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Generalized Thermodynamic Analysis of Steam Power Cycles with 'n' Number of Feedwater Heaters

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Abstract

Thermodynamic analysis has been carried out to analyze the effect of 'n' feedwater heaters (fwhs) on performance of a steam power cycle with a generalized mathematical formulation. The performance calculations are formulated separately to single fwh and extended to 'n' fwhs for parametric study. The optimum bled steam temperature ratio is found at 0.4 with single fwh at given working conditions. Similarly the optimum pressure in a steam reheater is obtained at 20-25% of the boiler pressure irrespective of the number of heaters. The results show that the maximum gain in the efficiency of cycle is obtained with the first fwh and the increment diminishes with the addition of the number of heaters. This work examined the improvements in efficiency with increases in boiler pressure, turbine inlet temperature and furnace temperature.

Key words: Energy, exergy, feedwater heaters, generalisation, Rankine cycle.

1. Introduction

The performance of the Rankine cycle can be improved by increasing the mean temperature of heat addition, i.e. increasing the degree of superheat, steam pressure and by multistage expansion with reheating. The research so far was focused on improving the degree of superheat, increasing steam pressure and reheating. The efficiency of the steam power cycle can be improved to a large extent by the incorporation of feedwater heaters (fwhs).

A thermodynamic analysis of a steam power plant with fwhs involves a lot of mass, energy and exergy balance equations. In this work an attempt has been made to simplify this tedious task with the aid of generalization of the problem with 'n' fwhs. This work is useful for analyzing the steam power cycle with any number of heaters. Tsatsaronis and Winhold (1984) carried out detailed mass, energy, exergy and money balances for a reference steam power plant and investigated the effect of the most important process parameters on the exergetic efficiency. Cerri (1985) analyzed the influence of steam cycle regeneration with direct contact heaters on combined plant performance from the thermodynamic point of view. Not much information is reported in the literature to analyze the Rankine cycle with the formulation of 'n' fwhs. The performance of the Rankine cycle can be effectively improved by reducing the amount of heat added at a low temperature region in the economizer with extracted steam at the intermediate state of the steam turbine (Nag, 2001). The irreversibility of the economizer may be eliminated if the feedwater enters the steam generator at a saturated liquid state. There are many methods available in the literature to improve the efficiency of the steam power cycle. Different ways of enhancing the performance of the coal-fired thermal power plants were presented by Bhatt and Rajkumar (1999). Reini

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and Taccani (2002) introduced a simulation model of a real 320 MW steam power plant to diagnose the losses. Szargut (2005) determined the influence of the regenerative fwhs on the operational costs of a steam power plant by means of incremental energy efficiency.

Capata and Sciubba (2006) described the feasibility of different power cycles from both thermodynamic and operative points of view. Dincer and Muslim (2001) conducted a thermodynamic analysis for a Rankine cycle reheat steam power plant. Rosen (2001) made a thermodynamic comparison of coal-fired and nuclear electrical generating stations using energy and exergy analyses. Rosen (2002) also explained his views regarding energy efficiency, loss and exergy based measures. Srinivas *et al.* (2006) presented thermodynamic analyses, i.e. both energy and exergy analyses for a coal based combined cycle power plant consisting of a pressurized circulating fluidized bed (PCFB) partial gasification unit and an atmospheric circulating fluidized bed (ACFB) char combustion unit. Rosen and Raymond (2006) carried out energy and exergy analyses for a coal-fired steam power plant and evaluated possible modifications to improve the efficiency of the plant. Zaleta *et al.* (2007) evaluated steam turbines with the concept of exergoeconomic audits. They presented typical expressions for the exergy of solid fuel, heat transfer and shaft work. In the present work the chemical and physical exergetic components for the exergy analysis are determined with the method suggested by Kotas (1995).

The main aim of this work is to simplify the thermodynamic analysis of a regenerative steam power cycle with mathematical formulation for 'n' number of fwhs. This work is useful for thermodynamic modeling of a steam bottoming cycle in a combined cycle without repeating the mass and energy balance equations. In the present work, an attempt has been made to analyze the steam power cycle with fwhs from an exergy point of view.

