THE IMPACT OF COOLING WATER CONDITIONS ON EXERGY DESTRUCTION AND PERFORMANCE OF A STEAM POWER PLANT

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الملخص

الظروف التشغيلية للمكثف في محطات القوى البخارية لها تأثير مباشر على تغير ضغط المكثف وبالتالي على أداء المحطة. هذه الورقة خصصت لدراسة تأثير حالات مياه التبريد من تغير في درجة حرارتها وكمية سريانها على التحطيم في الاكسيرجي والأداء لمحطة قوى بخارية محطة قوى بخارية بقدرة 200 ميجاوات اختيرت لهذه الدراسة النموذج الرياضي يأخذ في الاعتبار ظروف التشغيل المتغيرة للمحطة وأيضا العمليات الثيرموديناميكية لدورة البخار الكاملة. تحليل الاكسيرجي يبين بالتفصيل التحطيم في الاكسيرجي لكل المحطة وعناصرها. تم بناء برنامج حاسوب متكامل اعتمادا على النماذج الرياضية المنجزة. اخذ في الاعتبار التغيرات الحقيقية لدرجة حرارة مياه التبريد ومعدل تدفقها وتأثيرها على التحطيم في الاكسيرجي وأداء المحطة. حيث أعتبر التغير في درجة حرارة مياه التبريد من 10 °C إلى 30 °C والتغير في كمية معدل التدفق لمياه التبريد من 60% إلى 100% من القيمة التصميمية. تم حساب التحطيم في الاكسيرجي ، القدرة المنتجة، كفاءة الاكسيرجي ومعدل استهلاك الطاقة الحرارية عند درجات حرارة دخول مياه التبريد ومعدل تدفق لقيم مختلفة وكذلك عند ثبوت الحمل الحراري للغلاية بحمل كامل.تم استنتاج أن قدرة المحطة البخارية تتغير بمقدار يصل حتى 17 ميجاوات نتيجة التغيرات التي سبق ذكرها.

ABSTRACT

Condenser operating conditions have been found to have a significant impact on condenser pressure and subsequently plant performance. This paper is devoted to study the impact of cooling water conditions, such as inlet temperature and flow rate on the exergy destruction and performance of a steam power plant. A 200 MW steam power plant is selected. The mathematical models considered take into account the off-design operating conditions of the power plant and also the thermodynamic processes that compose whole steam cycle. The exergy analysis provides a detailed breakdown of the exergy destruction for the overall plant and its components. A fully interactive computer program based on the implemented models is developed. The effects of typical variations of cooling water temperature and flow rate on plant exergy destruction and performance are considered. These include water temperature (range from 10 to 30 °C), and flow rate (range from 60 percent to 100 present of the design value). The exergy destruction, power output, exergy efficiency and heat rate were calculated at different values of condenser inlet cooling water temperatures and flow rates and at full load

boiler capacity. It is found in the present work that the capacity of the plant may differ by 17 MW due to the changes in the parameters under study.

KEYWORDS: Condenser pressure; Exergy destruction; Cooling system; Steam power plant.

INTRODUCTION

For the power plant working at varying operating conditions, the operating conditions of the condenser vary significantly [1-3]. The most essential parameters which determine the alteration of condenser conditions are the mass of steam flow to condenser, cooling water inlet temperature and flow rate.

The vacuum performance of a steam turbine condenser directly affects the performance of its associated plant cycle, as it controls the temperature at which heat is rejected to the surrounding. The efficiency of the condensing power plant, inter alias depends on the conditions of heat transfer in surface heat exchangers. Being involved in the thermodynamic processes composing the steam cycle, the exchangers play a significant role. The overall heat transfer coefficient U reflects the conditions mentioned above.

Actual values of the overall heat transfer coefficient (U) in steam condenser and in other heat exchangers of the plant often differ during the operation from the design value. These differences may be due to the variations of cooling water inlet temperature or flow rate. When, the value of the coefficient U decreases then the terminal temperature difference and the condenser pressure increases [4]. The exergy destruction due to irreversibility of the heat transfer through a finite temperature difference also increases. The power generation and the overall efficiency of the plant decrease as well.

One of the important factors contributing to the lower overall efficiency is the high condenser pressure. A detailed energy-exergy analysis has been carried out on a different power plants installed capacities [3, 5-8]. The literature on exergy analysis reveals a lack of clear information in particular thermal power plant under certain operating conditions. The present study is one such effort to show the effect of condenser pressure on the exergy destruction and performance of a unit of 200 MW capacities. It is well known that the condenser pressure is affected by varying cooling water inlet conditions, such as the temperature and flow rate. Some reasons are [9,10]:

- Insufficient numbers of circulating cooling water pumps comparing to the load demand.
- Water intake blockage due to seaweeds, sands, etc,
- Failure of band screens or any filtration devices among the intake structure and the condenser.
- Failure of coloration plant and no provision program of condenser tubing cleaning system.
- Using the cooling towers, thus at hot weathers the cooling water temperature will be equal the surrounding air temperature.

