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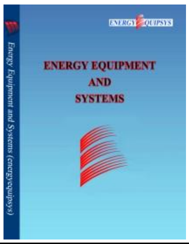

The optimum pressure for working fluid in feed water heaters of steam power plants

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ABSTRACT

The aim of this study is to find the optimal water pressure and percentage of supply vapour in the feed water heaters (fwhs) of steam power plants, such that they maximize the thermal efficiency of the Rankine cycle within pre-specified values of minimum and maximum pressures of the thermodynamic cycle. Thermal efficiency is defined as a function of unknown variables (fluid pressure and vapour percentage of each fwh), and it is maximized numerically using the nonlinear constraint optimization method. Precise values of enthalpy are used in computations of thermal efficiency during the nonlinear optimization process. The enthalpy and entropy values at different points of the thermodynamic cycle are calculated utilizing the industrial formulation of IAPWS-IF97.

Keywords: Steam Power Plant, Feed Water Heater, Optimum Pressure.

1. Introduction

In thermal power plants that are equipped with feed water heaters (fwhs), a percentage of the steam is taken out from the turbine and mixed with the incoming water from the pumps placed after condenser. Utilizing fwh increases the thermal efficiency apart from reducing the net output power of the cycle. The steam may be taken during the high-, middle-, or low-pressure stages of the turbine; each case has a different total thermal efficiency. It is clear that the pressure and percentage of the steam taken from the turbine have considerable effects on the overall thermal efficiency of the cycle. Therefore, their values should be chosen such that the highest degree of thermal efficiency is achieved. A few authors have proposed theories for choosing the best value of steam pressure (optimum pressure of fwh). Srinivas *et al.* [1] studied a steam power cycle with one reheater and closed fwhs. For the cycle with a single fwh, they defined the bled steam temperature ratio as:

$$\theta = \frac{T_{sat.,bled\ steam} - T_{sat.,Condenser}}{T_{sat.,Boiler} - T_{sat.,Condenser}} \qquad (1)$$

The optimum value of θ was 0.4. Bejan [2], in chapter 8 of his book, presented an optimization approach for the thermodynamic cycle having n numbers of fwhs, and concluded that the best pressure (or saturated temperature) of every fwh must be a value where the increase of the water enthalpy becomes equal in all "n" fwhs. He also provided a list of previous papers on this subject in the same chapter of his book. El-Wakil [3] stated that the temperature rise of the feed water in all fwhs should be equal, in order to reach the optimized cycle. In other

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words, if there are "n" fwhs in the cycle, the optimal temperature rise of every fwh is computed by:

$$\Delta T_{optimum} = \frac{T_{sat.,Boiler} - T_{sat.,Condenser}}{n+1} \quad (2)$$

Burghardt and Harbach [4] stated a similar expression in their book. Reference [5] recommended the use of this approach as well. Equation (2) is the traditional theory of equal rising of temperature in each fwh. Cao et al. [6] considered biomass-fired Kalina cycle with ammonia-water solution as the working fluid. They modelled this cycle with a regenerative reheater (fwh) as well as without any fwh.. The optimization variables were pressure and temperature at the turbine inlet, the vapour extraction percentage and its pressure from the turbine (in the case of regenerator fwh), and the separator temperature. They also set several limitations on their optimization variables; for example, they set the range of 8–12MPa on the turbine inlet pressure. They found the optimized system efficiency to be 28.43% with one fwh and 27.72% without it. Sengupta at al. [7] conducted a complete exergy analysis of a coal-fired steam power plant using actual operating data. The power plant had three low-pressure feed water heaters and two high-pressure fwhs. In the paper by Sengupta et al., the pressures of fwhs were all constant. However, they found that withdrawal of two high-pressure fwhs from the cycle reduces the exergy efficiency of the total cycle by 1.5%. Gupta and Kaushik [8] studied a direct steam generation solar thermal power plant. With water as its working fluid and a maximum pressure of 84.5bar in a 5MW conceptual design, they found the optimum values for the bleed pressures and mass fractions of the bleed steam in the system with up to three fwhs. Moghadassi et al. [9] determined the operating conditions in a regenerative cycle with one open feed water heater to maximize the output the cycle. They used genetic algorithm as the optimization tool and artificial neural networks as a tool for finding thermodynamic properties of water at different points the cycle. Their case study was a subcritical steam cycle with 5Mpa pressure for the boiler. Farhad et al. [10] performed exergy analysis in four operating power plants using pinch technology to reduce the irreversibility of fwh networks. The pressures of all boilers were subcritical in their work. Akolekar et al. [11] found the optimum pressures of reheater and fwh in a cycle which had both reheater and fwh. They maintained the boiler pressure up to 22MPa. Similarly, in the Organic Rankine cycles (ORC), where organic fluids instead of water are used as the working fluid, the regenerative heat exchangers can be utilized within the cycle. There are a number of papers dealing with the usage of fwh in ORC; two relevant ones are mentioned here. Ventura and Rowlands [12] numerically modelled ORC for a wide range of working fluids. They considered cycles both with and without the recuperator heat exchanger. The recuperator inlet hot flow was directly connected to the outlet of the turbine (no partial extraction from turbine). For hot source temperature up to 210°C, they proposed an appropriate type of organic fluid to be used in the cycle. Le et al. [13] studied supercritical ORC both in basic and in single regenerative (single fwh) configurations. They considered eight organic fluids in their study. With a maximum heat source temperature of 150°C, their optimum system efficiency was 13% for the organic fluid R152a.