2. Thermodynamic Analysis

2.1 Thermodynamic model

The schematic flow diagram of a steam power cycle is shown in *Figure 1*. The proposed model consists of 'n' fwhs and a steam reheater. The temperature-entropy diagram for the proposed model is shown in *Figure 2*. High pressure steam from the boiler enters the steam turbine to generate power. Some quantity of steam is extracted from the intermediate stage of the steam turbine to heat the feedwater which is entering the boiler through the fwh. A thermodynamic analysis is applied to the steam cycle having a single fwh to find the state of steam extraction. The working equations for 'n' fwhs are developed with a unit mass of steam at the turbine inlet. The temperature of the bled steam in a single fwh plays a key role on performance of the cycle. It is expressed in an excess temperature ratio as

$$\theta = \frac{T_{bled\ steam} - T_{co}}{T_{bo,\ sat} - T_{co}} \tag{1}$$

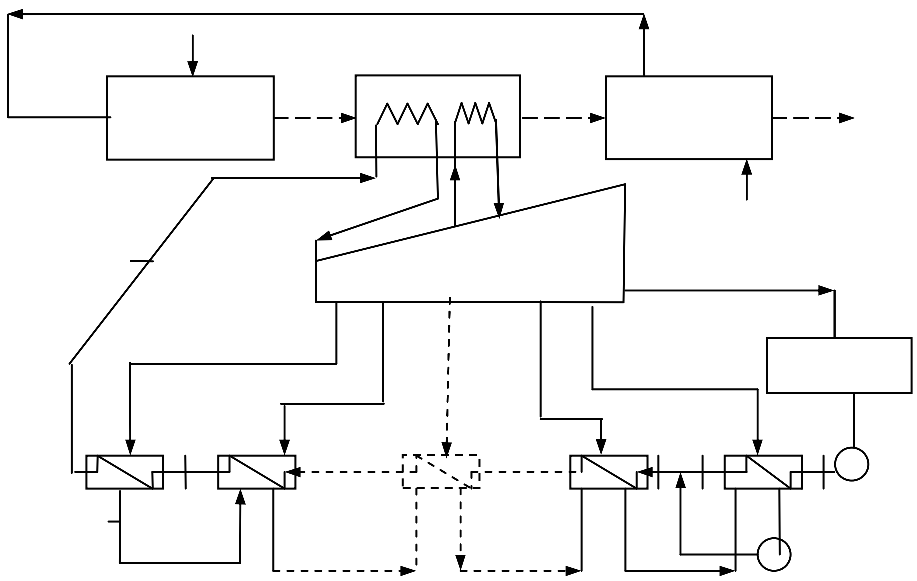

Figure 1. Schematic flow diagram of steam cycle with 'n' fwhs and a reheater

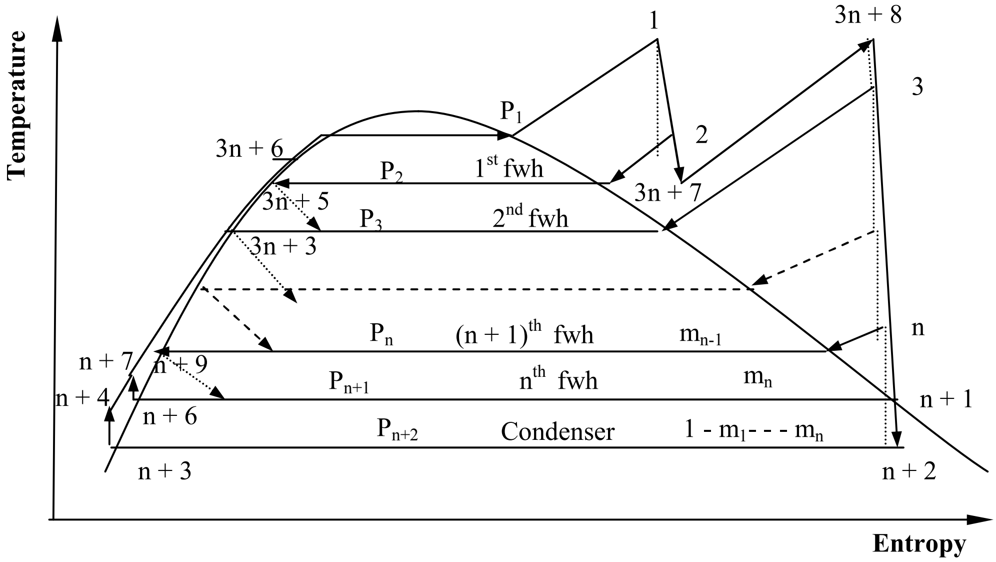

Figure 2. Temperature-entropy diagram of Rankine cycle with 'n' fwhs and a reheater

The following assumptions have been made for the analysis of the steam Rankine cycle with fwhs.

