International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 3 Issue 10, October- 2014

Analysis of Ultra Super-Critical Power Plant and Comparison Based on Reheating System

Abhishek Sarkar
Department of Mechanical Engineering, Delhi Technological University, Bawana Road, New Delhi, 110042, India

Dipak Kumar Sarkar
Research and Development Division, Engineers India Limited, Gurgaon, 122001, India

Abstract: - An investigative study has been made on ultra super-critical power plant models. The steam pressure is kept above the critical limit, allowing flash phase change to occur from water to steam. Thermal properties have been calculated and used for performance analysis. The thermal efficiency, net power output and exergetic efficiency are calculated to estimate the optimum pressure for plant operation. The mass flow rate of steam and maximum steam temperature have been kept constant in the two models. A comparative study has been conducted to establish the benefits of reheating system in the power plant model based on improvements in thermal efficiency and power output. Finally, the paper concludes that the reheat based ultra super-critical power plant is a more efficient and higher power generating system as compared to basic thermal power plants.

Keywords:Ultra super-critical,Power plant,Reheat,Exergy,Second Law Analysis

I. INTRODUCTION:

In the present day scenario, there is an advent of advanced communication and technology. The world has become a global family, and the need of energy has incremented exponentially in the past few decades. The need in the consumer market for higher and higher demand of power had led to the development of advanced technologies for power generation. In India, for instance, there is a need of 172,286 MW of energy by June, 2014 and the per capita energy consumption was 612 KW. The electricity sector in India supplies the world's 6th largest energy consumer, accounting for 3.4% of global energy consumption by more than 17% of the global population. Due to the fast paced growth of Indian economy there has been an average increase of 3.6% in the energy demand per annum over the last 30 years. Thus, it can be concluded that the demand surpluses supply, causing a situation of energy crisis.

There is an immediate need of paradigm shift from the conventional methodologies of power generation to methods that yield higher power at better efficiency. The conventional sub-critical power plants produce low output at approximate of 30% efficiency, which means that it consumes tremendous amount of fuel to give only thirty percentage of it as output. According to Yang et al. [1] with such coal consumption, controlling the pollutants emissions is an unavoidable topic. To achieve sustainable development, the focus on power system efficiency moves from analysis of just economic benefits to environmental efficiency studies that assess both economic benefits and carbon emissions. Thus, new technologies such as CO_2 capture have the potential to significantly reduce pollutant emissions. However, industrial tests and techno-economic analysis of CO2 capture in a demonstrating coal-fired power station show that the electricity purchase price increases by 29% with CO_2 capture. In fact, currently new technologies (not only CO_2 capture and sequestration) for reducing pollution from power generation regarded too risky or too expensive. Thus, the best alternative for reducing emissions is still to increase the plant efficiency. In this context, supercritical and ultra-supercritical (USC) coal- fired power (CP) generation is regarded to be significant. The technology currently achieves a plant efficiency of around 45% at pressure over the steam critical pressures. Thus, with proper approaches for reducing gross heat losses, for example, reducing unburned combustible loss and advanced waste heat water recovery technology the lossesand pollutant concentration can be further reduced.

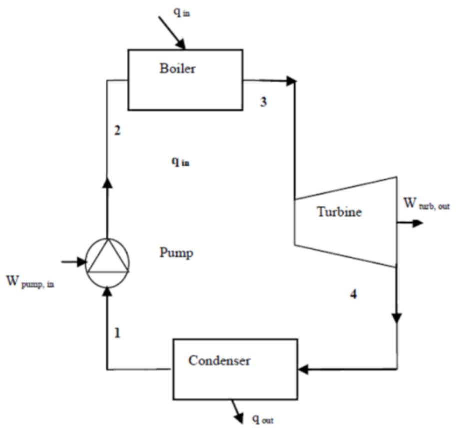
According to Boehm [2] the importance of developing thermal systems that effectively use energy resources such as oil, natural gas, and coal is apparent. Effective use is determined with both the First and Second Laws of thermodynamics. Energy entering a thermal system with fuel, electricity, flowing streams of matter, and so on is accounted for in the products and by-products. Energy cannot be destroyed - a First Law concept. The idea that something can be destroyed is useful. This idea does not apply to energy, however, but to exergy - a Second Law concept. Moreover, it is exergy and not energy that properly gauges the quality (usefulness) of, say, 1kJ of electricity generated by a power plant versus one kilojoule of energy in the plant cooling water stream. Electricity clearly has the greater quality and, not incidentally, the greater economic value. For industries where energy is a major contributor to operating costs, an opportunity exists for improving competitiveness through more effective use of energy resources. This is a well-known and largely accepted principle today.

