THERMAL SCIENCE, Year 2012, Vol. 16, Suppl. 2, pp. S431-S442

ANALYSIS OF THE EVAPORATIVE TOWERS COOLING SYSTEM OF A COAL-FIRED POWER PLANT

by

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Original scientific paper

The paper presents a theoretical analysis of the cooling system of a 110 MW coal-fired power plant located in central Serbia, where eight evaporative towers cool down the plant. An updated research on the evaporative tower cooling system has been carried out to show the theoretical analysis of the tower heat and mass balance, taking into account the sensible and latent heat exchanged during the processes which occur inside these towers. Power plants which are using wet cooling towers for cooling condenser cooling water have higher design temperature of cooling water, thus the designed condensing pressure is higher compared to plants with a once-through cooling system. Daily and seasonal changes further deteriorate energy efficiency of these plants, so it can be concluded that these plants have up to 5% less efficiency compared to systems with once-through cooling. The whole analysis permitted to evaluate the optimal conditions, as far as the operation of the towers is concerned, and to suggest an improvement of the plant. Since plant energy efficiency improvement has become a quite common issue today, the evaluation of the cooling system operation was conducted under the hypothesis of an increase in the plant overall energy efficiency due to low cost improvement in cooling tower system.

Key words: *cooling tower, condensing pressure, energy efficiency*

Introduction

Circulating cooling water system is an important aspect of a thermal power plant that affects the investment decision, plant siting, and the performance of the plant. Circulating water requirement in a thermal power plant is a major water resource issue and has tremendous effect on the surrounding environment, population, and animal and aquatic life. For cooling its condenser, steam power plants use basically two types of cooling systems: open-cycle and closed-cycle [1]. Open-cycle or once-through cooling systems withdraw large amounts of circulating water directly from and discharge directly to streams, lakes, reservoirs, and embayment through submerged diffuser structures or surface outfalls. Open-cycle systems depend upon adequate cool ambient water to support generation at full capacity.

Closed-cycle cooling systems (fig. 1) transfer waste heat from circulating water to air drawn through cooling towers. The fresh water quantity requirement is only 5% of the open-cycle system.

Cooling towers are heat exchangers that are used to dissipate large heat loads to the atmosphere. Although cooling towers can be classified in several ways, the primary classification is into dry towers or wet towers, while some hybrid wet-dry combinations also exist. Subclassifications can include the draft type and/or the location of the draft relative to the heat transfer medium, the type of heat transfer medium, the relative direction of air movement, and the type of water distribution system.

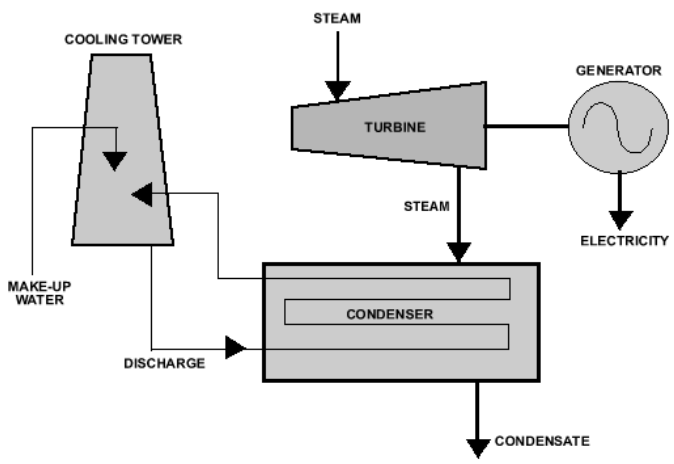

Figure 1. Typical closed-cycle cooling system [2]

Conventional wet cooling towers depend on evaporative heat exchange and require a continuous source of freshwater to replace evaporation losses of the system. Mechanical draft cooling towers are installed in the examined plant: the air enters the tower through a fan or several fans and passes through the packing onto which the water is sprayed (fig. 2). The water flows through the packing, in contact with the flow of air and is collected in a cold water basin at the bottom of the tower. The warm, moist air leaves from the top of the tower. Towers of this kind give good thermodynamic efficiency and the relatively high air velocity out of the tower gives fewer tendencies to fog formation in the vicinity. These towers are also easy to maintain. This choice increases the costs related to the installation and the operation, but in this way, the location of the plant does not need the nearby water basins, with huge capacity, to guarantee water supplies to the towers.