1. Ambient air pressure and temperature are taken as 1.01325 bar and 25°C respectively.
2. The flue gas inlet and outlet temperatures in the boiler are 1300°C and 300°C respectively.
3. Air inlet and outlet temperatures in the preheater are at 25°C and 250°C respectively.
4. The pressure and temperature at the steam turbine inlet are 160 bar and 550°C respectively.
5. The condenser pressure is assumed to be 0.085 bar
6. The terminal temperature difference (TTD) in fwh is 3.
7. The isentropic efficiency of the steam turbine equals 85%.
8. The heat loss in boiler, turbines, condenser and fwhs are neglected.

2.2 Analysis of coal

Anthracite coal fed to the furnace has been assumed to be of the following composition (Perry and Green, 1984).

Ultimate analysis: carbon 78.2%, hydrogen 2.4%, oxygen 1.5%, sulphur 1.0%, nitrogen 0.9%, moisture 8.0% and ash 8.0% (by mass).

The lower heating value (LHV) for coal is taken as 29,617 kJ/kg coal. The chemical availability or exergy of a fuel is the maximum theoretical work obtainable by allowing the fuel to react with oxygen from the environment to produce environmental components of carbon dioxide and water vapor. When the difference in availability between states of the same composition is evaluated, the chemical contribution cancels, leaving just the thermo mechanical contributions. This will be the case while doing exergy analyses for compressors and turbines. However, the chemical contribution will come into the picture during analyses of the coal furnace and fuel gas combustor. The standard chemical exergy for coal is taken at 31,430 kJ/kg coal (Kotas, 1995).

2.3 Mathematical generalization for 'n' fwhs

Let 'n' be the number of closed fwhs. The enthalpy of the condensate at outlet of condenser having 2, 3 and 4 fwhs is taken as h_5, h_6 *and* h_7 respectively. Therefore for *'n'* fwhs, the enthalpy of the condensate becomes h_{n+3}.

Similarly, the remaining state points in the cycle are generalized for '*n*' fwhs. For a single fwh the equations are exclusively developed.

Condenser pressure, $P_{n+2} = 0.085$ bar.

Overall temperature difference,

$$\Delta T_{OA} = T_{bo} - T_{co} \tag{2}$$

Saturation temperature at P_i,

$$T_{sat,\ i} = T_{sat,\ n+2} + \left(\frac{n+2-i}{n+1}\right)\Delta T_{OA} \tag{3}$$

(*i* increases from 2 to $n + 1$)

Enthalpy of water at outlet of feed pumps in kJ per kg steam,

$$h_{n+4} = h_{n+3} + v_{n+3}\,(P_1 - P_{n+2})100 \tag{4}$$

$$h_{n+7} = h_{n+6} + v_{n+6}\,(P_1 - P_{n+1})100 \tag{5}$$

Temperature of feedwater after heating,

$$T_i = T_{sat,j} - TTD \tag{6}$$

(i increases from $n + 8$ with the increment of 2 in n times and j decreases from $n + 1$ with the decrement of 1 in n times.)

To simplify the analysis between n^{th} and $n - 1$ fwh, it is assumed that enthalpy before mixing is equal to after mixing for feedwater, i.e., $h_{n+8} = h_{n+5}$.

2.4 Energy analysis

The requirement of steam (kg/kg steam) to heat the feedwater is obtained from the enthalpy balance in heaters.

For single fwh, mass of bled steam,

$$m = \frac{h_6 - h_5}{h_2 - h_7 + h_6 - h_5} \tag{7}$$

For multi fwhs, mass of bled steam for the 1st fwh,

$$m_1 = \frac{h_{3n+6} - h_{3n+4}}{h_2 - h_{3n+5}} \tag{8}$$

Mass of bled steam for 2nd to n-1 fwh,

$$m_i = \frac{(m_1 + m_2 + ---- m_{i-1})(h_p - h_q + h_r - h_s)}{h_{i+1} - h_p} \tag{9}$$

In the above equation (9), i increases from 2 to n - 1. The variables p, q, r, and s vary with the decrement of 2 from $3n + 3$, $3n + 5$, $3n + 4$ and $3n + 2$ respectively.