To put on to account this vacuum effects for a power plant operating at varying operating conditions, a mathematical model has been developed upon the data concerning the construction and operating conditions of the condenser **SF-11420** for a 200 MW unit [11]. A computer program is especially designed to calculate overall efficiency and exergy destruction. These calculations were carried out for various cooling water inlet temperatures and flow rates. Full load boiler capacity as well as off-design operating conditions of the power plant components is considered during these calculations [2, 11].

PROCESS DESCRIPTION OF 200 MW THERMAL POWER PLANT

A detailed flow-sheet of the considered 200 MW thermal power plant is shown in Figure (1). The plant consists of three turbines namely the high-pressure turbine (HPT), intermediate-pressure turbine (IPT) and low-pressure turbine (LPT). All three are mounted on a single shaft and the generator is coupled with them. In the plant regeneration feedwater system, there are four low pressure heaters (LPH), steam cooler, deaerator and three high pressure heaters (HPH). The turbine is throttle-governed with control valves before the HP stages. In the steady state operation the control valves are throttled at different loads to maintain constant pressure upstream to the HP control valve. The LPH-1 and LPH-2 heaters consist of only one zone, with temperature differences, 7.0 and 7.6 ^{0}C, respectively. LPH-3, LPH-4, HPH-1, HPH-2 and HPH-3 heaters consist of three zones namely, drain cooling zone, condensing zone and desuperheating zone, with terminal temperature differences, 6.3, 4.0, 4.8 and 4.3 ^{0}C, respectively. The main thermodynamic parameters (pressure and temperature) of the steam in the cycle are indicated in the flow sheet. The inlet turbine live steam parameters are, pressure 12.7MPa and temperature 535 ^{0}C. The design condenser pressure is 4.3 kPa.

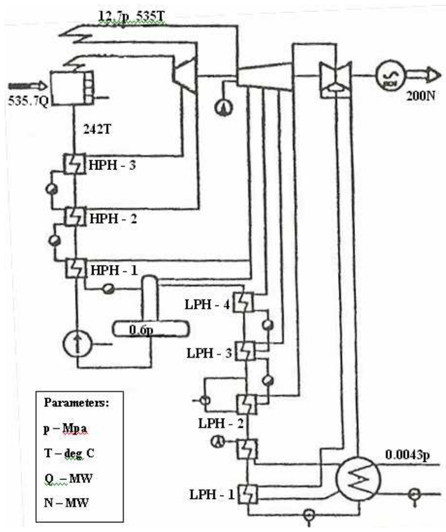

Figure 1: Schematic diagram of 200 MWe steam cycle power plant.

MATHEMATICAL MODEL AND NUMERICAL CALCULATIONS

Condenser Model

The design thermodynamic parameters, flows, and construction data considered for calculation of the condenser performance at varying operating conditions are: condensation pressure P=0.043 bar, cooling water inlet and outlet temperatures t_1/t_2=17/26 °C, minimum temperature difference δt_{min}=4.°C, cooling water mass flow rate 7229.3 kg/s, condensate steam mass flow rate 118.3 kg/s, surface area A=11420. m^2, pipes diameters d_o/d_i=0.03/0.028 m, pipe length L_p= 9 m and number of pipes per one bundle Np=842 pipes.

The model allows calculating the outlet cooling water temperature, condenser pressure for the given inlet cooling water's mass flow rate and temperature and mass flow rate of steam from turbine. The overall heat transfer coefficient may be calculated as follows [4, 12]:

The overall heat transfer coefficient U is given by:

$$U = \left(\frac{d_o}{d_i \alpha_w} + \frac{d_o \ln(d_o / d_i)}{2\lambda_p} + \frac{1}{\alpha_c} + \frac{1}{\alpha_f} + \frac{1}{\alpha_a} \right)^{-1} \tag{1}$$

Where various symbols are defined below:

