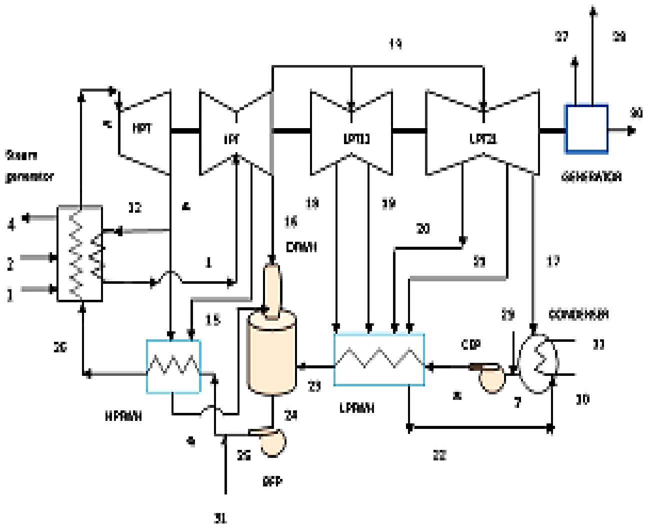

Fig. 1: Schematic layout of a coal based power plant unit with Deaerator (Open Feed Water Heater (OFWH))

$$\frac{V_t^2(1-\eta_n)}{2000}$$

Where V_t = Theoretical velocity

3. Loss of energy in moving blades per stage:

$\frac{V_{r1}^2(1-K^2)}{2000}$, where K varies between 0.8 to 0.95 and that depends upon the blade height and deflection angle

4. Power absorbed by Disc Friction:

$$(P_{loss})_{df} = m(\Delta h_{loss})_{df} = VD^2\left(\frac{u}{1000}\right)^2 \rho$$

$$\log_{10}\left(\frac{V}{0.735}\right) = 1.277 - 0.2\log_{10}\left(\frac{Du\rho}{\mu}\right)$$

$$\mu = (0.173t + 40.68)x10^{-7}$$

Where D= Diameter of disc
U= Peripheral velocity of disc
P= density of fluid in which disc is rotating
t= temperature

5. Power absorbed by Disc Friction and blade windage losses in Steam Turbine:

$$(P_{loss})_{df,bw} = m(\Delta h_{loss})_{df,bw}$$

$$= \lambda\left[1.07D^2 + 0.61Z(1-\varepsilon)^{1.5}\right]D\left(\frac{u}{1000}\right)^2 \rho$$

Stodola's equation
Where λ=1.15 for air and highly superheated steam
D= mean blade ring diameter
Z= number of velocity stages on the disc
ε= Degree of partial admission of steam
λ = fraction of whole circumference to which the nozzle arc is extended

$(P_{loss})_{df,bw} = \beta D^4 N^3 l \rho \text{ x } 10^{-10}$ Forners equation

Where
l= height of blades
B=2.80 for 3 row disc

6. Loss of heat content in the constriction of steam turbine is given by

$$\Delta h_{leak} = \frac{m_{leak}}{m}(h_1 - h_{2a})$$

Where m= mass flow rate of steam through given steam turbine Kg/s

m_{leak} = mass flow rate of steam through given clearance space in turbine Kg/s

h_1 =enthalpy of steam before diaphragm (KJ/Kg)

h_{2a} =actual enthalpy of steam after the diaphragm

$$m_{leak} = A_g \sqrt{\frac{P_1^2 - P_2^2}{ZP_1 v_1}}$$

A_g =Area of the annular gap= $C_c\pi\varepsilon D$

Where
Cc= Coefficient of constriction
ε= Radial clearance or width of a gap, m
D= mean diameter of the packing strip, m

v_1 = specific volume of steam before diaphragm m³/Kg

P_1 & P_2 = steam pressure before and after diaphragm in N/m²

For superheated steam $P_c = \dfrac{0.85P_1}{\sqrt{(Z_c + 1.5)}}$

Zc= Number of labyrinth chambers in steam turbines

7. Heat loss through radial clearance of the reaction turbine may be given by

$$h_{leak} = \frac{\delta_r}{l\sin\alpha_1}(h_1 - h_2)$$

δ_r = width of the clearance in steam turbine, mm

l= height of the guide blade angles, mm

h_1 = enthalpy of superheated steam before guide blades

h_2 = enthalpy of superheated steam after moving blades

α_1 = guide blade angle

8. Loss due to wetness of steam

$$\Delta h_{wetness} = (1-x)\Delta h_2$$

Δh_2 = heat drop utilized in a turbine stage considering all the losses

$$x = \frac{x_1 + x_2}{2}$$

Where x_1 & x_2 dryness fraction of steam before the nozzles and after the moving blade stage.