Mass of bled steam for n^{th} fwh,

$$m_n = \frac{(m_1 + m_2 + --+ m_{n-1})(h_{n+6} - h_{n+9}) + (1 - m_1 - m_2 --- m_{n-1})(h_{n+5} - h_{n+4})}{h_{n+1} - h_{n+4} - h_{n+6} + h_{n+5}} \tag{10}$$

Steam generation in boiler per kg coal,

$$m_s = \frac{h_{gi} - h_{go}}{h_1 - h_{3n+6} + m_{rh}(h_{3n+8} - h_{3n+7})} \tag{11}$$

Where m_{rh} is steam flow in reheater per kg steam.

$$h_{gi} = (T_{gi} - T_0)\, \Sigma_k n_k\, \bar{c}_{p,k}^{h} \tag{12}$$

and

$$h_{go} = (T_{go} - T_0)\, \Sigma_k n_k\, \bar{c}_{p,k}^{h} \tag{13}$$

Where $\bar{c}_{p,k}^{h}$ is the specific heat of k^{th} chemical component in kJ/kg mol.

Turbine work per unit mass of coal,

$$w_{st} = m_s \sum_{i=1}^{n+1} (1 - m_{i-1})(h_i - h_{i+1}) + (h_{3n+8} - h_{3n+7}) \tag{14}$$

Pump work per unit mass of coal,

$$w_P = m_s\, [(1 - m_1 --- m_n)(h_{n+4} - h_{n+3}) + (1 + m_1 + --- m_n)(h_{n+7} - h_{n+6})] \tag{15}$$

Net Work,

$$w_{net} = w_{st} - w_p \tag{16}$$

Energy efficiency of the cycle,

$$\eta_1 = \frac{w_{net}}{LHV} \times 100 \tag{17}$$

2.5 Exergy analysis

Exergetic losses associated in all the components are estimated per unit mass of fuel (Bejan, 1982).

The combustion equation in the furnace is

$$(6.516C + 1.2H_2 + 0.03214N_2 + 0.046875O_2 + 0.03125S + 0.44\, H_2O) + x\,(O_2 + 3.76N_2) = 6.516\, CO_2 + 1.644H_2O + 0.03125SO_2 + (x - 7.1)O_2 + (0.03214 + 3.76x)\, N_2 \tag{18}$$

The amount of air required (x) is determined by the energy balance in a coal furnace to get the required conditions.

Chemical exergy,

$$e_{ch} = \sum_k n_k \overline{\varepsilon^0{}_k} + \bar{R}T_0 \Sigma_k n_k \ln.[P.x_k] \tag{19}$$

Where x_k is the mol fraction of k^{th} component.

Physical exergy,

$$e_{ph} = h - \Sigma_k T_0 s_k \tag{20}$$

Total exergetic loss,

$$e = e_{ch} + e_{ph} \tag{21}$$

Exergetic loss in a furnace due to combustion of coal with air,

$$i_{fur} = e_{coal} + e_{ao} - e_{gi} \tag{22}$$

Exergetic loss in boiler due to heat transfer between flue gas and water/steam,

$$i_{bo} = e_{gi} - e_{go} - m_s\{h_1 - h_{3n+6} + m_{rh}(h_{3n+8} - h_{3n+7}) - T_0\,[s_1 - s_{3n+6} + m_{rh}(s_{3n+8} - s_{3n+7})]\} \tag{23}$$

Exergetic loss with the exhaust,

$$i_{ex} = e_{go} \tag{24}$$

Total exergetic loss with combustion and heat transfer in the boiler,

$$i_{bot} = i_{fur} + i_{bo} \quad (25)$$

Exergetic loss in a steam turbine,

$$i_{st} = m_s T_0 \sum_{i=1}^{n+1} (1 - m_{i-1})(s_{i+1} - s_i) + (s_{3n+7} - s_{3n+8}) \quad (26)$$

Mass of cooling water required in a condenser,

$$m_w = [m_s\,(1 - m_1 - m_n)\,(h_{n+2} - h_{n+3})] \,/\, (c_{pw}\,(T_{out} - T_{in})) \quad (27)$$