In the 1970s the method of Second Law optimization or entropy generation minimization (EGM) emerged as a distinct field of activity at the interface between heat transfer, engineering thermodynamics, and fluid mechanics. The position of the field is illustrated in the first book ever published on this method (Bejan 1982) [3]. The method relies on the simultaneous application of principles of heat and mass transfer, fluid mechanics, and engineering thermodynamics, in the pursuit of realistic models of processes, devices, and installations.

A critical early step in the design of a system is to pin down what the system is required to do and to express these requirements quantitatively, that is, to formulate the design specifications. A workable design is simply one that meets all the specifications. The paper deals with the thermal design of ultra super-critical power plant based on second law analysis and comparison of two models depicting the advantages of reheating.

II. MODEL:

Coal based thermal power plant works based on Rankine power cycle [4]. The ideal vapor power cycles have many impractical concepts associated with the Carnot cycle that can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser. The cycle that results is the Rankine cycle, which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibility



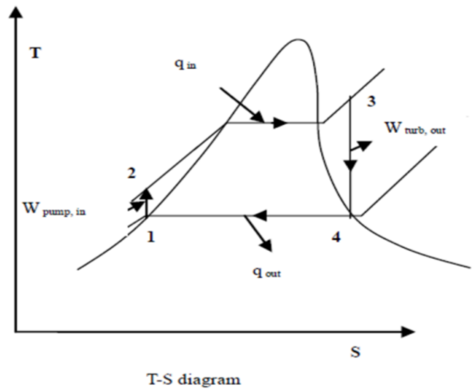
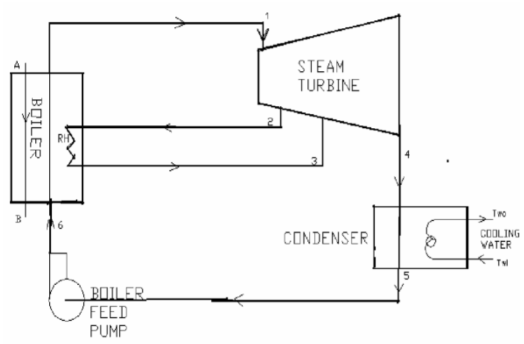


Fig1(a,b). Basic steam circuit and corresponding T-s Diagram

and consists of the following four processes:

1-2 Isentropic compression in a pump;
2-3 Isobaric heat addition in a boiler;
3-4 Isentropic expansion in a turbine;
4-1 Isobaric heat rejection in a condenser

Reheating process can be incorporated into the steam power plant design in order to get higher work output and efficiency. Reheating process means to take out the partially expanded steam from the high pressure turbine and heat it (isobaricaly) back to the turbine inlet temperature and then send it to the low pressure turbine for further expansion to condenser inlet pressure.



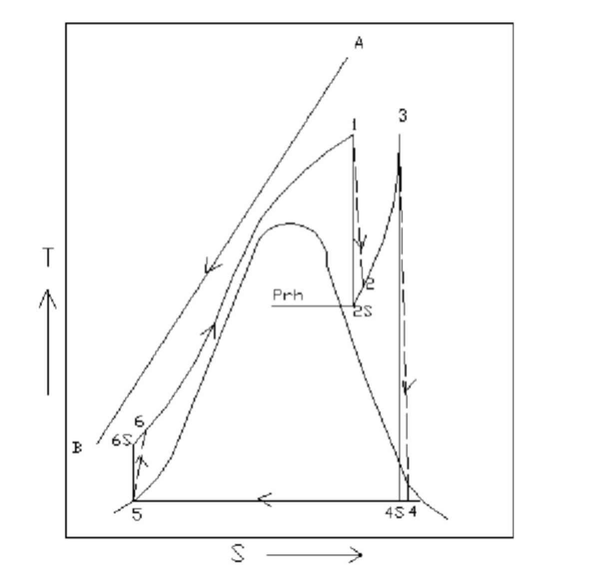


Fig2(a,b). Steam circuit with reheat and corresponding T-s Diagram

The process is as follows:
1-2s Isentropic expansion in HP Turbine
2-3 Isobaric Reheat in Boiler
3-4s Isentropic expansion in LP Turbine
4-5 Isobaric condensation in Condenser
5-6s Isentropic compression in Pump
6-1 Isobaric Heat addition in Boiler