9. Pressure drop in the exhaust piping of condensing turbines:

$$P_2 - P_{2p} = \lambda\left(\frac{V_p}{100}\right)^2 P_{2p}$$

P_2 = Pressure of steam after the blades, bar

P_{2p} =Pressure of steam in the exhaust piping, bar

V_p = Velocity of steam in the exhaust pipe, m/s

λ = Coefficient =0.07 to 0.1

3. Results & Discussions:

a) Live steam pressure:

Increase in the initial temperature should be accompanied by raising the pressure of steam, because this is accompanied by increase in the thermal efficiency of the cycle. It should be taken into account that supercritical pressure is appropriate only at a high output. For example, at the same steam flow rate and the same temperature T= 580°C, increase in the pressure from 221 to 300 bar means one-third decrease in the specific volume of steam and hence in the heights of the blades of the first stages. For a Unit Power = 300 MW and these parameters the first stage has a height of 15 -20 mm at throttle steam distribution, and leakages through the front seal increase by 25%. At throttle steam distribution and an output of 300 MW the leakages amount to 0.35% of the consumption of live steam. In the case of nozzle steam distribution these values do not differ much from the mentioned ones. Thus, at low pressure turbine outputs, increase in the pressure and the respective decrease in the volume flows of steam at the inlet to the turbine make it necessary to supply steam to the regulating stage with a low degree of dryness fraction and to use short blades of the first stages of HPT. This lowers the internal relative efficiency of the high-pressure Turbine (HPT), and the increase in pressure does not lead to the required increase in the efficiency. A preliminary estimation has shown that the pressure of live steam at the outlet from the boiler should be at a level P = 290 bar (at the inlet to the turbine it should be 280 bar) for an output of 500 MW.

b) Choice of reheat pressure:

From recent International Journal publications on the use of steam reheat shows that the reheat pressure at active turbine plants usually amounts to 0.15 -0.25 to the initial pressure [12]. In other countries Supercritical Steam Parameters (SSP) generating units with a capacity of 300 MW the reheat steam pressure is 45–60 bar. Computation of variants with reactive blading of HPT and IPT at variable pressure in the cold reheat conduit have shown the presence of a flat efficiency optimum in the 35 - 45 bar pressure range at the outlet from the HPT. Preliminary estimates from the standpoint of maximum efficiency of a power unit shown that the steam pressure at the outlet from the HPT can be chosen to be 40 bar and that at the inlet to the IPT of the turbine it should be 38 bar.

c) Number of reheats:

From the standpoint of economic parameters it would be the most beneficial to develop a power unit with two reheats. Results from International publications shown that the introduction of one more reheat saves 1.2 - 1.5% fuel [11]. This creates the difficulties with assembly of the second reheater in the boiler.

Table 2: Efficiency of the (Turbine, Unit) & Output Power in a layout with and without deaerator

Component	Without Deaerator	With Deaerator
Efficiency of Turbine (%)	48.8	49.06
Efficiency of Unit (%)	45.74	45.97
Output Power (MW)	563.28	565.92

Table 3: Turbine Efficiency, Unit Efficiency and Output Power in a layout as the function of Steam Pipe resistance

Steam Pipe Resistance (%)	Turbine Efficiency (%)	Unit Efficiency (%)	Power Output (MW)
2.5	49.17	46.05	566.67
3.0	49.13	46.01	566.67
3.5	49.09	46.05	566.52
4.0	49.13	46.01	566.52
4.5	49.09	46.01	566.52
5.0	49.09	46.01	566.38
5.5	49.09	45.94	566.38
6.0	49.06	46.01	565.94
6.5	49.06	46.01	565.80
7.0	49.06	46.01	565.51
7.5	49.06	46.01	565.51
8.0	49.02	45.91	565.22
8.5	49.06	45.91	565.07
9.0	49.02	45.87	564.93
9.5	49.02	45.91	564.49
10.0	48.95	45.83	564.49

d) Temperature of feed water:

The temperature of feed water is one more important factor for determining the efficiency in supercritical thermal power plant units. In theory of heat cycles that every turbine plant is characterized by some thermodynamically optimum feed water temperature at which the efficiency of the plant is the highest. The actual temperature of feed water is taken to be lower than the thermodynamically optimum one. This makes it possible to lower the cost of the regeneration system but decreases the efficiency. There exists an optimum, which is determined by simultaneous computation of the boiler flow diagram (optimization of the temperature of exhaust gases) and of the turbine plant (feed water temperature).