Exergetic loss in a condenser,

$$i_{co} = m_s\,(1 - m_1 - m_n)\,(s_{n+3} - s_{n+2}) + T_0\,[m_w\,c_{pw}\,(T_{out} / T_{in}) \quad (28)$$

Exergetic loss in hot water from a condenser,

$$i_w = m_w[\,c_{pw}\,(T_{out} - T_0) - T_0\,R.log(T_{out} / T_o)] \quad (29)$$

Total (internal + external) exergetic loss in condenser,

$$i_{cot} = i_{co} + i_w \quad (30)$$

For single fwh, exergetic loss in fwh,

$$i_{fwh} = m_s\,T_0\,[\,m(s_7 - s_2) + (\,1 - m)(s_6 - s_5)] \quad (31)$$

For multi fwhs,

Exergetic loss in 1st fwh,

$$i_{fwh1} = m_s\,T_0\,[m_1(s_{3n+5} - s_2) + (s_{3n+6} - s_{3n+4})] \quad (32)$$

Exergetic loss in fwhs from 2nd to *n-1*,

$$i_{fwhi} = m_s\,T_0\,[m_i\,(s_p - s_{i+1}) + (m_1 + m_2 + \text{- -}\ m_{n-1})\,(s_p - s_q) + s_r - s_s)] \quad (33)$$

The variations in i, p, q, r and s are the same as in equation (9).

Exergetic loss in n^{th} fwh,

$$i_{fwh,n} = m_s\,T_0\,[(m_1 + m_2 + \text{- -}\ m_{n-1})\,(s_{n+6} - s_{n+9}) + m_n(s_{n+6} - s_{n+1}) + (1 - m_1 - m_2 \text{- - -}\ m_n)\,(s_{n+5} - s_{n+4})] \quad (34)$$

Exergetic loss in all '*n*' fwhs,

$$\mathrm{i_{fwh}} = \sum_{i=1}^{n} i_{fwhi} \quad (35)$$

Exergy efficiency,

$$\eta_2 = \frac{w_{net}}{e_{coal}} \times 100 \quad (36)$$

3. Results and Discussion

A thermodynamic analysis has been carried out in up to ten heaters in order to investigate the effect of heaters on the performance of a steam cycle. The net work output of cycle, in the percent of lower heating value as well as exergy of coal, is expressed as energy and exergy efficiencies respectively to evaluate the cycle (Dincer, Muslim, 2001).

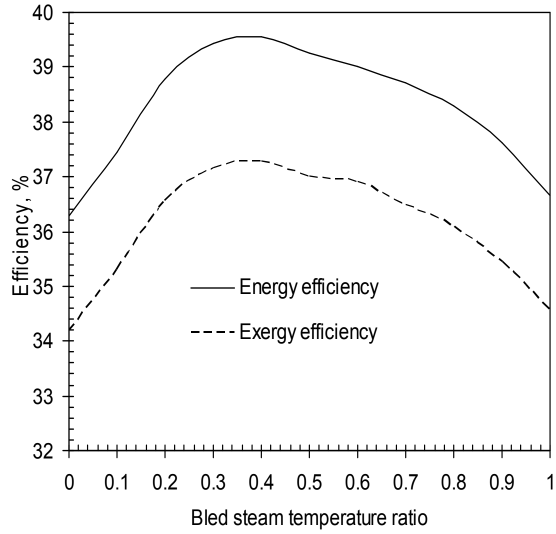

Figure 3. Effect of bled steam temperature ratio on efficiency of the cycle with single fwh

3.1 Effect of bled steam temperature ratio on cycle performance with single fwh

The effect of bled steam temperature on energy and exergy efficiencies of a steam cycle with single fwh has been developed in *Figure 3.* The efficiency of cycle maximizes at a particular bled steam temperature ratio for given conditions. The work output from the steam turbine as well as heat input to the boiler decreases with an increase in the steam temperature ratio because of the increased heating value in extracted steam. The regeneration effect improves the efficiency but the drop in the steam turbine output declines the same. Therefore the efficiency increases and afterwards decreases with the increase in the bled steam temperature ratio. The optimum temperature ratio is found at 0.4 for single fwh at given working conditions.