The power plant is run under ultra super-critical pressure conditions. The boiler is such designed as to incorporate variable steam pressures which are above critical pressure of steam ($p_{critical}$=221bar). The mass flow rate of the power plant is kept at constant condition ($\dot{m}$=311kg/s) and the condenser pressure is also fixed (p_{cond}=0.06bar). The cooling water used for condensation of the steam is taken at an inlet temperature (T_{cinlet}=25^{0}C) and the permissible temperature rise is $\Delta T_c = 5^0 C$. The reheating inlet pressure is taken 0.2 times of the HP turbine inlet pressure. The steam maximum temperature is fixed ($T_{throttle}$=800^{o}C).

III. MATHEMATICAL FORMULATION:

The models under consideration are based on an ultra super-critical coal based steam power plant with/without reheating. The following assumptions are considered during the design and formulation of the model:

1. The phase change occurs with flash transformation, i.e. under pressures above critical condition, the liquid to vapour transition occurs without any intermittent phase change.
2. Pressure drops in the boiler and condenser linings are considered to be negligible and ignored in the calculations.
3. Heat losses due to convection or radiation from the interlinking piping system is neglected in order to simplify the analysis.
4. The turbine and pumps are considered to run at 90% and 80% efficiencies respectively.
5. The reheating, if present, is done up to the steam maximum temperature.

The HP Turbine has the inlet of steam at $p_{throttle}$ and $T_{throttle}$ inlet conditions, with a mass flow rate of $\dot{m}$. The expansion is first considered isentropic and the enthalpies and entropies at 1 and 2s are calculated. The actual enthalpy at point 2 is given by:

$$h_2 = h_1 - \eta_{turb}\,(h_1 - h_{2s}) \qquad (1)$$

The same equations can be used for calculate the enthalpy at the end of LP Turbine (Point 4).

The reheat line is considered at 20% of $p_{throttle}$.Thus it can be calculated by:

$$p_2 = 0.2 * p_1 \qquad (2)$$

For the pump, the enthalpy is calculated using the isentropic compression from points 5-6. The actual enthalpy can be calculated as:

$$h_6 = h_5 + \frac{(h_{6s} - h_5)}{\eta_{pump}} \qquad (3)$$

The total work output can be calculated as the enthalpy difference during the expansion in the two stages of the turbine minus the work lost during pumping the water to the boiler. The following equations provide the work output in the cycle:

$$W_{turbine} = (h_1 - h_2) + (h_3 - h_4) \quad (4)$$

$$W_{pump} = (h_6 - h_5) \quad (5)$$

$$W_{total} = W_{turbine} - W_{pump} \quad (6)$$

The heat addition occurs in the boiler in two parts. First the heat is added to convert water to steam and then reheating is done to heat the expanded steam to turbine inlet temperature. The net heat addition can be calculated as:

$$Q_{addition} = (h_1 - h_6) + (h_3 - h_2) \quad (7)$$

The First Law efficiency (thermodynamic efficiency) can be calculated by:

$$\eta_{th} = \frac{W_{total}}{Q_{addition}} \quad (8)$$

The total power output can be estimated as a product of the mass flow rate with the total power output. The following equation provides the power output:

$$P_{net} = \dot{m} * W_{total} \quad (9)$$

The exergy calculations involve estimation of the total exergy destroyed in each component of the cycle. The boiler has the exergy destruction based on the energy gained from the fuel fusion minus the heat addition to the steam.

$$E_{xA} = 958953.69\ kJ$$
$$E_{xB} = 68474.05\ kJ$$

Where, E_{xA} and E_{xB} are the exergy of the inlet and outlet flue gases in the furnace respectively.

For the steam, exergy at boiler outlet and inlet can be calculated respectively by the Gibb's energy equation:

$$E_{s1} = \dot{m}(h_1 - T_0 s_1) \quad (10)$$
$$E_{w6} = \dot{m}(h_6 - T_0 s_6) \quad (11)$$

Where, T_0 is the ambient temperature considered 298K (25^0C)

The exergy destruction of boiler can be calculated as:

$$I_{boiler} = (E_{xA} - E_{xB}) - (E_{s1} - E_{w6}) (12)$$

The irreversibility rate in the steam turbine is given by:

$$I_{turbine} = T_0 \dot{m}((s_2 - s_1) + (s_4 - s_3)) \quad (13)$$

Mass of cooling water can be calculated by equating the heat lost by steam to the heat gained for a 5^0C rise in cooling water:

$$\dot{m_{cw}} c_p \Delta T_c = \dot{m}(h_4 - h_5) (14)$$

Irreversibility in condenser is given by:

$$I_{condenser} = T_0 (\dot{m_{cw}} c_p \ln\left(\frac{T_f}{T_d}\right) - \dot{m}(s_4 - s_5)) \quad (15)$$