There are some shortcomings in determining the pressure of fwhs as far as the previously mentioned studies are concerned: (1) In the methods of equal rising of temperature or enthalpy in each fwh, some simplifying assumptions and approximations have been made, e.g. they are fit for open fwhs not closed ones. (2) Some methods did not use the precise numeric values of enthalpy of water for calculating the efficiency. (3) Most of them considered thermodynamic cycles where the maximum pressure of the cycle (boiler pressure) is less than the critical pressure of water. In steam power plants that work with supercritical pressures, up to $30MPa$ for example, it is not possible to determine the optimal pressure of fwh based on the previous theories, as the saturated temperature is not defined for supercritical pressures. (4) The previous traditional theories are relevant to the cycles that do not have reheaters. Therefore, for reheat cycles, the traditional methods of equal rising of temperature or enthalpy in all fwhs may not be accurate.

The approach proposed in this work aims to cover these shortcomings. It defines thermal efficiency directly as a function of unknown variables (pressure and percentage of steam in every fwh). If arbitrary values are set for these variables, the thermal efficiency can be calculated based on the enthalpy values at different points of the thermodynamic cycle.

The values for the unknown variables are determined such that they maximize the function of thermal efficiency. This is fulfilled by the nonlinear constraint optimization method. The results are used to evaluate the previous approach (Eq. (2)) for determining pressure of fwh.

In this paper, the enthalpies at different points of the cycle are calculated based on the industrial formulation of International Association for the Properties of Water and Steam (IAPWS-IF97), as well as a recent paper by the current author [14]. The IAPWS formulation is available in several manuscripts within its website [15]. With the use of IAPWS-IF97, accurate numeric values of enthalpy and entropy can be calculated for pressures up to $100MPa$. Therefore, a power plant whose maximum water pressure is higher than the critical pressure ($22.064\ MPa$) can also be modelled by the proposed method.

Nomenclature

h	Enthalpy per unit mass of the working fluid, [kJ/kg]
n	The number of feed water heaters
m'	Fraction of the vapour mass flow rate taken from turbine to the total mass flow rate of working fluid
P	Pressure, [MPa]
q	Heat transfer per unit mass of the working fluid, [kJ/kg]
T	Temperature, [°C]
w	Work per unit mass of the working fluid, [kJ/kg]

Subscript

f	Saturated fluid
i	Index of feed water heaters, i=1,2,…,n

Greek

η	Thermal efficiency of the cycle

2. The description of the problem

The considered thermodynamic cycle shown in Fig. 1 consists of "n" fwhs. For comparison to the previous methods, the fwhs are of the open type. All processes are considered reversible. Also, turbines and pumps are adiabatic. Thermal efficiency of the cycle is defined as:

$$\eta = \frac{w_{net}}{q_H} = \frac{w_{turbine} - \sum_{i=i}^{n+1} w_{pump,i}}{q_H} \quad (3)$$