The exergetic loss in the boiler is depicted in *Figure 4* with the bled steam temperature ratio. The exergetic loss in a boiler occurs with combustion and heat transfer. The exergetic loss due to combustion of coal is independent of steam temperature ratio. The exergetic loss developed in a boiler due to heat transfer between flue gas and water/steam decreases with an increase in the steam temperature ratio. The major portion of exergetic loss of cycle is in the boiler compared to the other components of cycle. Therefore the plots are prepared for boiler exergetic loss only. The combustion exergetic loss is moreover the heat transfer exergetic loss. At 0.4 temperature ratio, the total exergetic loss in the boiler is around 49% of exergy value of the coal. The total exergetic loss decreases with an increase in the temperature ratio.

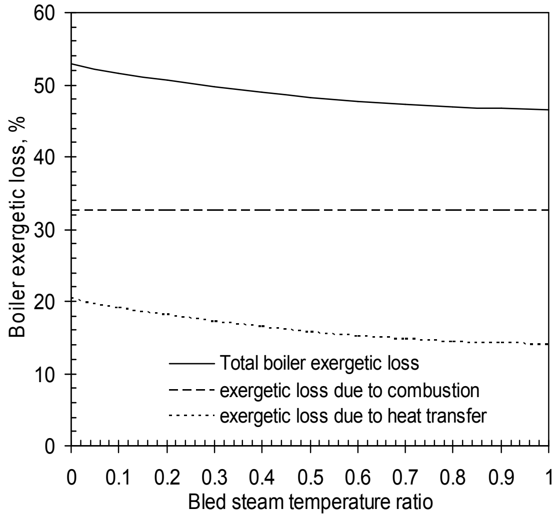

Figure 4. Effect of bled steam temperature ratio on exergetic loss in boiler with single fwh

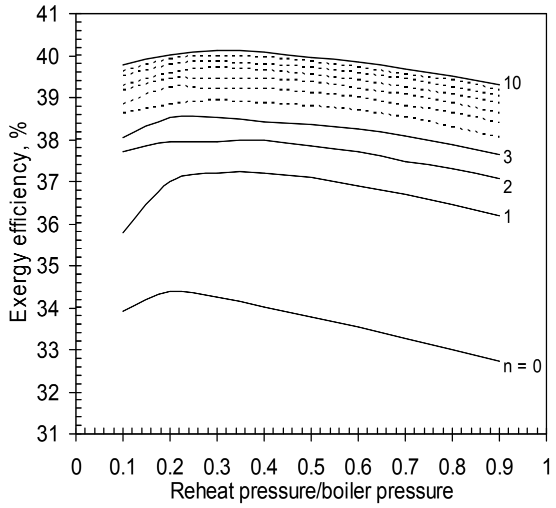

Figure 5. Effect of reheat pressure with heaters on exergy efficiency

3.2 Effect of the number of fwhs on cycle performance

The effect of reheat pressure, boiler pressure, turbine inlet temperature and flue gas temperature has been discussed parametrically on the performance of cycle by means of a different number of fwhs. The results obtained with the mathematical formulation are matched with the work of Tsatsaronis and Winhold (1984). Steam reheat pressure is expressed as a ratio of boiler pressure. *Figure 5* shows the effect of the reheat pressure ratio on the exergy efficiency of the steam power cycle. The steam reheat pressure ratio varies from 0.1 to 0.9 i.e., 10-90% of boiler pressure. The work output from steam turbine and heat input to boiler decreases with increase in the reheat pressure. Therefore the efficiency of the plant maximizes at the optimum reheat pressure. The exergy efficiency increases with an increase in the number of heaters at a diminishing rate. The reheat pressure is optimum at 20-25% of the boiler pressure.

Figure 6 generates the effect of the reheat pressure ratio on the total exergetic loss (combustion and heat transfer) in the boiler. The total boiler exergetic loss decreases first, thereafter increases with an increase in the reheat pressure as shown. The addition of heaters decreases the exergetic loss of the boiler due to the decreased temperature difference between the flue gas and the water/steam.

The effect of boiler pressure on the exergy efficiency of the steam power cycle with the number of fwhs is shown in *Figure 7*. The efficiency increases with an increase in boiler pressure and also with the addition of heaters. At high steam pressure, steam carries more enthalpy and produces more work resulting in high thermal efficiency as expected.

