

## IMPACT OF THE COLD END OPERATING CONDITIONS ON ENERGY EFFICIENCY OF THE STEAM POWER PLANTS

by

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*The conventional steam power plant working under the Rankine Cycle and the steam condenser as a heat sink and the steam boiler as a heat source have the same importance for the power plant operating process. Energy efficiency of the coal fired power plant strongly depends on its turbine-condenser system operation mode.*

*For the given thermal power plant configuration, cooling water temperature or/and flow rate change generate alterations in the condenser pressure. Those changes have great influence on the energy efficiency of the plant. This paper focuses on the influence of the cooling water temperature and flow rate on the condenser performance, and thus on the specific heat rate of the coal fired plant and its energy efficiency. Reference plant is working under turbine-follow mode with an open cycle cooling system. Analysis is done using thermodynamic theory, in order to define heat load dependence on the cooling water temperature and flow rate. Having these correlations, for given cooling water temperature it is possible to determine optimal flow rate of the cooling water in order to achieve an optimal condensing pressure, and thus, optimal energy efficiency of the plant. Obtained results could be used as useful guidelines in improving existing power plants performances and also in design of the new power plants.*

*Key words: cold end, cooling water, condensing pressure, energy efficiency*

### Introduction

The need for electrical energy will certainly continue to grow, and it has become imperative to lower the cost of electricity and enhance the operational economy of the turbine unit. For the conventional steam power plant working under the Rankine Cycle, the steam condenser as a heat sink and the steam boiler as a heat source have the same importance for the power plant operating process.

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The cold end system of the steam turbine is composed of the last stage group that includes the low pressure cylinder, the condenser, the circulation water supply system, air extract system and, in case of closed-cycle cooling system, includes cooling tower. It can be divided into two subsystems – the condensate system and the circulation water system. The condenser is the key heat exchange system of the cold end.

The condenser operating conditions are of the great influence on the maximum generated power and the heat rate value. In the same time, the operating conditions of the cooling water system determine the operating conditions of the condenser.

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 or reservoirs through submerged diffuser structures or surface outfalls. An open-cycle system depends on the adequate cool ambient water to support the generation at full capacity. A closed-cycle cooling system transfers waste heat from circulating water to air drawn through cooling towers. Conventional wet cooling towers depend on evaporating heat exchange and require a continuous source of fresh water to replace evaporation losses. The ability of cooling towers to provide cold water to steam condensers of a thermoelectric unit decreases with increasing air temperatures and, in case of wet cooling tower, increasing humidity.

Heat rate and generated power output of a turbo generator unit strongly depend on the condenser pressure. The influence of the condensing pressure is similar in units operating in turbine-follow mode and for the units working in the boiler-follow mode [2]. In the turbine-follow mode the turbine governor is set to control the generator load. Steam boiler control system adjusts the fuel firing rate and other parameters so as to maintain the parameters of the steam at the turbine throttle as close as possible to their design values. In the boiler-follow mode, the fuel-firing rate is set at a fixed value. In this case, the turbine governor adjusts the throttle valve (and therefore generated power) so as to maintain the steam pressure at the throttle valve inlet at its design value.

On the turbo generator units controlled in turbine-follow mode, a rise in the condenser pressure will cause a rise in the enthalpy at the end point. This will result in the reduction in generated power. However, turbine governor will increase the throttle and exhaust flows in order to set the generated power at the designed level. Steam boiler control system will increase the fuel firing rate, and specific heat rate of the turbo generator unit will increase due to fuel firing rate increasing. Otherwise, on the fall in the condenser pressure, the reverse will occur, but only if it does not fall below the point where the exhaust annulus become choked. In the boiler-follow mode, a rise in condenser pressure will cause a drop in power for the same net heat input to the system, while the reverse will occur when the pressure falls.

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.

The designed characteristics of the condenser have a significant impact, but it is very expensive to replace them when the plant is operating. Still, with the constant science and technology development, today it is possible to improve the design for new plants, for

example, by using the thin-wall titanium tubing of the condenser [3]. It is also possible to make significant improvements in the existing operating plants, for example, by using the modular condenser replacement, with new condenser water boxes and titanium tube bundles [4].

Proper maintenance of the condenser is necessary for its proper operation. Constant cleaning of the condenser tubes is necessary to maintain a degree of fouling in acceptable level. Also, it is necessary to prevent any air leakage into the system and to assure constant air removal. Allowing non-condensing gases to accumulate increases thermal resistance on the shell-side, and thus the overall heat transfer coefficient of the tubes. The proper venting equipment has to remove any non-condensing gas from the system.