Each of the parameters, $w_{turbine}$, q_H, and $w_{pump,i}$, are stated in terms of the enthalpies of the unit mass (h) at the proper points of the cycle, using the first law of thermodynamics. For example:

$$w_{turbine} = \left(h_d - h_{c_n}\right) + (1 - m_n')\left(h_{c_n} - h_{c_{n-1}}\right) + \cdots + \left(1 - \sum_{i=1}^{n} m_i'\right)\left(h_{c_1} - h_e\right) \quad (4)$$

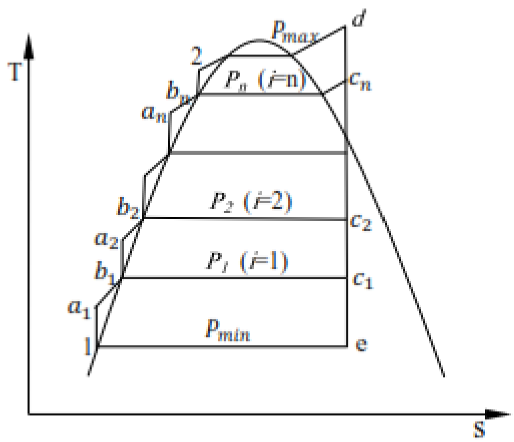

Fig.1. The considered ideal thermodynamic cycle of the steam power plant with n fwhs

where m'_i is the percentage of mass flow rate of the vapour extracted from turbine for the fwh number ith. to the total mass flow rate of the cycle ($m'_i = \dot{m}_i/\dot{m}_d$). The parameters of the cycle are: the minimum pressure of the cycle P_{min}(water pressure at condenser), the maximum pressure of the cycle P_{max}(water pressure at boiler), the maximum temperature of the cycle T_{max}(temperature at the boiler outlet), the pressures of fluid in each fwh (P_i), and all $m'_i s$. If these parameters are known, the values of enthalpies at important points of the cycle can be calculated using IAPWS-IF97. When the enthalpies are known, the thermal efficiency of the cycle (Eq. (3)) is calculated. For the optimization problem, only P_{min}, P_{max}, and T_{max} are known initially, while P and m' of each fwh are considered variable. Thus, η will be a nonlinear function of the following "$2\times n$" variables:

$$\eta = \eta(P_1, P_2, \dots, P_n, m'_1, m'_2, \dots, m'_n) \qquad (5)$$

The values of pressure and m' of fwhs must be determined in such a way that η reaches the maximum value under the specified constraints. This problem is solved by the nonlinear constrained optimization methods. The constraints are as follows:

$$P_{min} \le P_1 \qquad (6\text{-a})$$

$$P_i < P_{i+1} \quad \text{for } i = 1,2,\dots,(n-1) \qquad (6\text{-b})$$

$$P_n \le P_{max} \qquad (6\text{-c})$$

$$0 < m'_i \le \{m'_{i,max} = ([h_f]_{P_i} - h_{c_i})/(h_{a_i} - h_{c_i})\} \; ; \text{if } P_i \le P_{\text{critical point}} \qquad (6\text{-d})$$

Constraint (6-d) guarantees that the point b_i (the outlet of fwh number i^{th}) is not in the saturated region as this is important for proper function of the pumps.

In this study, the optimization is carried out using the Active Set Algorithm of the unconstrained nonlinear optimization method. A simple flowchart of the overall optimization process is given in Fig. 2.