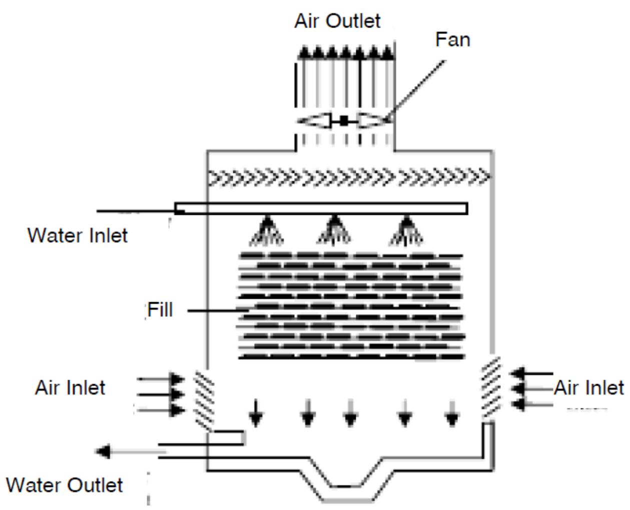

Figure 2. Induced draft cooling tower

Whether a power plant cooling system is closed- or open-cycle, energy efficiency of the power plant is determined by its cold end performances. Heat rate and generated power output of a turbo generator unit strongly depend on the condenser pressure. The pressure in the surface steam condenser will depend on condenser design, an amount of latent heat to be removed, cooling water temperature and flow rate, maintenance of the condenser and air removal system. At any given time these operating conditions will determine the relationship between the heat rate and the power output.

In an ideal situation, when the venting system properly removes air from the steam condenser, the achievable condensing pressure is determined by temperature of the cooling water. For the steam power plant with once-through cooling system, cooling water temperature is determined by natural water source (*i. e.* river) temperature. This means that cooling water temperature is changing with weather conditions in particular region, and cannot be changed in order to achieve better condenser performances (*i. e.* higher vacuum in the condenser) [3]. The problem is more severely set for plants with a closed-cycle cooling system. Dry bulb temperature, wet bulb temperature, atmospheric pressure, flow rate of the circulating water , characteristic of cooling tower fill, resistance coefficient of the parts in a tower, and working conditions of a water distribution system and so on, can affect the outlet water temperature of the cooling tower [4].

The thermodynamic model of reference of the chosen power plant

Evaporative water cooling decreases its temperature during simultaneous execution of two physical processes [5, 6, 7]:

- Heat convection due to temperature difference of water and air, and
- Evaporation of water in atmospheric air due to differences of concentrations.

Heat transfer by convection is determined by Newton's law, in the relation:

$$q_\alpha = \alpha(t_\mathrm{w} - t_\mathrm{a}) \tag{1}$$

Heat evaporation of water flux is:

$$q_\beta = r_\mathrm{w}\mathrm{d}q_{m\mathrm{w}} \tag{2}$$

The total heat flux of cooling water is given by:

$$q = q_\alpha + q_\beta \tag{3}$$

The theoretical limit of water cooling temperature is wet bulb temperature, t_WR [K]. In practice, the actual temperature of water cooling is higher than wet bulb temperature t_WR [K] for (5 -10) K. The cooling characteristic of the cooling tower is represented by the Merkel eq. [8]:

$$\frac{B_{xv}V}{G} = \int_{t_1}^{t_2} \frac{c_w}{h^{"} - h}\mathrm{d}t \tag{4}$$

The Merkel equation primarily says that at any point in the tower, heat and water vapor are transferred into the air due (approximately) to the difference in the enthalpy of the air at the surface of the water and the main stream of the air. Thus, the driving force at any point is the vertical distance between the two operating lines. Therefore, the performance demanded from the cooling tower is the inverse of this difference.