α_W : It is the water side heat transfer coefficient and is calculated as follows:

$$\alpha_W = (\lambda_W / d_i)(0.023) \mathrm{Re}_W^{0.8} \mathrm{Pr}_W^{0.4} \tag{2}$$

$$\mathrm{Pr}_W = C_W \mu_W / \lambda_W, \tag{3}$$

$$\mathrm{Re}_W = (\rho_W V_W d_i) / \mu_W, \tag{4}$$

and $$V_w = m_w / (\rho_w \pi d_i^2 N_p / 8) \tag{5}$$

α_C: It is film wise condensation heat transfer coefficient which is corrected using Nusselt's relation given by

$$\alpha_c = 0.725 \left[\frac{\lambda_c^3 \rho_c^2 h_{fg} g}{\mu_c d_o (t_I - t_{t_o})} \right]^{0.25} \phi_1 \cdot \phi_2 \tag{6}$$

here ϕ_1 is the vapour shear correction factor. Fujii [4,12] recommended the following relation

$$\phi_1 = 1.4 \left[\frac{V_m (t_I - t_{t_o}) \lambda_c}{g d_o \mu_c h_{fg}} \right]^{0.05} \tag{7}$$

for $\phi_1 < 1$. Other wise equation (6) is used with ϕ_1=1.0. In equation (7) V_m stands for vapour approach velocity, calculated for maximum cross-sectional area of tubes bundle. ϕ_2 in equation (6) is the effect of condensate drips from one tube onto that of the next. Nusselt recommended the relation

$$\phi_2 = (1/N)^{0.25} \tag{8}$$

for a vertical stack of N-tubes. For the condenser under this analysis, the value of N is taken as 15 tubes. In equation (6) the physical properties, are evaluated (as a liquid) at the mean film temperature [$t=(t_{t_o}+t_I)/2$], except h_{fg}, which is found at saturation temperature.

α_a: It is heat transfer coefficient of air, and is evaluated by Berman and Fuks relation [4] given by:

$$\alpha_a = \frac{aDA}{d_o} \mathrm{Re}_s^{1/2} \left(\frac{p_m}{p_m - p_s} \right)^b (p_m)^{1/3} \left(\frac{\rho_s h_{fg}}{t_m} \right)^{2/3} \left(\frac{1}{(t_m - t_I)^{1/3}} \right) \tag{9}$$

Where a=0.52 and b=0.7 for $\mathrm{Re}_S < 350$,
a=0.82 and b=0.6 for $\mathrm{Re}_S > 350$

and $$\mathrm{Re}_s = \rho_s d_o V_m / \mu_s \tag{10}$$

In equation (9) t_m is the saturation temperature at p_s , and p_m is the mixture pressure obtained from:

$$p_m = p_a + p_s \tag{11}$$

with p_a,as the air pressure and is obtained from

$$p_a = \left(\varepsilon \frac{1}{v_s} \right) R_a t_m \tag{12}$$

here ε is the air-to-steam ratio, and is taken as 0.008 [4], DA in equation (9) is given by

$$DA = \lambda_a / \rho_a C_a \tag{13}$$

α_f : It is the heat transfer coefficient due to fouling, and is taken as 28.5 kW/m^2°C.

The overall heat balance through the considered media, is given by:

$$d_i \alpha_w (t_{t,i} - t_{wa}) = d_o \alpha_c (t_I - t_{t,o}) = d_o \alpha_a (t_{sat} - t_I) = d_o K (t_{sat} - t_{wa}) \tag{14}$$

where $t_{t,i}$, $t_{t,o}$ are the surface temperatures of inner and outer tube, t_{wa} is average cooling water temperature and t_I interface temperature between steam and condensate.

The pressure drop in cooling water side is given by:

$$\Delta p = \Delta p_n \left(\frac{m_w}{m_{w,n}} \right)^2 \tag{15}$$

Exergy Model

The exergy balance applied to the considered power plant as follows [1, 13, and 14]: Turbines, pump shaft work and electrical energy are full transfers of exergy. The energy flow carried away from the system in a useless form and is not recovered by any method, not further used in the power plant, e.g. exhaust gases from boiler and outlet cooling water flow are taken as zero exergy (such exergy flows values are not included in the exergy balance equation for the considered component). The fuel energy supplied to the boiler is taken as full of exergy. In fact the energy efficiency is greater the exergy efficiency, the difference in around 4% [13, 14]. The fuel energy supplied to boiler at full load is 535.757 MW, and considered constant through the calculations of the present study.

The exergy destruction in *ith* component is based on the exergy balance for incoming and outgoing flows as:

$$\delta\pi_i = \sum_{i=1}^{N} a_i Ex_i \tag{16}$$

Where, coefficient a_i determines the exergy flow entering and leaving the balance node, and:

$$Ex_i = m_i\left[\ \left(h_i - h_o\right) - T_o\left(s_i - s_o\right)\ \right] \pm Q \tag{17}$$

total exergy destruction of the whole power plant is given by:

$$\Pi = \sum \delta\pi_i \tag{18}$$

the exergy efficiency is calculated using [11,13,14]:

$$\eta_{Ex} = \left(1 - \frac{\Pi}{Ex_f}\right) * 100\% \tag{19}$$

Power Plant Model

The model allows carrying out the influence of the changes in the condenser's inlet cooling water temperature and flow rate on the electrical power output of the turbo set and on the change of exergy destruction in the particular component of the unit. The exergy destruction is calculated not only for the condenser in which changes in the heat transfer coefficient take place, but also in all other components of the power plant under consideration. The model consists of the following segments:

MBE: A set of mass-energy balance equations. This segment enables the solution of the balance problem, that is, evaluation of mass and energy flows for input thermodynamic parameters. The segment consists of equations of balances, most of which are linear.