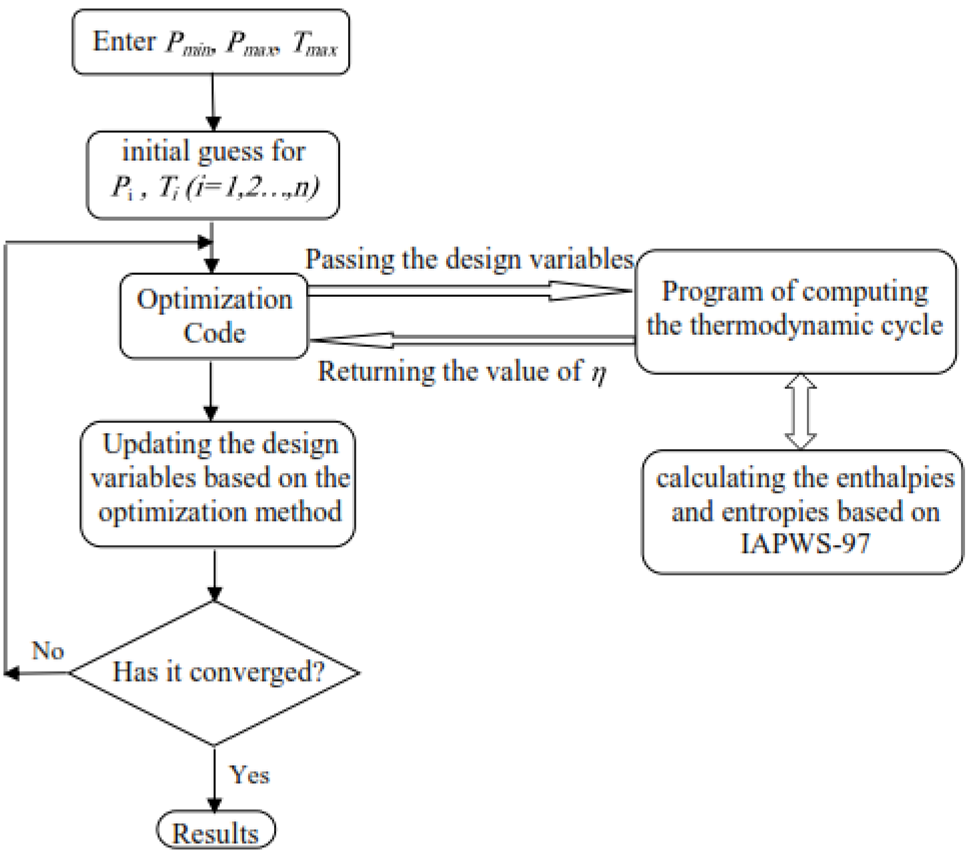

Fig.2. Simple flowchart for efficiency optimization of the cycle

3. Results

The pressure at the condenser is set to 10 kPa, while the maximum temperature in the cycle is set to 650℃ (at the outlet of boiler). The cycle is optimized for different pressures in the boiler. At first, a cycle with one fwh is modelled, and the results are presented in Table 1. In this table, as in all following tables, method ① refers to the optimization approach proposed in this paper, and method ② applies to Eq. (2) for setting the pressure in fwhs. In method ②, m_i' cannot be considered as an optimization variable, so it is set to $m_{i,max}'$ (defined by Eq. (6d)).Based on Table 1, it can be concluded that the previous theory (method ②), in spite of its simplicity, finds the optimum pressure close to the optimized value obtained from the numerical modelling of the total cycle. Two methods arrive at an identical value of thermal efficiency when the boiler pressure is less than the pressure of the critical point. The direct numerical modelling of the cycle (method ①) finds the best value of m_i' equal to $m_{i,max}'$. This means that the percentage of the vapour taken from turbine should be as high as possible, such that the outlet of fwh is the saturated liquid. In this work, no constraint was applied regarding the quality of fluid in turbine. Hence, the quality of the points near the outlet of the turbine obtain values lower than 0.9 in some cases. To avoid this, the reheater must be added to the cycle. This is possible in method ①; but not in method ②. So, the results of these two methods are not comparable in the reheat cycle. Since this study intends to compare the results of two methods in the same cycles, applying constraints on the quality of the turbine outlet is left to the future studies.

In order to have a better understanding of the effect of fwh pressure on the cycle performance, the pressure of fwh was changed from P_{min} to P_{max}, and η and w_{net} were calculated for each pressure. The results are illustrated in Fig. 3. In Fig. 3-a, it is seen that applying a single fwh with the optimum pressure can raise the total thermal efficiency of the cycle by 3% in its best state, as compared to the cycle without fwh. However, fwh always reduces the net work regardless of its pressure (Fig. 3b).