For technical calculations, it can be considered with sufficient accuracy that unsaturated moist air obeys the laws of a mixture of ideal gases.

The absolute air humidity (ω) is the actual mass of water vapor present in 1kg of dry air and can be calculated as:

$$\omega = 0.622 \frac{p_{wp}(t)}{p_{ap}} = 0.622 \frac{p_{wp}(t)}{p_a - p_{wp}(t)} \tag{5}$$

The relative humidity (φ) of an air-water mixture is defined as the ratio of the partial pressure of water vapor in the mixture to the saturated vapor pressure of water at a prescribed temperature.

$$\phi = \frac{p_{wp}(t)}{p_{ws}(t)} = \frac{\omega}{(0.622+\omega)} \frac{p_a}{p_{ws}(t)} \tag{6}$$

Moist air is a mixture of dry air and water vapor. The enthalpy of moist and humid air includes the enthalpy of the dry air - the sensible heat – and the enthalpy of the evaporated water - the latent heat. Specific enthalpy - h [Jkg^{-1}] of moist air is defined as the total enthalpy [J] of the dry air and the water vapor mixture - per unit mass (kg) of moist air.

Specific enthalpy of moist air on the tower inlet can be expressed as:

$$h = h_a + xh_w = 1.0048t + \frac{0.622 p_{ws}(t)\phi}{p - p_{ws}(t)\phi}(2500 + 1.86t) \tag{7}$$

where h – specific enthalpy of moist air [kJ kg^{-1}], h_a – specific enthalpy of dry air [kJ kg^{-1}], x – humidity ratio [kg kg^{-1}], h_w – specific enthalpy of water vapor [kJ kg^{-1}], and t – air temperature.

The saturated vapor pressure can be calculated by different methods: using Goff and Gratch formulation [9], British Standard BS 4485 formulation [10], Richards and P. R. Lowe [11, 12], by direct integration of the Clausius-Clapeyron equation and other methods. In this paper, the IAPWS-97 method is used [13].

Determination of the volumetric heat and mass transfer coefficients is done for the cooling devices in which water is sprayed through nozzles or in the form of drops flowing on the grid. The size of surface cooling, which refers to the active volume unit, is changed in this case, depending on the amount of water entering the cooler and air speed, which is reflected in the value of the heat and mass transfer coefficients. Criteria equations do not include changing the surface of liquid, so in the absence of exact methods it is common to use the purely empirical formulation. In [8] empirical formula is given as

$$\beta_{xv} = A(w\rho)^m q_1{}^n, \ \left[\text{kg m}^{-3}\text{ h }\left(\text{kg kg}^{-1}\right)\right] \tag{8}$$

where $q1$ – specific mass flow rate [kg m^{-2} s], w – air velocity, and A=1050, m=0.53, and n=0.39 – constants.

For the analytical solution of this integral, it is necessary to find appropriate dependence of specific enthalpy of saturated air and temperature (the parabolic dependence):

$$h^{"} = 0.019t^2 - 1.575t + 40, \ [\text{kJ kg}^{-1}] \tag{9}$$

For the calculation of heat exchange in the cooling tower in this paper, parabolic dependence of the enthalpy of saturated air and temperatures is applied. Calculated enthalpy of saturated air shows very little deviation from the exact values obtained from psychrometric chart, as it is shown in fig. 3.