If the venting equipment is not adequate, the pressure in the condenser will rise, and efficiency of the turbine will be reduced [5]. In this case, venting equipment is limiting the vacuum in the condenser. If the venting equipment is properly sized, vacuum will be set by temperature of the cooling water and heat transfer rate. More efficient venting systems are also developed in order to reduce problems caused by non-condensing gases [6].

This paper focuses on the influence of the cooling water temperature and flow rate on the condenser performance, and thus on the power output, heat rate and energy efficiency of the power plant.

### The thermodynamic model of reference of the chosen power plant

This simulation model provides a modular structure so that a plant model quickly can be adapted to represent various power plant configurations. Figure 1 shows the process flow diagram of the plant model set up for power plant "Kostolac B".

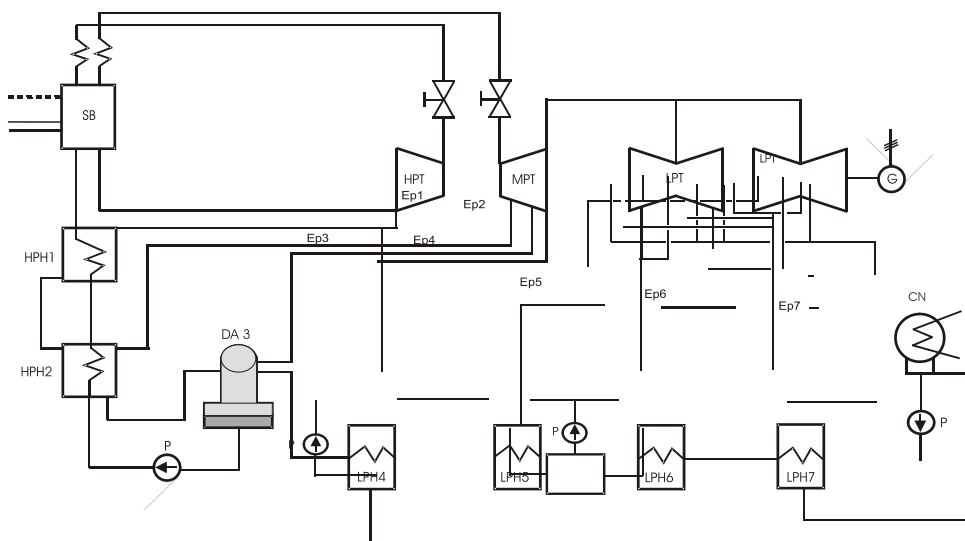


Figure 1. Thermodynamic plant model "Kostolac B"

In this model emphasis has been put on modeling the turbine and the cold end. For the analysis of the impact of the cold end, the boiler performance was considered less important. Therefore, the boiler has not been modeled in detail.

**Table 1. 18 K 348 turbine and condenser properties**

<b>18 K 348 turbine properties</b>	
Nominal output	348.5 MW
Number of revolutions	3000 rev per min
Fresh steam pressure	17.9 MPa
Fresh steam temperature	537 °C
Fresh steam flow rate	277.78 kg/s
Pressure of steam leaving HPT	4.73 MPa
Temperature of steam leaving HPT	338 °C
Pressure of steam in front of MPT	4.24 MPa
Temperature of steam in front of MPT	537 °C
Pressure of steam leaving LPT	0.6 MPa
Temperature of steam leaving LPT	265 °C
Outlet pressure	0.0042 MPa
Number of extraction ports	7
<b>18 K 348 turbine condenser properties</b>	
Technical data	
Heat transfer surface	8468 m <sup>2</sup>
Number of tubes	17 550
Tube diameter x length	24 x 6400 mm
Hotwell volume	31 m <sup>3</sup>
Operating data at 100% turbine load	
Steam flow rate	183.1 kg/s
Pressure	0.044 bar
Cooling water flow rate	13 m <sup>3</sup> /s
Cooling water velocity	1.95 m/s
Cooling water temperature	12 °C

Turbine 18 K 348 of this plant is four-cylindrical, reaction-axial flow with overheating between stages. It consists of the high pressure turbine (HPT), the medium pressure turbine (MPT) and two low pressure turbines (LPT). The steam turbine set is

equipped with seven extraction ports. The plant has four low-pressure feedwater heaters (LPH4, LPH5, LPH6, and LPH7). Heating up the boiler (SB) feed water to the final stage at the input of the boiler is done by two high pressure regenerative heaters (HPH1, HPH2). These devices are supplied with heating steam from the extraction ports (Ep1, Ep2 ...). Removing the non-condensable gases from the steam cycle is being realized by the deaerator (DA3). Exhaust steam from the low pressure turbine is completely condensed in the surface condenser (CN). The technical characteristics for this turbine and the technical data for this condenser are given in the Table 1 [7].

For cooling its condenser, the “Kostolac B” power plant uses cooling water from the Danube. The temperature of this cooling water varies from around 4 °C in winter up to 28 °C in summer. At this moment, flow rate of the cooling water cannot be controlled in a wider range.