MPT: A set of equations involving thermodynamic parameters. This segment allows the evaluation of thermodynamic parameters for given mass flow. This segment includes equations describing heat transfer, losses in pressure of the power plant components and changes of the turbine parameters. These equations are mostly nonlinear.

MPP: A set of equations representing the performance of the thermal power plant. This segment allows the evaluation of the exergy destruction, net power plant heat rate, energy consumption necessary for self needs of the power plant. The values obtained in MBE and MTP are used in MPP.

The detailed equations are not presented here in view of brevity. The problem is attempted by means of a general method of defining the state of thermal system at off-design operating conditions as suggested in reference [2, 11]. A simplified scheme of the solution algorithm is shown in Figure (2).

With the assumptions made and starting guess of the solution, the operations that take place in the *jth* iterative process are:

- The evaluation of energy flows Mj base on MBE model, and
- The evaluation of the thermodynamic parameters TPj, based on MTP model and Mj.

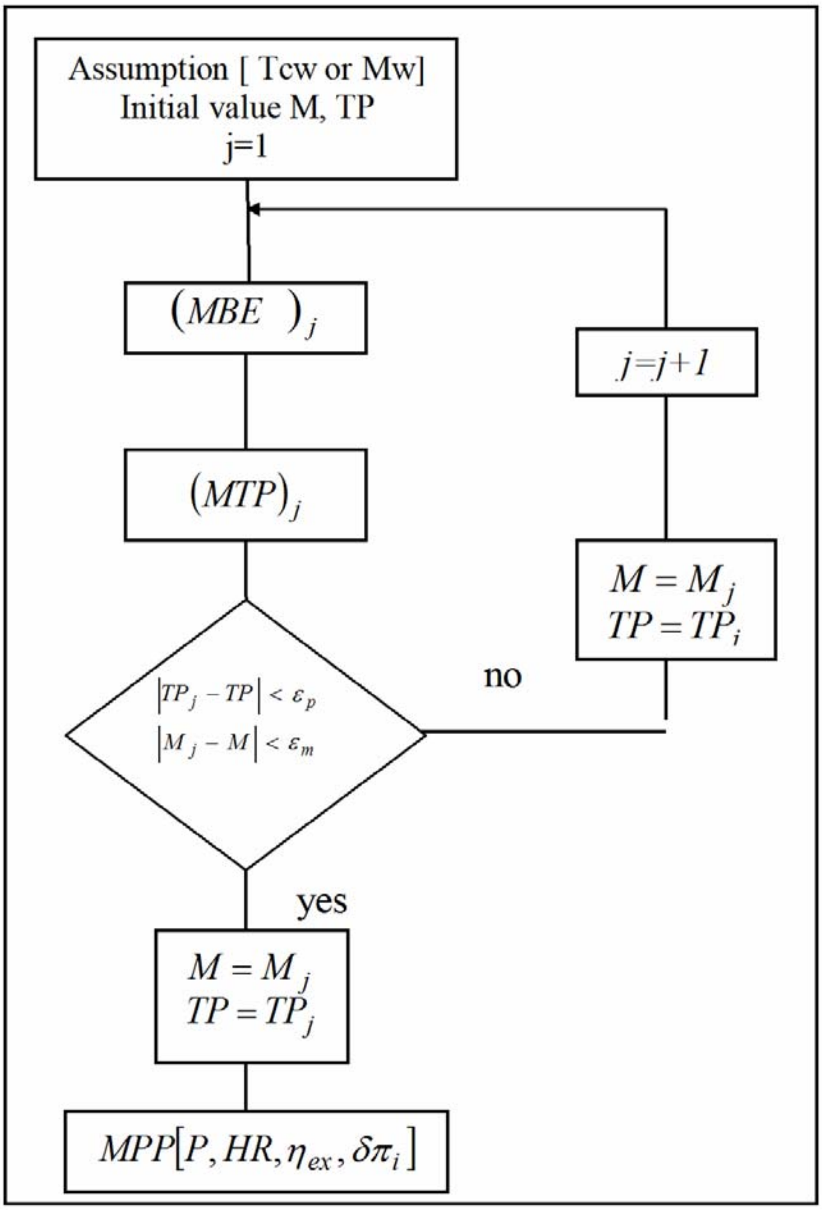

Figure 2: The scheme of solution algorithm

These operations are repeated in each iteration till significant improvement in the solution is noticed. This is the convergence criterion of the method. Next, the balance problem is solved again. Finally, the quantities defined by MPP model are determined. A computer program is developed consisting of subprograms, which consider the off-design operating conditions for the power plant components, namely: turbine stages, steam control-governor, boiler, condenser, coolers, feedwater heaters and boiler feedwater pump.