Solving the Merkel integral and using geometrical parameters of the cooling tower, the following equation can be written:

$$\begin{gathered} 0.045t_{w,out}^2 + \left(0.03t_{w,in} + \frac{c_w}{2\lambda} + \frac{Gc_w}{V\beta_{xv}} - 0.785\right)t_{w,out} + \\ + \left(0.045t_{w,out}^2 - \left(0.785 + \frac{c_w}{2\lambda} + \frac{Gc_w}{V\beta_{xv}}\right)t_{w,in} + h_{in} + 40\right) = 0 \end{gathered} \tag{10}$$

The solution of this equation is temperature of the water on the outlet of the cooling tower, as function of atmospheric air parameters, inlet water temperature, flow rate of the water and air flow rate. This dependence gives an opportunity for overall consideration of the influence of different parameters on the outlet water temperature, thus on the overall energy efficiency of the power plant.

For the simulation of the power plant cold end operation, the simulation model developed by authors and described in [3, 14, 15] was applied. This model provided modular structure so that a plant model could be quickly adapted to represent various power plant configurations. The mathematical model of the proposed thermodynamic problem formulation was created as a type of steady state simulation. Flowsheeting formulation (given all input information, determines all output information) was developed by applying the conservation laws for mass and energy balance [16, 17].

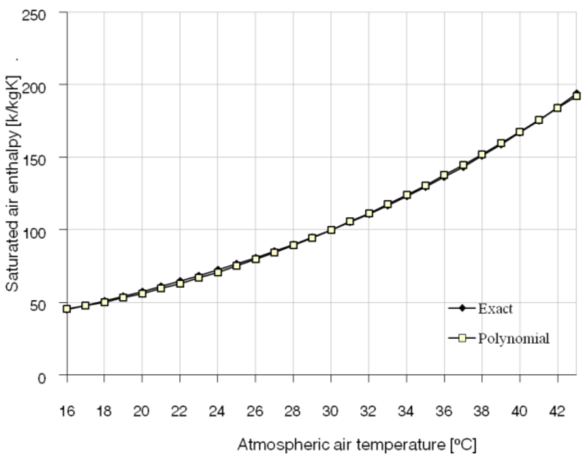

Figure 3. Saturated air enthalpy

The examined coal-fired power plant

The study was done regarding the cooling system of 110MW coal-fired power plant working under the Rankine Cycle, TPP "Kolubara A", unit A5, located in Donji Crljeni, Serbia.

The cooling system is made with two condensers, each of which has an exchange surface of 3450m^2. The chambers are crossed by cooling pipes inside of which circulate the water that is afterwards sent to the evaporative towers. In the examined power station, the towers are made of 10 cells which are structurally the same, but only 8 are in function, fig. 4.

Figure 4. Cooling towers of the power plant

Figure 5. Nozzles and fill packing

The hot water arriving from the condensers is taken to the higher part of the towers, at a height of about 7.5m. The contact surface between the two fluids must be maximized in order to allow the heat exchange. To meet this requirement, water is sprayed by a system of nozzles and then falls into the filling made of plastic materials, fig. 5.

In order to describe correctly how the plant works, according to the various environmental conditions to which it is subjected, several reference days were chosen, and specifically one day in summer, during which the station worked without being stopped or restarted. Herein, the results for the summer conditions are given, since it is a critical period for achieving the optimum efficiency of the plant.

The results of the cooling tower operation simulation

Based on the mathematical model, construction and meteorological data for the reference plant, the temperature of the cooled water change due to different parameters was obtained. In order to clearly represent the influence of the temperature and relative humidity of the atmospheric air on the temperature of the water at the cooling tower outlet, the water inlet temperature and hydraulic load were considered as constant. The results are given in fig. 6.

It is obvious that with dry bulb temperature and increasing relative humidity (*i. e.* increasing wet bulb temperature), the capability of the cooling tower decreases. The decrease is more severe for the higher atmospheric air temperature, that is, in the summer period. This means that in the summer the condenser cooling water temperature will be higher, and overall energy efficiency of the plant will be decreased, due to change in the condenser pressure [4].