The results of the parameter optimization are presented in Table 2 for the cycle with two fwhs, and in Table 3, for the cycle with three fwhs. The biggest difference between the two methods in optimizing thermal efficiency is 0.005 for a cycle with $P_{max} = 20\ MPa$ and three fwhs. While the outlet of fwh (point b_i in Fig. 1) can be anywhere between a_i and the saturated liquid state, the proposed method finds m_i' such that the outlet of fwh automatically lies on the saturated liquid point.

Table 1. The optimum values of the parameters for a cycle with one fwh
①: the proposed optimization approach
②: applying Eq. (2)

$\mathbf{P_{max}}$ [MPa]	$(\mathbf{P_1}[\mathbf{MPa}], \mathbf{T_1}[°C])_{opt.}$ ①	$(\mathbf{P_1}[\mathbf{MPa}], \mathbf{T_1}[°C])_{opt.}$ ②	$\mathbf{m_1'}$ ①	$\lvert \mathbf{m_1'} - \mathbf{m_{1,max}'} \rvert$ ①	$\mathbf{w_{net}}$[kJ/kg] ①	$\mathbf{w_{net}}$[kJ/kg] ②	$\boldsymbol{\eta}$ ①	$\boldsymbol{\eta}$ ②
5	0.48087, 150.37	0.54167, 154.88	0.15646	7.9e-10	1330.83	1322.48	0.423226	0.423188
10	0.87921, 174.37	0.96638, 178.41	0.19624	3.83e-8	1361.19	1353.07	0.453764	0.453738
15	1.28996, 191.26	1.36840, 193.98	0.22500	2.2e-06	1357.50	1351.70	0.470765	0.470745
20	1.68718, 203.95	1.75161, 205.78	0.24787	1.1e-05	1342.34	1338.36	0.482130	0.482128
30	2.7780, 229.60	-----	0.29314	2.0e-08	1280.40	-----	0.49656	-----

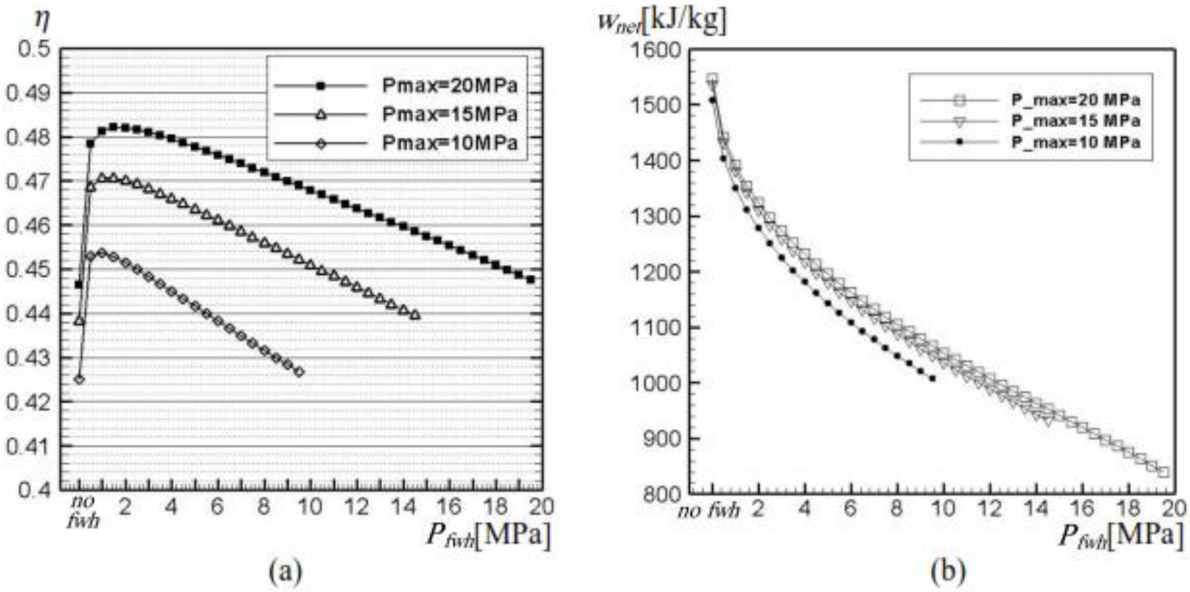

Fig.3. The effect of the fluid pressure in fwh on the efficiency of the cycle (a) and on the net generated work per unit mass of fluid (b), when one fwh is utilized in cycle