RESULTS AND DISCUSSION

The effect of cooling water conditions (temperature and flow rat) on steam plant exergy destruction and performance has been studied. For this purpose a full computer program is constructed using models considered in this paper. The calculations have been carried out for a fixed boiler capacity at 535.76 MW. Also the range of study for cooling water inlet temperature and flow rate is considered from 10 to 30 °C and 60% to 100% of design value respectively. When the cooling towers are used, the cooling water temperature is equals the ambient cooling air temperature. The results are presented in Figures 3 through 9.

Figures (3- 4) shows the condenser pressure and temperature variation at various inlet cooling water temperatures and different flow rates. The decrease of the flow rate increases the condenser pressure and temperature. These condenser pressure and temperature values increase as the cooling water temperature increases for the same flow rate.

The effects of cooling water conditions on both power output and heat rate are clearly shown in figures (5-6). The obtained variation is about 17 MW and 700 kJ/kWh respectively.

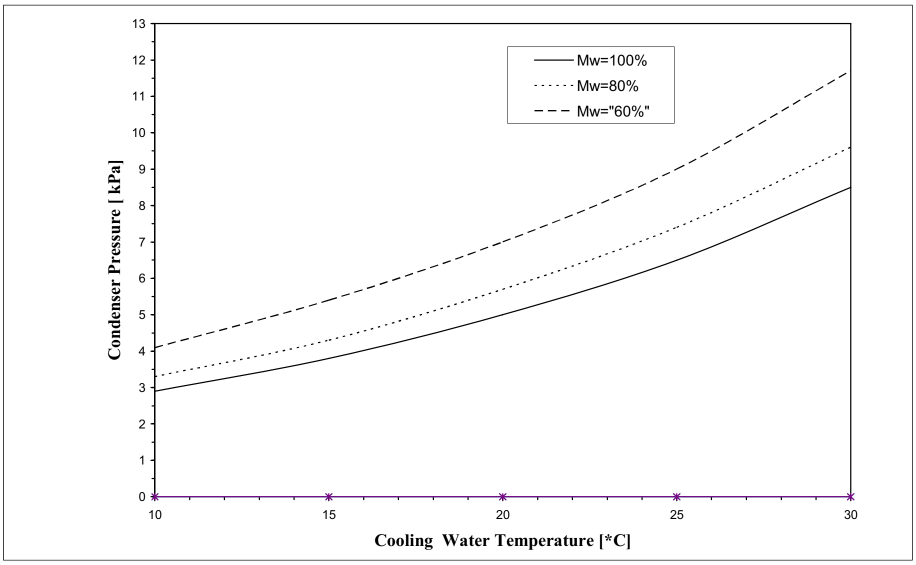

Figure 3: Variation of condenser pressure with inlet cooling water temperature at different cooling flow rates.

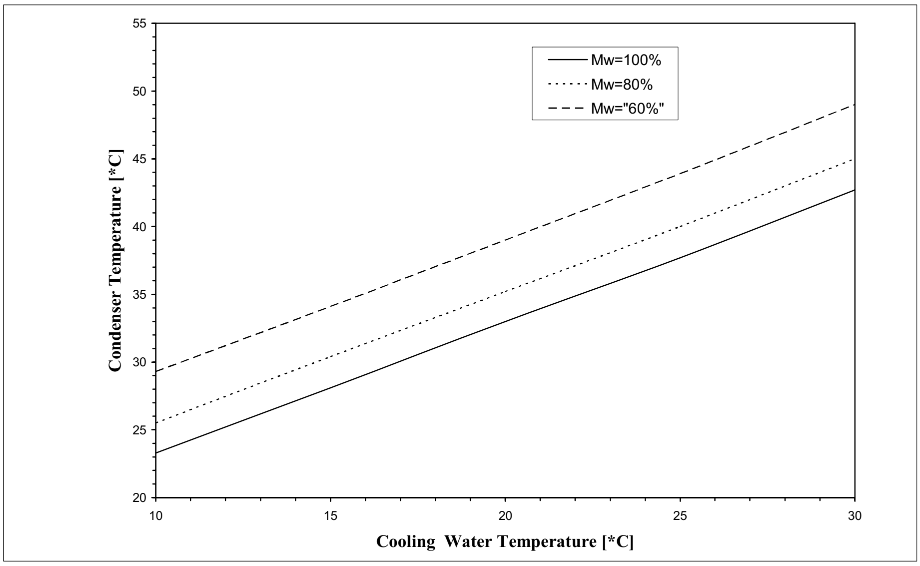

Figure 4: Variation of condenser temperature with inlet cooling water temperature at different cooling flow rates.