In order to determine the daily change of the temperature of the water cooled in the cooling tower, one average summer day was chosen. Relevant parameters of the atmospheric air were measured every hour for the chosen day and they are given in tab. 1.

It is important to highlight the fact that temperature of water entering the cooling tower is not constant during the day, due to changes in the condenser operating conditions, together with changes in the cooling tower operating conditions.

In this analysis, the input parameters in the simulation of the cooling tower operation were atmospheric air temperature and humidity, but also the temperature of the cooling water leaving the condenser.

Due to the change in air parameters, temperature of the cooled water will change, thus, the inlet temperature of the cooling water in the condenser is also changed. This change leads to change in condenser operating conditions, which will influence both overall energy efficiency of the plant and inlet water temperature in the cooling tower.

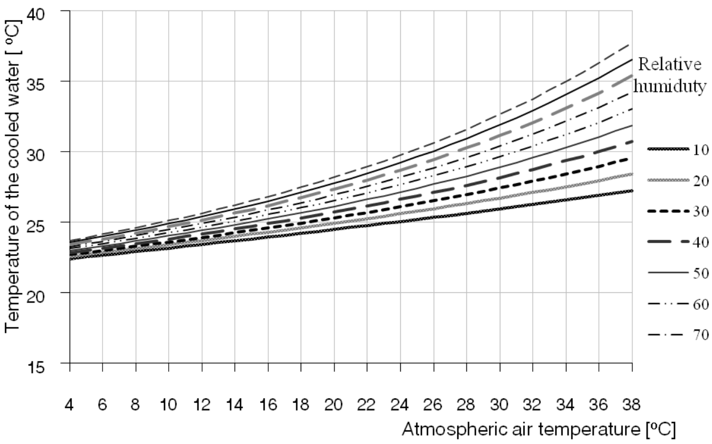

Figure 6. Temperature of the cooled water in the cooling tower unit A5, TPP "Kolubara A"

Table 1. TPP"Kolubara A", atmospheric air parameters, 15.08.2008.

Time(CEST):	Temperature	Dew point	Rel. humidity	Pressure
00:00 AM	27.0°C	13.0°C	42%	1012 hPa
1:00 AM	26.0°C	14.0°C	47%	1012 hPa
2:00 AM	25.0°C	13.0°C	47%	1011 hPa
3:00 AM	23.0°C	14.0°C	57%	1011 hPa
4:00 AM	23.0°C	14.0°C	57%	1010 hPa
5:00 AM	25.0°C	14.0°C	50%	1009 hPa
6:00 AM	25.0°C	14.0°C	50%	1009 hPa
7:00 AM	26.0°C	14.0 C	47%	1009 hPa
8:00 AM	28.0°C	14.0°C	42%	1009 hPa
9:00 AM	31.0°C	14.0 C	35%	1009 hPa
10:00 AM	32.0 C	14.0°C	33%	1009 hPa
11:00 AM	34.0 C	15.0°C	32%	1008 hPa
12:00 PM	35.0°C	13.0°C	26%	1008 hPa
1:00 PM	36.0 C	14.0°C	27%	1007 hPa
2:00 PM	38.0 C	12.0°C	21%	1006 hPa
3:00 PM	38.0 C	10.0°C	18%	1005 hPa
4:00 PM	38.0°C	10.0°C	18%	1005 hPa
5:00 PM	38.0 C	12.0°C	13%	1003 hPa
6:00 PM	37.0 C	11.0°C	21%	1003 hPa
7:00 PM	35.0°C	11.0°C	23%	1003 hPa
8:00 PM	34.0°C	12.0°C	26%	1002 hPa
9:00 PM	33.0°C	13.0°C	29%	1002 hPa
10:00 PM	32.0 C	13.0°C	31%	1003 hPa
11:00 PM	31.0 C	11.0°C	29%	1002 hPa

It is important to highlight the fact that temperature of water entering the cooling tower is not constant during the day, due to changes in the condenser operating conditions, together with changes in the cooling tower operating conditions. In this analysis, the input parameters in the simulation of the cooling tower operation were atmospheric air temperature and humidity, but also the temperature of the cooling water leaving the condenser. Change of temperature of the water leaving the cooling tower *i. e.* entering the condenser is given in fig. 7.