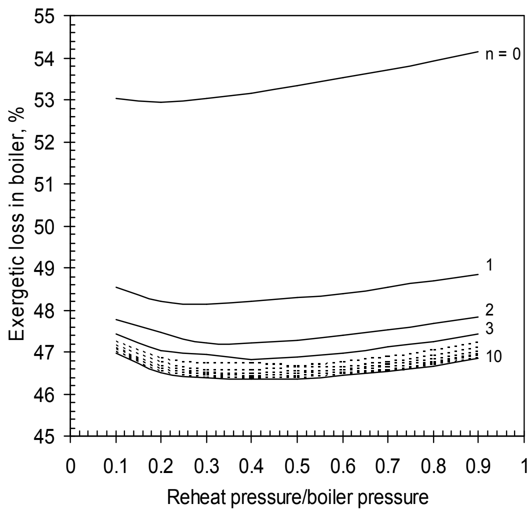

Figure 6. Effect of reheat pressure with heaters on exrgetic loss in boiler

Even though the exergy efficiency increases with the number of heaters, it may not be advisable to have more than the optimum number as for more fwhs there will not be much enhancement in the efficiency of the cycle. The exergy efficiency has been increased from 34 to 37% at the boiler pressure of 160 bar with the addition of first fwh.

The effect of number of fwhs on the exergy efficiency of the cycle for various turbine inlet temperatures is presented in *Figure 8.* The exergy efficiency is found to be increasing with an increase in the turbine inlet temperature and also with the addition of fwhs. The enthalpy of steam at the turbine inlet increases with an increase in the inlet temperature at a fixed pressure.

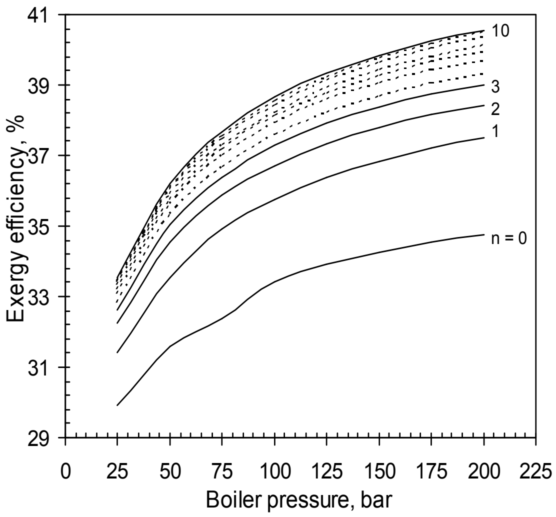

Figure 7. Effect of number of fwhs on exergy efficiency for different boiler pressures

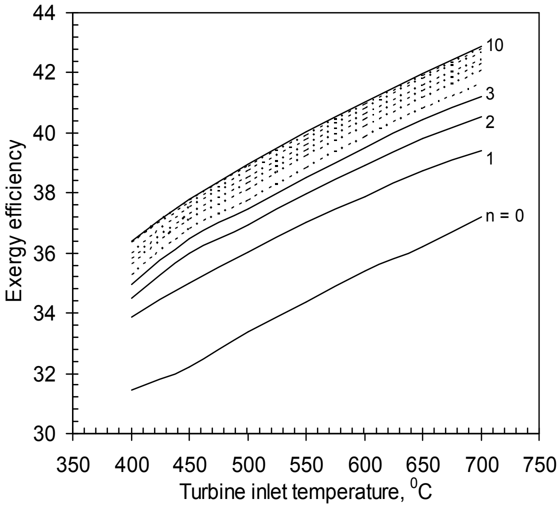

Figure 8. Variation exergy efficiency with number of fwhs for different turbine inlet temperatures

The work output, thereby the efficiency of the cycle, increases with an increase in the turbine inlet temperature. The influence of steam temperature on the efficiency of the cycle is more than the steam pressure. Therefore the slope in the temperature curve is more than in the pressure curve.

The effect of the flue gas temperature in the furnace on exergy efficiency is presented in *Figure 9.* There is a marginal increase in exergy efficiency with the increase in flue gas temperature. The steam generation hence the turbine output increases with an increase in furnace temperature.