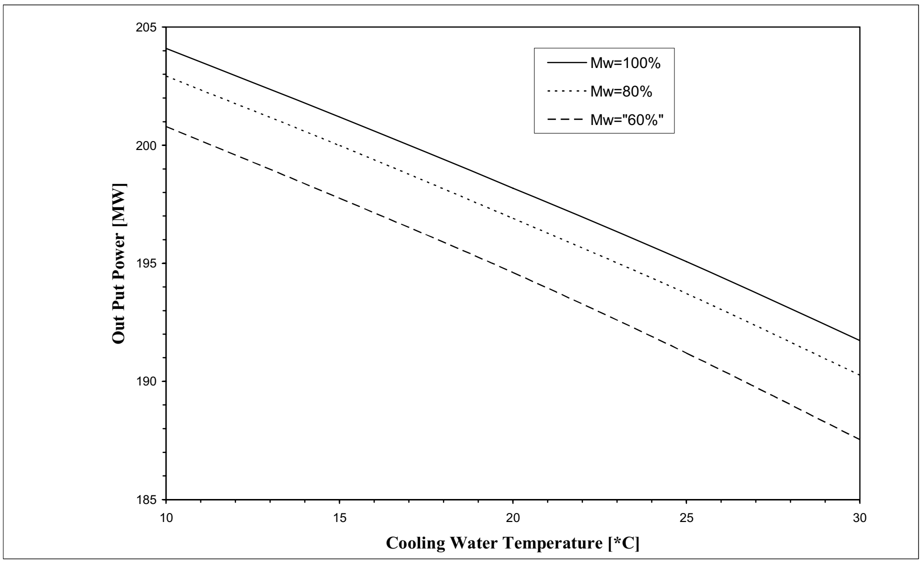

Figure 5: Variation of out-put power with inlet cooling water temperature at different cooling flow rates.

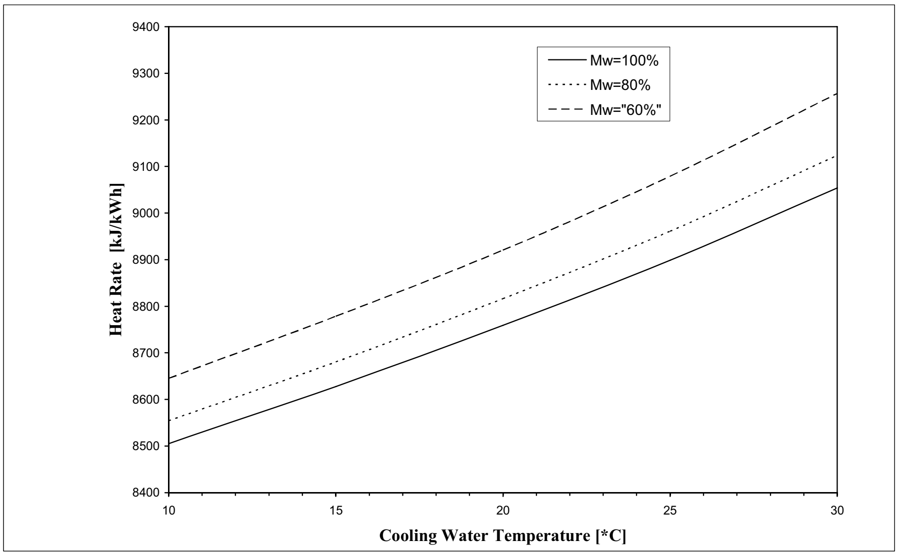

Figure 6: Variation of heat rate with inlet cooling water temperature at different cooling flow rates.

Figures (7-8) shows the exergy efficiency and total exergy destruction variation at various inlet cooling water temperatures and different flow rates. The increase of the flow rate increases the exergy efficiency and decreases the total exergy destruction. The same effects on the exergy efficiency and total exergy destruction were observed, as the cooling water temperature decreases for the same flow rate. It is seen that the exergy efficiency increase by more than 3.0% and total exergy destruction decrease result in around16 MW (the exergy destruction in steam traps, coolers and small mixing streams connections of the power plant are not included in the total exergy destruction value).