The exergy utilised in boiler feed pump:

$$I_{pump} = T_0 \dot{m}(s_6 - s_5) (16)$$

Irreversibility or exergy loss through the exhaust gas:

$$I_{exhaust} = E_{xB} (17)$$

The total irreversibility is given by:

$$I_{total} = \sum I = I_{boiler} + I_{turbine} + I_{condenser} + I_{pump} + I_{exhaust} \quad (18)$$

The exergetic efficiency can be found as the exergy lost to the net exergy input:

$$\eta_{exergy} = \frac{(E_{xA} - I_{total})}{E_{xA}} (19)$$

The entropy generation is the total irreversibility per unit ambient temperature:

$$S_{gen} = \frac{I_{total}}{T_0} (20)$$

IV. NUMERICAL RESULTS AND DISCUSSION:

The above given modelling and formulation is based on an ultra super-critical steam power plant with reheating system. The model runs using coal as the fuel and the estimation is doe on second law analysis.The model is design for optimization under variable pressure and temperature conditions. The exergetic efficiency and entropy generation number are used to attain the optimum or apex at the plant will run at lowest exergy loss. Thermal (First law) efficiencies are also found under various pressure conditions. The mass flow rate of the plant is kept constant in the analysis. The modelling and estimation is done in EES platform [5].
Two cases and a comparative study are done to show the benefit of using a reheating facility in the ultra super-critical power plant. Comparisons are drawn based on constant mass flow rate of water (or steam) in the system.

CASE I – WITHOUT REHEAT:

This case deals with the model without reheating system. The steam is pressurized to super critical pressures and output plots are made for thermal efficiency, power output and second law efficiency. All the plots are made corresponding to maximum temperature of 800^{o}C.

The plot of thermal efficiency versus maximum steam pressure (Fig3.) depicts the steep increment of first law efficiency with steam pressure. The transition from sub to super critical pressure provided a more efficient power system. Thus, higher the pressure of boiler feed water is made the more efficient the power plant becomes.

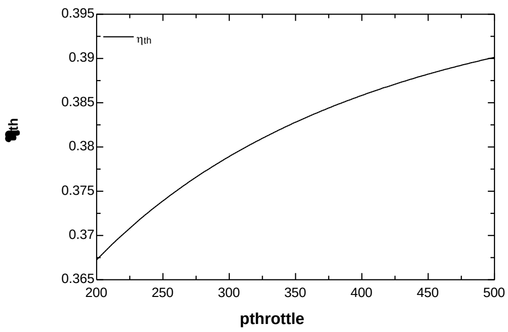


Fig3. Thermal Efficiency vs. Maximum Steam Pressure (bar)

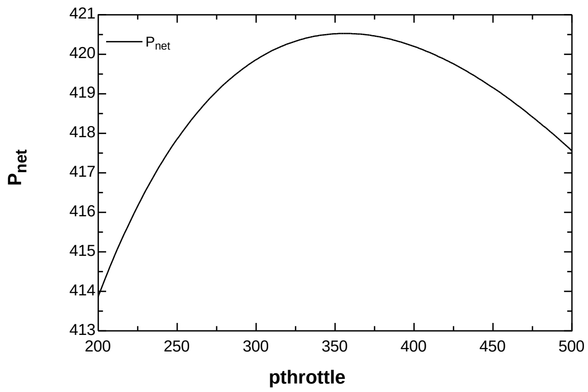


Fig4. Net Power Output (MW) vs. Maximum Steam Pressure (bar)

The Figure 4 represents the change of net output power with the maximum steam pressure. The net power output means the work output from the turbine minus the power consumed by the pump. It can be observed that till a particular steam pressure the curve rises to maxima (at 370 bars) and then falls down. This occurs because the pumping power increases at a faster rate than turbine power output, causing the net power to decrease after 370 bar of pressure.

The Figure 5 shows the change of Second Law (Exergetic) efficiency with respect to steam pressure. The second law analysis allows the estimation of exergy destruction. In the given figure, the exergetic efficiency is maximum at 370 bars of pressure. This depicts that the available energy (exergy) loss to the surrounding is minimum at this pressure, allowing the maximum utilization of energy in the power plant.