Table 4: Increase in the efficiency of a turbine plant due to improving the parameters of thermodynamic cycle is presented below for 712 MW.

S.No	Steam Parameter Variables	Relative Increase in efficiency
1	Increase in the temperature of live steam	0.02% for 1°C
2	Increase in the pressure of live steam	0.01% for 10 bar
3	Increasing the reheat temperature	0.01% for 1°C
4	Decrease in the condenser Pressure	1% for 10 bar
5	Increasing the feed water temperature	0.02% for 1°C
6	Use of second reheat	1.2%

Increase in the steam parameters from P = 240 bar, T = 540 ^{0}C to the values discussed (P = 300 bar, T = 600°C) increases the efficiency of the unit by about 3.5% for one component of coal-fired power units. The other components are the improved efficiencies of the boiler, turbine, flow diagram, and auxiliary equipment. An important component of the work is the creation and testing of a tool that would make it possible to analyze reliably the effect of various engineering solutions on the efficiency of the unit as a whole. This will be especially important for the choice of actual equipment for optimizing the choice of engineering solutions for components of the power unit. The simulation software EXCEL 2007 for designing a power unit with the use of a mathematical model tool. In particular, these computations allowed determining that the transition from Low Pressure feed water heaters Nos. 1 to 2 to deaerator gives a 0.16% gain in the efficiency of the power unit. The efficiency of the unit increases with decrease in the steam pressure on the conduit and with increase in the efficiency of the feed-pump turbine. However, an electric drive is the most economical variant. The optimum pressure on the Intermediate-pressure turbine is 2.58 bar and has been used in the development of engineering solutions for designing the flow-through part of the LPT. The pressure in the condenser affects the efficiency of the power unit. At pressures lower than 0.029 bar the moisture content in the last stage increases above the permissible 12 %. We chose a conventional pressure of 0.0343 bar.

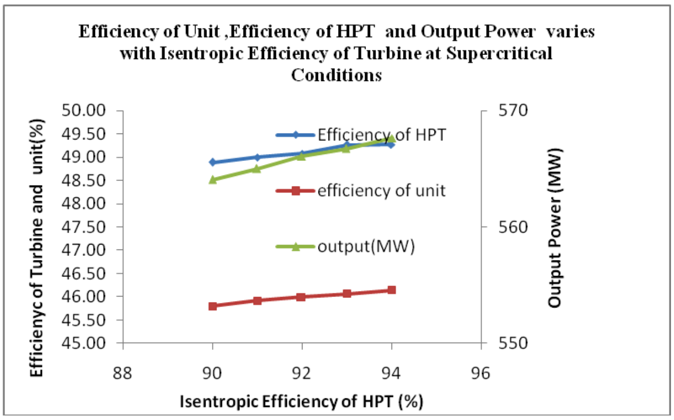

Fig. 2 : Efficiency of High Pressure turbine (%) and Output Power (MW) as a function Of Isentropic Efficiency of HPT

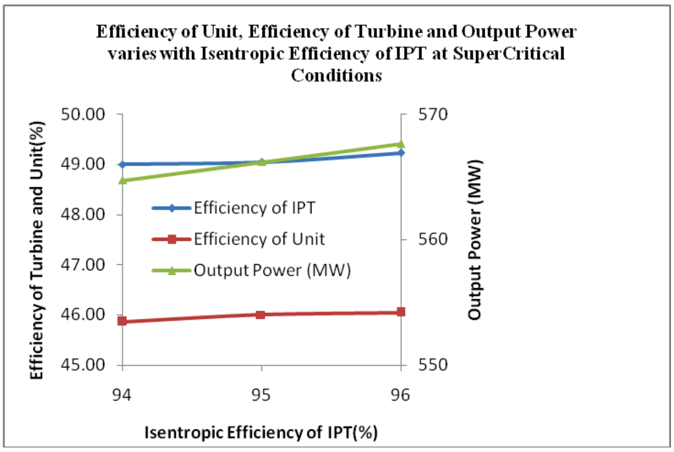

Fig.3 : Efficiency of Intermediate Pressure turbine (%) and Output Power (MW) as a function of Isentropic efficiency of IPT.