The exergetic losses of the cycle components in the percent of standard chemical exergy of coal are compared with different numbers of heaters in *Figure 10.* The major exergetic loss observed in the boiler resulted in combustion and heat transfer. It shows the need for research to reduce the exergetic loss in a boiler. The exergetic loss in a furnace can be decreased with fluidized bed combustion. The exergetic loss in a boiler decreases with the introduction of fwhs as shown. The exergetic loss in a steam turbine and fwhs increases with an increase in the number of heaters. The condenser exergetic loss decreases with the incorporation of additional heaters. The exergetic loss in an air preheater and the exhaust remains the same with the number of heaters due to the fixation of air and flue gas outlet temperatures.

This work suggests the modification required in the coal furnace. The exergetic loss in boiler can be decreased with the coal gasification in which synthetic gas is produced from coal (Srinivas *et al*. 2006).

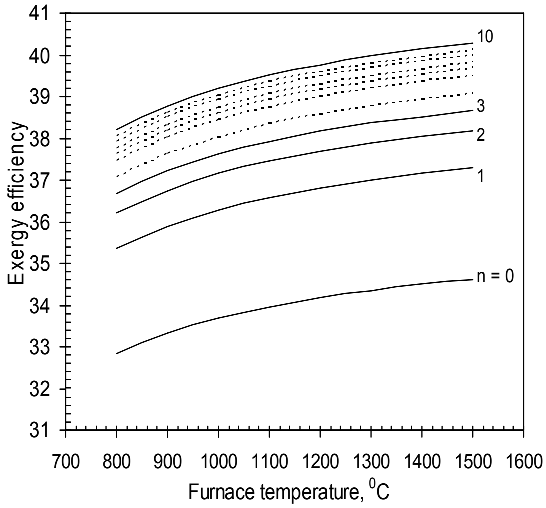

Figure 9. Effect of furnace temperature with fwhs on exergy efficiency of the cycle

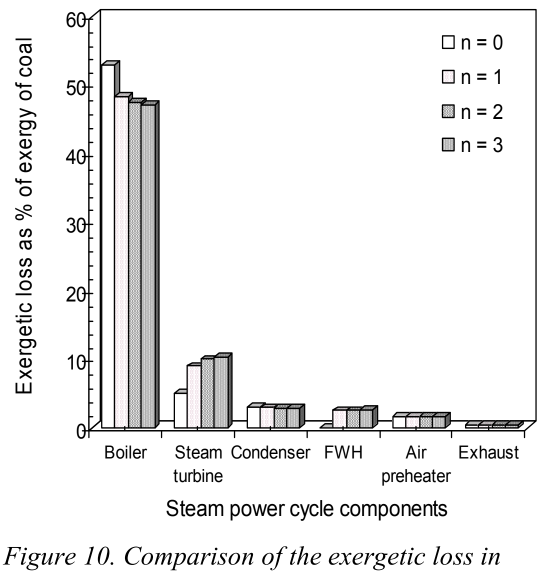

Figure 10. Comparison of the exergetic loss in cycle components in percent of exergy of coal with and without fwhs

4. Conclusion

In this work, a complex thermodynamic analysis of a steam power cycle with fwhs is simplified with the introduction of a generalized concept for ‘n’ fwhs. This work can be applied to thermodynamic modeling of nuclear power plants or the steam bottoming cycle in a combined power cycle. For a single fwh, the efficiency is maximized at a bled steam temperature ratio of 0.4. The efficiency of the cycle is high at the reheater pressure of 20-25% of the boiler pressure. The exergetic loss in the boiler decreases with the addition of heaters. The results show that the exergetic loss associated with the combustion of fuel is more when compared to heat transfer.

Nomenclature

e	specific exergy, kJ/kg
h	specific enthalpy, kJ/kg
i	specific irreversibility, kJ/kg
LHV	lower heating value, kJ/kg coal
n	number of heaters
P	pressure, bar
$\overline{R}$	universal gas constant, kJ/kg mol K
s	specific entropy, kJ/kg. K
T	temperature, K
TTD	terminal temperature difference, K
v	specific volume, m^3/kg
w	specific work, kJ/kg coal
ε	standard specific exergy, kJ/ kg
η	efficiency
θ	excess temperature difference

Subscripts

bo	boiler
ch	chemical
co	condenser
fur	furnace
fwh	feedwater heater
in	inlet
OA	over all
out	outlet
p	pump
ph	physical
rh	reheater
s	steam
sat	saturation
st	steam turbine
w	water
0	reference state
1	first law
2	second law

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