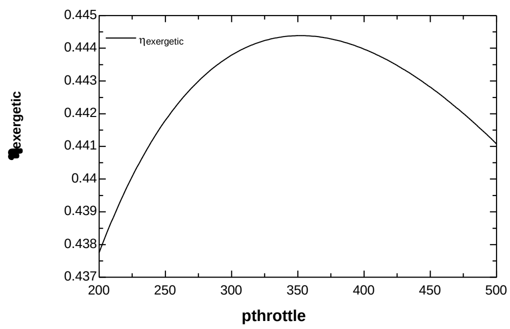


Fig5. Second Law Efficiency vs. Maximum Steam Pressure (bar)

CASE II – WITHOUT REHEAT:

This case deals with the model with reheating system. The steam is pressurized to super critical pressures and output plots are made for thermal efficiency, power output and second law efficiency. All the plots are made corresponding to maximum temperature of 800^{o}C.

The plot of thermal efficiency versus maximum steam pressure (Fig6.) depicts the increment of first law (thermal) efficiency with steam pressure. Thus, higher the pressure of boiler feed water is made the more efficient becomes the power plant.

The Figure 7 represents the change of net output power with the maximum steam pressure under a reheat system in application. It can be observed that till a particular steam

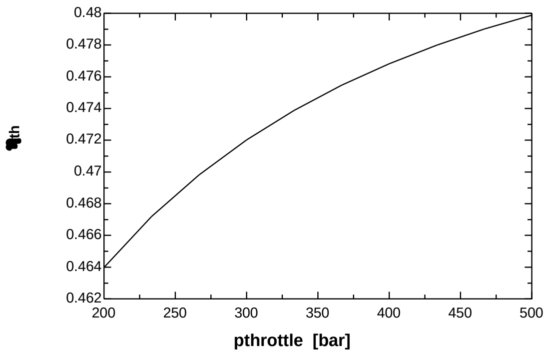


Fig6. Thermal Efficiency vs. Maximum Steam Pressure (Reheat)

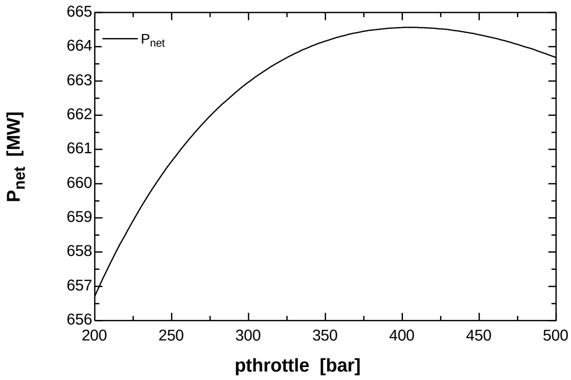


Fig7. Net Power Output vs. Maximum Steam Pressure (Reheat)

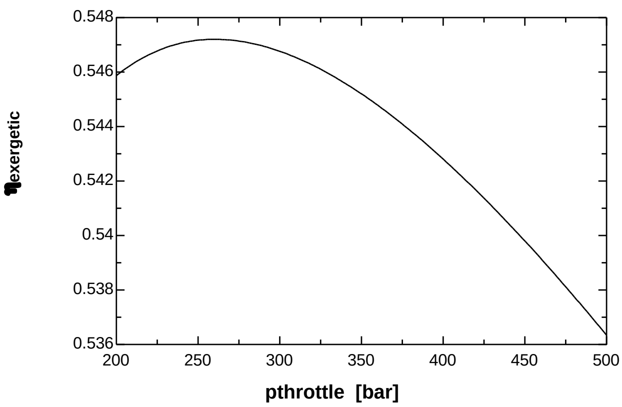


Fig8. Second Law Efficiency vs. Maximum Steam Pressure (Reheat)

pressure the curve rises to maxima (at 400 bars) and then falls down. Thus, to obtain the maximum power output from the plant, the steam should be pumped at 400 bars of pressure.

The Figure 8 shows the change of Second Law (Exergetic) efficiency with respect to steam pressure in the reheat based model. In the given figure, the exergetic efficiency is maximum at 260 bars of pressure. This depicts that the available energy (exergy) loss to the surrounding is minimum at this pressure, allowing the maximum utilization of energy in the power plant at the considered pressure.