Figure 7. Temperature of the cooled water during one average summer day

As the climatic parameters are constantly changing without any established rules, the most realistic picture of the impact of these changes in plant operation can be obtained using the measured air temperature and relative humidity for the one year period. For one year, air temperature and relative humidity were measured every day at 00:00, 3:00, 6:00, 12:00, 15:00, 18:00 and 21:00. For each parameter the mean value for each day was obtained, and these mean values were used in the simulation of plant cold end. Cooling tower simulation output parameter was the cooled water temperature. This value was then taken as the input value for the condenser operation simulation. For each day of the considered period for the given weather conditions, condensation pressure value was obtained. Annual change of the water temperature at the exit from the cooling tower is shown in fig. 8.

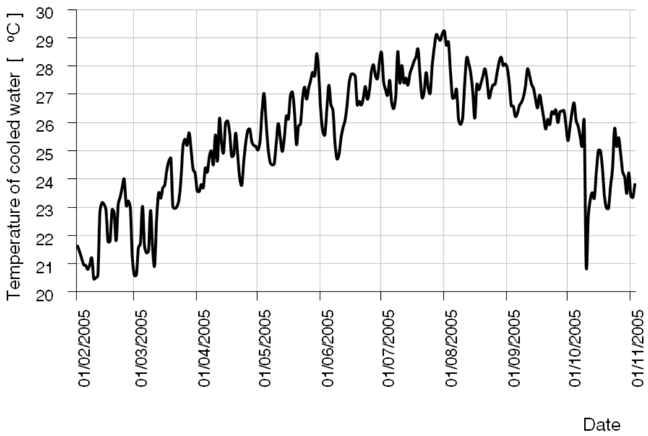

Figure 8. Temperature of the cooled water during one year

The lowest water temperature can be achieved in the winter months, while the increase in atmospheric air temperature in summer leads to reducing of cooling tower capability to remove waste heat, and an increase in water temperature at the exit from a tower (and thus at the entrance of the condenser) is inevitable. Specific hydraulic load of the fill has also great influence on the temperature of the cooled water. The change of the temperature with the change of hydraulic load for the examined cooling tower is given in fig. 9.

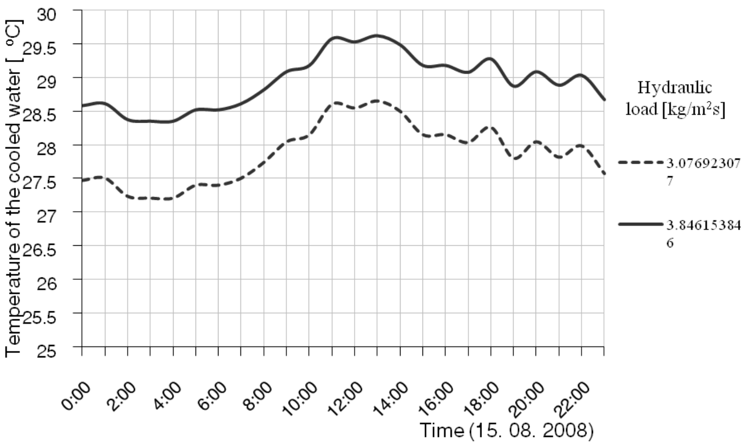

Figure 9. Temperature of the cooled water due to hydraulic load change

With the hydraulic load decrease, the temperature of the water cooled in the cooling tower also decreases, at any atmospheric air parameters. This means that if the same flow rate of the cooling water from condenser is distributed not to current 8 towers (hydraulic load is then 3.846 kg m^{-2} s), but ten existing towers (hydraulic load is 3.07 kg m^{-2} s), by decreasing hydraulic load of each tower, the lower temperature of the water can be obtained. Having in mind that two more cooling towers are already constructed on the plant site, it is obvious that relatively small investments can improve the cooling tower system.