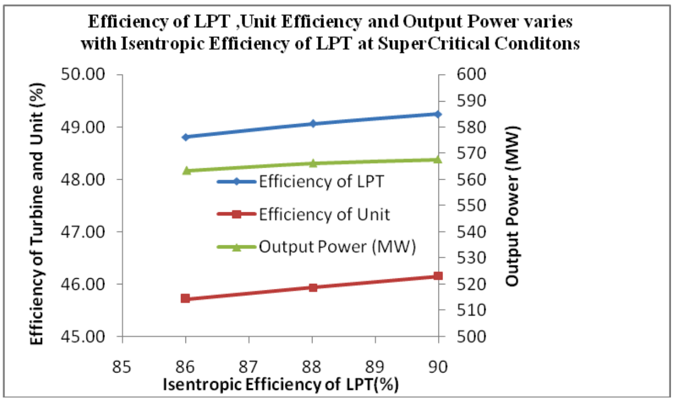

Fig.4 : Efficiency of LP turbine (%) and Output Power (MW) as a function of Isentropic Efficiency of LPT

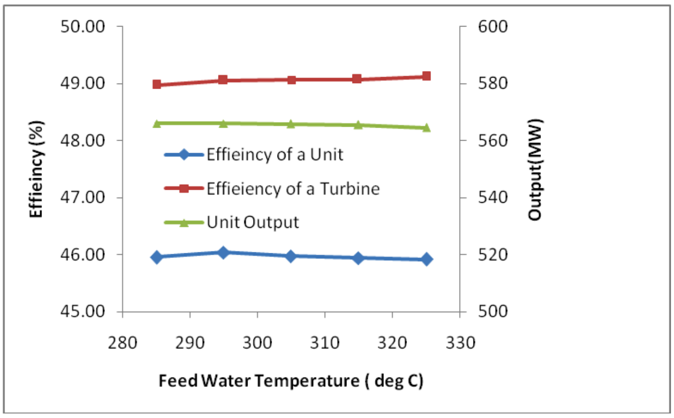

Fig.5. Efficiency of LPT(%), Unit Efficiency and OutputPower (MW)as a function of Feed Water Temperature (^{0}C)

2. CONCLUSIONS:

The simulation software gives the results of the efficiency of a turbine, efficiency of a unit & output of the Unit Power with & without deaerator as shown in Table 2. Table 3 shows that as the steam pipe resistance increases, there will be a decrease in efficiency of a turbine, Efficiency of a unit and Output of the Unit Power and for analyzing the effect of Isentropic efficiencies of HPT, IPT, and LPT on the efficiency of the power unit and established a relation that the efficiency of the unit was more dependent on the efficiency of the LPT and IPT than on that of the HPT. Table 4: shows the potential growth in efficiency values when there will be rise in values of steam parameters .Finally, the software simulation tool was used for estimating the effect of the use of what is known as "turbine economizer" in the boiler, which was connected with respect to feed water in parallel to the high-pressure heaters (HPH) of the turbine plant, on the efficiency of the power unit. It turned out that the effect of the economizer on the efficiency of the power unit depended on the kind of the fuel fired and on its position in the convective gas conduit of the boiler. Computations show that the optimum feed water temperature from the standpoint of the efficiency of the power unit depends on the kind of fuel and on the design of the tail surfaces of the boiler and that it is necessary to perform joint optimization of the boiler and of the turbine. The computations show that the efficiency of the turbine plant should be 49 %, that of the boiler should be 93.66 %, and that of the power unit should be 46%. At the present time the use of the measures mentioned allows the following efficiency of the Turbines: 0.92 for HPT, 0.94 for IPT, and 0.9 for LPT, which corresponds to the world level.

In order to get good solutions for supercritical steam parameters, we should employ the latest advances in world steam turbine engineering, namely

1. Three-dimensional design of the flow-through parts;
2. Reactive blading of the HPT and IPT;
3. Use of liquid-metal seals for valve plungers;
4. Throttle steam distribution;
5. Mixed-flow steam discharge from HPT and IPT with tangential admission of steam;
6. Cellular seals;
7. Modified exhaust nozzles Turbines.
8. With the use of RSD cooling systems the reheat temperature can be increased to 620°C at the outlet from the boiler

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BIOGRAPHIES

Post graduated in Mechanical Engineering, (Thermal Engineering) and has been working as an Associate professor for about two years in LBRCE, AP, India.

Graduated in Mechanical Engineering and has been working as an assistant professor for more than two years now in LBRCE, AP, India.

Post graduated in Mechanical Engineering (Thermal Engineering) and has been working as an assistant professor for more than two years in LBRCE, AP, India.

Post graduated in Mechanical Engineering, and has been working as an associate professor for about one year in LBRCE, AP, India.