CASE III – COMPARISON OF THE ABOVE TWO MODELS:

The ultra super-critical power plant gives different performance under situations with/without reheating of steam. It can be clearly deduced from Table I. , that a reheat plant has higher thermal efficiency and maximum net power output than the system without reheat.

TABLE I.

Table1. Thermal Efficiency and Net Power Output vs. Maximum Steam Pressure (with/without reheat)

$p_{throttle}$ [bar]	η_{th}(Without Reheat)	η_{th} (With Reheat)	P_{net} (Without Reheat) [MW]	P_{net} (WithReheat) [MW]
200	0.3672	0.464	413.9	656.7
233.3	0.3719	0.4672	416.8	659.6
266.7	0.3757	0.4698	418.7	661.6
300	0.3789	0.472	419.9	663
333.3	0.3816	0.4739	420.4	663.9
366.7	0.3839	0.4755	420.5	664.4
400	0.3858	0.4768	420.2	664.6
433.3	0.3875	0.478	419.6	664.5
466.7	0.3889	0.479	418.7	664.2
500	0.3901	0.4799	417.6	663.7

V. CONCLUSION:

The present day scenario demands the development of power systems that are highly efficient and produce ample amount of power. The paper deals with two ultra super critical power plant models whose thermal performance is compared. The models are designed with and without reheating, to represent their individual benefits and to draw a comparison between them.

The models have been developed considering the same mass flow rate of steam and constant maximum temperature. The following points were established as conclusion:

1. The thermal efficiency of an ultra super-critical power plant increases with increase in steam pressure.
2. The power output of the plant increases first with pressure increment and then decreases. Thus, the maximum of the curve represents the optimum pressure for maximum net power output. For the first model, the maximum power output is 420.5 MW at 370 bars and for the second model (with reheat), it is 664.6 MW at 400 bars.
3. The maximum of the exergetic efficiency curve represents the pressure at which the exergy (available energy) loss is minimum. For model without reheat, the pressure is around 370 bars and with reheat it is around 260 bars that we get the minimum loss plant operation.
4. The comparative analysis implements that an ultra super-critical power plant with reheating provides a 20% increment in thermal efficiency and 37% increment in net power output.

Thus, it can be concluded that ultra-super critical power plants provide very high power outputs at sufficiently high thermal efficiency. Furthermore, the exergetic analysis provides us an insight of the pressure needed for the plant to operate at minimum losses.

REFERNCES:

[1] Mukesh Gupta, Raj Kumar, Bhupender Singh; Exergy Based Evaluation of Coal Based Thermal Power Plants: A Review; (ISSN 2250-2459 (Online), An ISO 9001:2008 Certified Journal, Volume 3, Special Issue 2, January 2013.

[2] YongpingYanga, LigangWanga, Changqing Donga, Gang Xua, Tatiana Morosuk, George Tsatsaronis; Comprehensive exergy-based evaluation and parametric study of a coal ultra super-crititcal power plant; Applied Energy 112 (2013) 1087-1099.

[3] Bejan, A. (1982). Entropy Generation Through Heat and Fluid Flow. New York: Wiley.

[4] I.Satyanarayana, Dr.A.V.S.S.K.S. Gupta, Dr.K.GovindaRajulu; Second Law Analysis of Super Critical Cycle; International Journal of Engineering (IJE), Volume(4): Issue(1).

[5] Klein SA (2008) EES - Engineering Equation Solver, version 8.158, F Chart Software, Middleton, WI.

Nomenclature

c_p	Specific heat of fluid
E_{s1}	Exergy of steam at boiler outlet
E_{w6}	Exergy of water at boiler inlet
E_{xA}	Furnace exergy
E_{xB}	Exergy of exhaust flue gas
h1	Enthalpy at boiler outlet
h2	Enthalpy at HP turbine outlet
h2	Enthalpy at pump outlet
h2s	Isentropic enthalpy at HP turbine outlet
h3	Enthalpy after reheating
h4	Enthalpy at LP turbine outlet
h4s	Isentropic enthalpy at LP turbine outlet
h5	Enthalpy at condenser outlet
h6s	Isentropic enthalpy at pump outlet
I	Irreversibility
$\dot{m}_{cw}$	Mass flow rate of cooling water
$\dot{m}$	Mass flow rate of steam
p1	Pressure at boiler outlet
p2	Pressure after first stage of expansion
P_{net}	Net power output
$p_{throttle}$	Maximum pressure (boiler outlet pressure)
q	Heat flux
Q	Heat rate
S_{gen}	Entropy generated
T_o	Ambient temperature

Greek symbols

η	Efficiency