Figure (9) shows the variation of exergy destruction in some selected components of the power plant at various inlet cooling water temperatures and constant flow rate (100% of design value). It may be noticed, that the exergy destruction increases with the increase in the cooling water temperature, in the boiler and condenser. However, in the turbine and feedwater heaters a decrease trend of the exergy destruction may be noticed which is due to the decrease in the turbine capacity and in the finite temperature difference in the feedwater heaters. For an increase in water temperature from 10 to 30 ^{0}C the change in exergy destruction result in around: 15 MW, 0.7 MW, -1.8 MW and -1.5 MW in the boiler, condenser, turbine and feedwater heaters respectively.

The reduced exergy destruction in the feedwater heaters can be attributed to the reduced heat transfer temperature difference, because of the feedwater inlet temperature to the heaters rises due to the condenser pressure increase. The large exergy destruction increase in the steam generation unit is attributed to the maximum exergy destruction on carrying out combustion and heat transfer at large temperature different.

The explanation behind the above result may be given after the investigation of the all results, which shows that, when the condenser pressure increase, feedwater temperature increase, turbine bled steam reduces and reheater steam flow increases. The combined effect causes additional heat transfer followed by additional exergy destruction in the boiler, and the exergy efficiency reduces. The increased exergy destruction in the condenser can be attributed to the larger heat transfer carried out by condenser. The exergy destruction in the case of steam turbine also decreases with the increasing cooling water temperature. This is due to higher condenser back pressure (higher enthalpy) resulting in lower power out put.

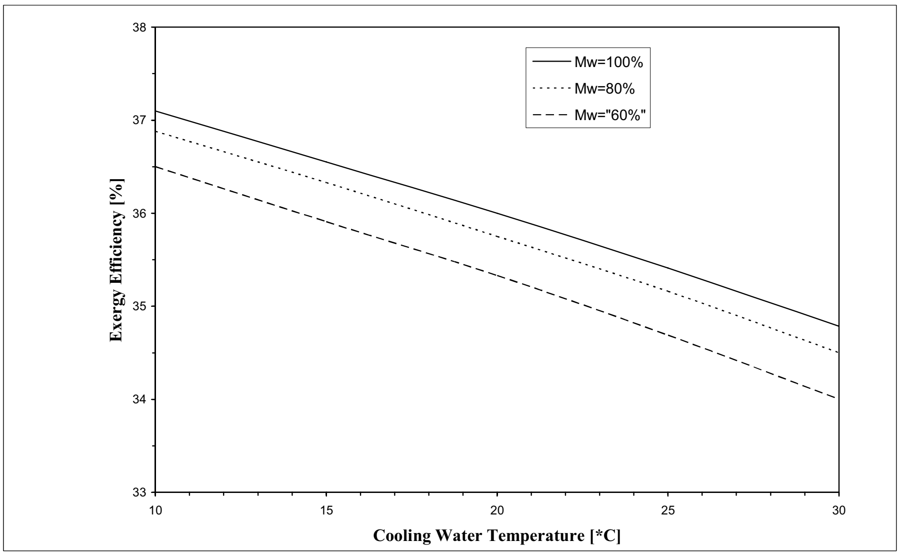

Figure 7: Variation of exergy efficiency with inlet cooling water temperature at different cooling flow rates.

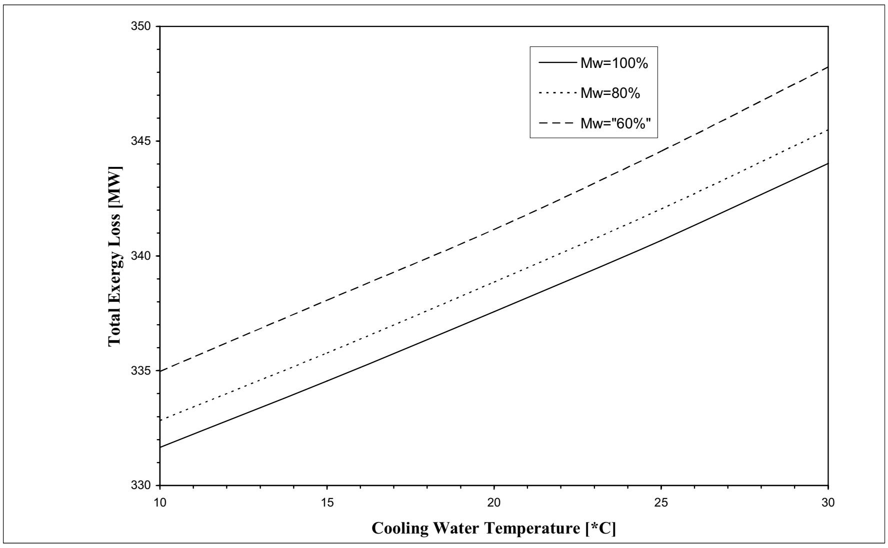

Figure 8: Variation of total exergy destruction with inlet cooling water temperature at different cooling flow rates.