Condenser performances due to cooling water temperature changes

Condenser heat transfer rate strongly depends on condensing pressure, cooling water flow rate and temperature. In an ideal situation, when the venting system properly removes air from the steam condenser, the achievable condensing pressure is determined by temperature of the cooling water, as it is mentioned above.

With cooling water temperature rising, the mean temperature difference in the condenser decreases, and condenser heat transfer rate for the same condensing pressure will also decrease. With cooling water temperature increasing, in order to maintain the designed heat transfer rate of the condenser, condensing pressure will increase. The turbine governor will increase the throttle and exhaust flows in order to set the generated power at the designed level, but with increasing of heat rate.

Condensing pressure dependence on cooling water temperature is obtained for the given water flow rate and steam load of the condenser. Steam load is considered constant, in order to obtain a clear illustration of this dependence, as it is shown in fig. 10. It is obvious that with cooling water increasing, pressure in the condenser will also increase.

Figure 11 shows the change of condensing pressure due to cooling water change for one summer day. It is obvious that the highest value of the condensing pressure will be reached at the warmest part of the day. For this calculation, the used measured values of the atmospheric air temperature and relative humidity are given in tab. 1.

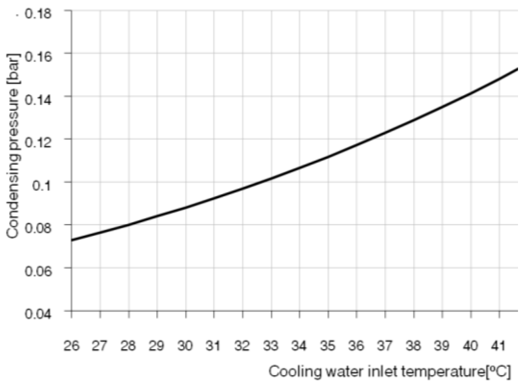

Figure 10. Condensing pressure change due to cooling water temperature change

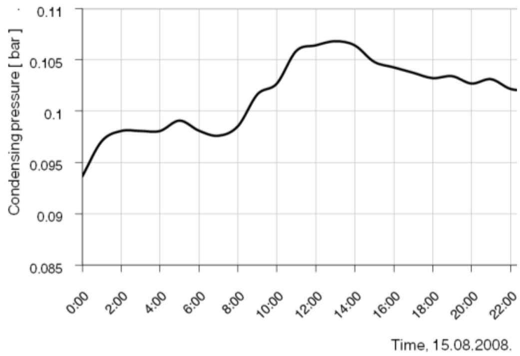

Figure 11. Condensing pressure change for one summer day

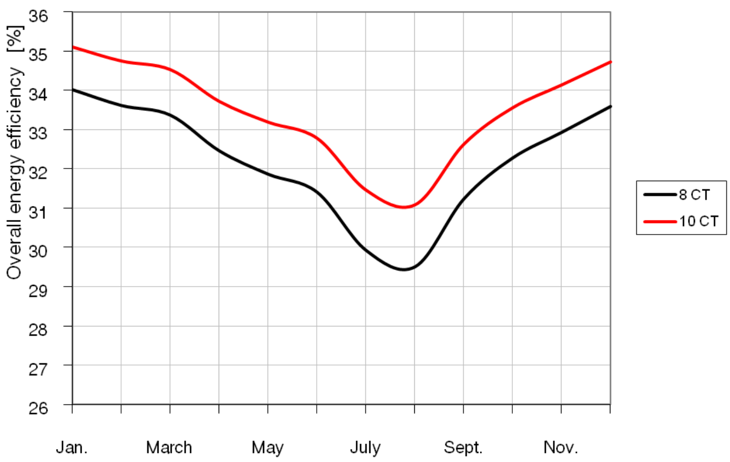

Figure 12. Annual change of energy efficiency of the plant A5 „Kolubara A“

Energy efficiency

Using the known assumption [18], that with increasing pressure in the condenser of 1kPa, efficiency decreases to 1.0-1.5%, and considering that in this particular case the reduction is 1.2%, the dependence of the energy efficiency (generated power divided by an amount of energy consumed) annual change of energy efficiency of the power plant with closed-cycle cooling system is obtained for two cases, the current state with 8 working cooling towers (8 CT) and the state with 10 cooling towers (10 CT), shown in fig. 12.

The annual change of energy efficiency is obtained using the data from [19] on the mean annual relative humidity and mean maximum and mean normal temperature of atmospheric air for 2009. In this diagram, it can be seen that in the summer, a serious drop in the energy efficiency of the power plant will occur due to the rise in the atmospheric air temperature, compared to energy efficiency at designed parameters. This fall of the efficiency comes as a result of increasing cooling water temperature at the entrance of the condenser *i. e.* the inability of the cooling tower to reject the sufficient amount of waste heat into the atmosphere at high atmospheric air parameters.