The mathematical model of the proposed thermodynamic problem formulation is 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 [8, 9]. This formulation for each component, subsystems and the whole system is presented by following mass balance eq. (1) and energy balance eq. (2) equations:

$$\sum_{j \in \text{in}(i,j)} \dot{m}_j - \sum_{j \in \text{out}(i,j)} \dot{m}_j = 0, \forall i \in I \quad (1)$$

$$\sum_{j \in \text{in}(i,j)} \dot{m}_j h_j - \sum_{j \in \text{out}(i,j)} \dot{m}_j h_j - \dot{W}_i + \dot{Q}_i = 0, \forall i \in I \quad (2)$$

$$\forall i \in \{LPT, MPT, HPT, CN, LPH_i, HPH_i, \dots\}$$

For calculating the heat transfer coefficient Berman’s method was used [10]:

$$K = 3500\beta \left( \frac{1,1W}{\sqrt[4]{D_1}} \right)^x \left[ 1 - \frac{0,42\sqrt{\beta}}{1000} (35 - t_{wl})^2 \right] \Phi_z \Phi_\delta \quad (3)$$

Coefficient  $x$  is calculated as following eq. (4):

$$x = 0.12\beta(1 + 0.15t_{wl}) \quad (4)$$

- $\beta$  – heat transfer surface fouling rate [-],
- $W$  – water velocity in tubes [ $\text{ms}^{-1}$ ],
- $D_1$  – tube inner diameter [m],
- $\Phi_z$  – factor which takes into account number of water passes through the condenser, eq. (5), [-],
- $\Phi_\delta$  – factor which takes into account influence of the condenser steam load, eq. (6.1), eq. (6.2), [-],
- $t_{wl}$  – cooling water inlet temperature,

$$\Phi_z = 1 + \frac{z-2}{10} \left( 1 - \frac{t_{w1}}{35} \right) \quad (5)$$

$z$  – number of water passages through the condenser.

For steam load between nominal value  $d_k^{nom}$  and boundary value  $d_k^{bound}$ :

$$d_k^{bound} = (0,09 - 0,012t_1) d_k^{nom}, \quad \Phi_\delta = 1 \quad (6.1)$$

$$d_k < d_k^{bound} \quad \Phi_\delta = \frac{d_k}{d_k^{bound}} \left( 2 - \frac{d_k}{d_k^{bound}} \right) \quad (6.2)$$

For once-through cooling and pure water, the fouling rate of the heat transfer surface is  $\beta = 0.80-0.85$ . For circulating cooling and chemically treated water  $\beta = 0.75-0.80$ , and for dirty water with mineral and organic sediments forming on the tubes,  $\beta = 0.65-0.75$ .

Cooling water velocity can be written in the following form

$$W = \frac{4m_w}{D_1^2 \pi n_t \rho_w} \quad (7)$$

After this replacement the heat transfer coefficient can be calculated from the following equation:

$$K = 3500\beta \left( \frac{4,4\dot{m}_w}{D_1^{9/4} \pi n_t \rho_w} \right)^{[0,12\beta(1+0,15t_{w1})]} \left[ 1 - \frac{0,42\sqrt{\beta}}{1000} (35 - t_{w1})^2 \right] \left[ 1 + \frac{z-2}{10} \left( 1 - \frac{t_{w1}}{35} \right) \right] \quad (8)$$

Considering the following relations:

– Energy balance formula eq. (9)

$$\dot{Q}_c = \dot{m}_w c_w (t_{w2} - t_{w1}) = \dot{m}_c (h_s - h_c) \quad (9)$$

– Logarithmic mean temperature difference eq.(10)

$$\Delta t_m = \frac{(t_{w1} - t_{w2})}{\ln \frac{(t_c - t_{w2})}{(t_c - t_{w1})}} \quad (10)$$

– Peclet equation eq.(11)

$$\dot{Q}_c = KA\Delta t_m \quad (11)$$

Where:

$\dot{Q}$  – condenser heat transfer rate [kW],

$\dot{m}_w$  – cooling water flow rate [kg s<sup>-1</sup>],

$\dot{m}_c$  – steam flow rate through condenser [kg s<sup>-1</sup>],

$t_c$  – condensing temperature [°C],

$t_{w2}$  – cooling water outlet temperature [°C],

$K$  – heat transfer coefficient [kWm<sup>-2</sup>K<sup>-1</sup>].