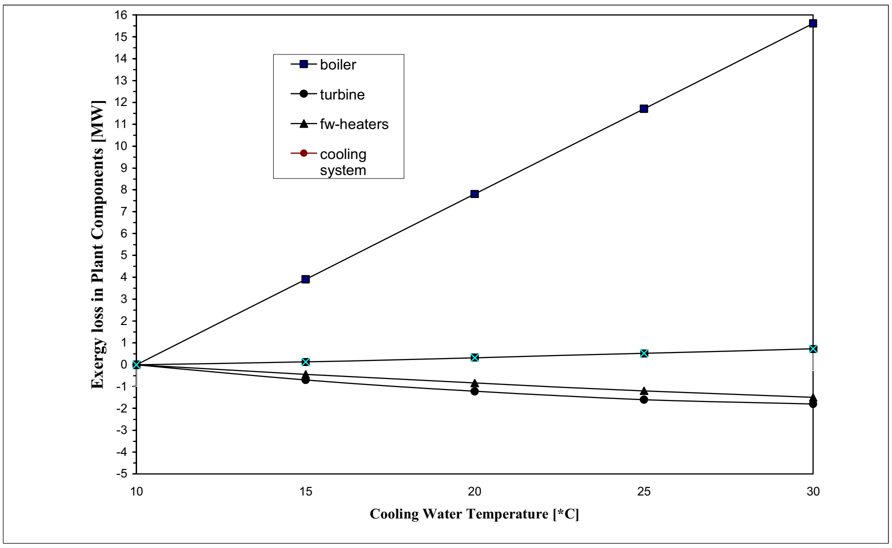

Figure 9: Variation of exergy destruction in plant components with inlet cooling water temperature at different cooling flow rates.

CONCLUSIONS

In the present work, an exergy analysis has been carried out to investigate the effect of the cooling water conditions on performance and exergy destruction of a 200 MW steam power plant. The plant performance was investigated at constant full load boiler capacity with variations in the cooling water's temperature and flow rate. A detailed off-design study has also been carried out to account for the contribution of irreversibilities in different components of the power plant. The results show that the condenser pressure is one of the important parameters influencing the plant exergy destruction and performance, which depends on the local cooling water conditions. The above changes in the cooling water conditions may result in:

(i) 17 MW increase in the total exergy destruction, which equals to17 MW decrement in the electrical power (1 kW exergy equal 1 kW electrical energy [13,14])

(ii) 700 kJ/k Wh increase in heat rate, which in turn increases fuel consumption,

(iii) More than 3% reduction in exergy efficiency.

. Finally, the present study has enabled us to identify sites where destruction of useful energy takes place in a relatively different steam plant capacity. The above results show that the cooling system of a plant design plays a significant role in the performance and fuel utilization of the power plants. The results obtained are useful in the decision-making process in planning of adding small dry or wet cooling towers beside the power plant.

Condenser inlet cooling water temperature and flow rate have been found to have a significant impact on condenser back pressure, and subsequently plant performance. Application of these, results in an improved condenser pressure, a decrease in plant heat rate and decrease in yearly fuel consumption. Overall, plant performance is improved. The important of a proper design and maintenance of condenser is evident from the above noted variation.

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NOMENCLATURE

d pipe diameter, m
Ex exergy flow, kW
g gravitational constant, m/s^2
h specific enthalpy, kJ/kg
HR heat rate, kJ/kWh
m mass flow rate, kg/s
Mw flow rate of cooling water, kg/s
Pr Prandtl number
Re Reynolds Number
P_e electric power, MW
Q energy flow kW
s - specific entropy, kJ/kg.K
R_f fouling resistance, $(m^2.°C)/kW$
R_p tube-side resistance, $(m^2.°C)/kW$
T temperature, °C
V velocity, m/s
U overall heat-transfer coefficient, $kW/(m^2.K)$

Greek Letters

η_{Ex} exergy efficiency (%)
$\delta\pi_i$ exergy destruction (kW)
Π total exergy destruction, (kW)
α heat-transfer coefficient, $kW/(m^2.°C)$
λ thermal conductivity, W/m.°C
ρ density, kg/m^3
μ viscosity, N.s/m

Subscripts

A *air*
c condensing
f fuel
i inside
m mixture
o outside, or dead state.
p pipe.
s steam
sat saturation.
w water.