If ten, instead of eight, cooling towers are used for cooling, hydraulic load of every cooling tower will decrease, and as a consequence, condenser cooling water temperature will also decrease, resulting in overall energy efficiency increase of 1.5%, especially in summer.

Conclusions

A steam power plant strongly depends on its cold end operating conditions, where the condenser is the key of the heat exchange system. On the other hand, operating conditions of the cooling water system determine condenser operating conditions.

Temperature and relative humidity of the atmospheric air constantly change on a daily and annual levels. This leads to daily and annual changes of condenser cooling water temperature.

The mathematical model of the heat and mass transfer in the cooling tower was created using the data from the documentation obtained from the reference plant "Kolubara A". Based on the mathematical model, the cooling water temperature dependences on different parameters are obtained. In this paper, the daily change of cooling water temperature is given for a random summer day and at the annual level.

Power plants that use closed systems for cooling condensers are vulnerable to daily changes in working conditions due to changes in the parameters of atmospheric air. Change in cooling water temperature leads to change of condensing pressure in order to maintain production at desired levels at the expense of changed specific heat rate. This will cause changes of the energy efficiency on the daily and annual levels.

According to the results shown in this paper, for the reference plant, there is an easy low cost way to increase energy efficiency. Two of the ten constructed cooling towers are not currently in function. The small investments are required to put into operation those two cooling towers, in order to increase the overall energy efficiency of the plant by 1.5%.

Nomenclature

h – specific enthalpy of moist air, [kJ kg^{-1}]
h'' – specific enthalpy of saturated air, [kJ kg^{-1}]
p_a – moist air pressure
p_{wp} – water vapor partial pressure [Pa]
p_{ap} – dry air partial pressure [Pa]
p_{ws} – water vapor saturated pressure [Pa]
G – water flow rate through condenser, [kg s^{-1}]
c_w – *water* specific heat, [kJ kg^{-1} K^{-1}]
q_1 – fill hydraulic load [kg m^{-2}s]
ρ_w – water density [kg m^{-3}]
$t_{w,in}$ – water inlet temperature [ºC]
$t_{w,out}$ – water outlet temperature [ºC]
β_{xv} – the volumetric mass transfer coefficients [kg m^{-3} h (kg kg^{-1})]
V – fill volume [m^3]
λ – air/water flow rate ratio [-]

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Paper submitted: April 26, 2012.
Paper revised: June 6, 2012.
Paper accepted: September 12, 2012.