Table 2. The optimum values of the parameters for a cycle with two fwhs

P_{max} [MPa]	$(P_1[MPa], T_1[°C])_{opt.}$		m'_1	$\lvert m'_1 - m'_{1,max}\rvert$	$(P_2[MPa], T_2[°C])_{opt.}$	
	①	②	①	①	①	②
10	0.15044, 111.44	0.306, 134.21	0.11330	3.7e-8	1.56252, 201.81	2.437, 222.6
15	0.15643, 112.61	0.411, 144.59	0.11913	9.8e-9	2.1951, 217.14	3.55, 243.38
20	0.15764, 112.85	0.5083, 152.46	0.1228	9.4e-9	2.74548, 228.99	4.6239, 259.10
30	0.11904, 104.55	-----	0.11442	7.7e-14	3.60906, 244.33	-----
33	0.12310, 105.52	-----	0.11737	5e-6	3.77258, 246.91	-----

Table 2. (Continued)

P_{max} [MPa]	m'_2	$\lvert m'_2 - m'_{2,max}\rvert$	w_{net}[kJ/kg]		q_h[kJ/kg]		η	
	①	①	①	②	①	②	①	②
10	0.14716	9.4e-8	1329.88	1283.52	2878.10	2783.65	0.46207	0.46110
15	0.17497	2.67e-8	1327.59	1264.17	2766.88	2644.66	0.47982	0.47800
20	0.1986	1.8e-8	1312.57	1235.06	2669.44	2525.79	0.49170	0.48898
30	0.24452	-4.2e-6	1271.48	-----	2509.18	-----	0.50673	-----
33	0.25075	1.2e-6	1259.40	-----	2469.91	-----	0.50990	-----

4. Conclusion

The optimum values for pressure and the percentage of vapor extraction from the turbine into fwh was sought by direct numerical modeling of the thermodynamic cycle for steam power plant. The results were compared with the previous theory. When the boiler pressure is subcritical, the current method and the preceding one find the saturated pressure and temperature of each fwh somewhat different; nonetheless the optimized efficiency values are nearly identical in two methods. The proposed method confirms accuracy of simple previous theory in regulating the fwh parameters to reach the highest thermal efficiency. However, when either reheat is used in cycle or boiler pressure is higher than $22.064\ MPa$, the previous theories do not work while the proposed method is applicable.

Table 3. The optimum values of the parameters for a cycle with three fwhs.

	method	$P_{max}=10[MPa]$	$P_{max}=15[MPa]$	$P_{max}=20[MPa]$	$P_{max}=30[MPa]$
$(P_1[MPa], T_1[°C])_{opt.}$	1	0.0997, 99.52	0.0562, 84.26	0.056, 84.2	0.0679, 89.13
	2	0.1538, 112.11	0.1980, 119.9	0.2380, 125.79	-----
$(P_2[MPa], T_2[°C])_{opt.}$	1	0.6956, 164.7	0.3623, 140.08	0.2824, 131.5	0.4154, 145
	2	0.9664, 178.40	1.3684, 193.98	1.7516, 205.78	-----
$(P_3[MPa], T_3[°C])_{opt.}$	1	2.3657, 221.04	2.8353, 230.74	2.9311, 232.57	4.2239, 253.6
	2	3.6322, 244.70	5.3387, 268.07	6.9930, 285.76	-----
m'_1	1	0.0953	0.0733	0.0751	0.0875
m'_2	1	0.1113	0.1012	0.0898	0.1093
m'_3	1	0.0986	0.1570	0.1785	0.2048
$w_{net}[kJ/kg]$	1	1299.51	1308.67	1315.30	1259.64
	2	1243.17	1211.92	1171.37	-----
$q_h[kJ/kg]$	1	2790.83	2704.06	2652.71	2482.59
	2	2680.43	2524.65	2391.08	-----
η	1	0.46563	0.48396	0.49583	0.51116
	2	0.46380	0.48004	0.48989	-----

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