The heat transfer rate of the condenser, depending on cooling water temperature and flow rate, condensing temperature (*i. e.* condensing pressure) and steam flow rate through the condenser, can be calculated from eqs. (12, 13) as follows:

$$\dot{Q}_c = \dot{m}_w c_w \left( 1 - e^{-\frac{AK}{\dot{m}_w c_w}} \right) (t_c - t_{w1}) \quad (12)$$

$$\dot{m}_c = \frac{\dot{m}_w c_w \left( 1 - e^{-\frac{AK}{\dot{m}_w c_w}} \right) (t_c - t_{w1})}{h_s - h_c} \quad (13)$$

Plant power output can be calculated from eq. (2).

The specific heat rate is one of the main thermodynamic parameters for the best evaluation of the energy efficiency of the steam power plant. The specific heat rate can be calculated as:

$$q = \frac{Q_{SB}}{\eta_{SB} N} = \frac{B H_i}{\eta_{SB} P} = \frac{\dot{m} (i_1 - i_{fw})}{\eta_{SB} P} \quad (14)$$

Where:

$q$  – specific heat rate [kJkW<sup>-1</sup>s<sup>-1</sup>],

$Q_{SB}$  – steam boiler heat transfer rate [kW],

$P$  – power output [kW],

$B$  – fuel mass flow rate [kgs<sup>-1</sup>],

$H_i$  – lower calorific value of the fuel [kJkg<sup>-1</sup>],

$h_i$  – enthalpy of the steam leaving steam boiler [kJkg<sup>-1</sup>K<sup>-1</sup>],

$h_{fw}$  – feed water enthalpy [kJkg<sup>-1</sup>K<sup>-1</sup>],

$\dot{m}$  – boiler steam load [kgs<sup>-1</sup>],

$\eta_{SB}$  – steam boiler efficiency [-].

Overall energy efficiency of the power plant can be calculated from eq. (14):

$$\eta = \frac{P}{BH_i} \quad (15)$$

Numerical integration was done using Microsoft Excel programming platform and Visual Basics for Applications. The software includes the water/steam thermodynamic properties simulator (settled in the Excel Add-in component), based on IAPWS Industrial Formulation 1997 (IAPWS-IF97). This formulation is proposed by the International Association for the Properties of Water and Steam. The validity field extends over to temperatures between 0-800°C, for pressures up to 100 MPa, [11].

Hereinafter certain of the obtained results are given and discussed.

### **Cooling water parameters influence on the condenser performance**

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. 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). Still, cooling water temperature directly affects condenser performances. Suitable parameter for on-line control is cooling water flow rate, and it can be varied in a wide range, with appropriate circulation pumps. During plant operation the objective is to operate at the optimum cooling water flow rate, which depends on cooling water temperature and power demand [12]. In that manner, cooling water temperature and flow rate are considered variable parameters in the simulation of the plant operating conditions.

#### ***Condensing pressure and condenser heat transfer rate***

With cooling water temperature rise, the mean temperature difference in the condenser decreases, and condenser heat transfer rate for the same condensing pressure will also decrease, as it is shown in figure 2. On this figure, an interesting detail can be seen: at cooling water temperature nearly designed, condenser heat capacity has its maximum, and lower temperature will not lead to significant increase of the heat transfer rate.

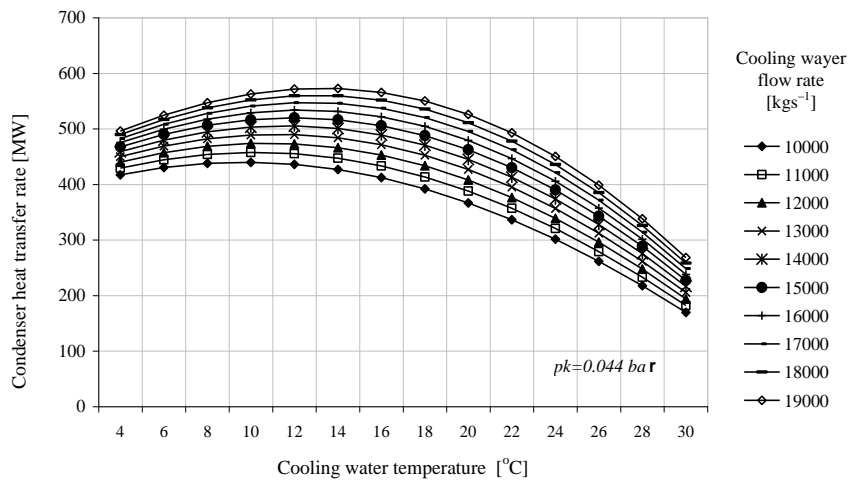
It means that this particular condenser is designed at its maximum heat transfer rate and with increased cooling water temperature it cannot achieve required value. In this way, the question of the valid designed parameters is opened. Increasing of cooling water flow rate will increase the condenser heat transfer rate for the given cooling water temperature.

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 figure 3. It is obvious that with cooling water increasing, pressure in the condenser will also increase.

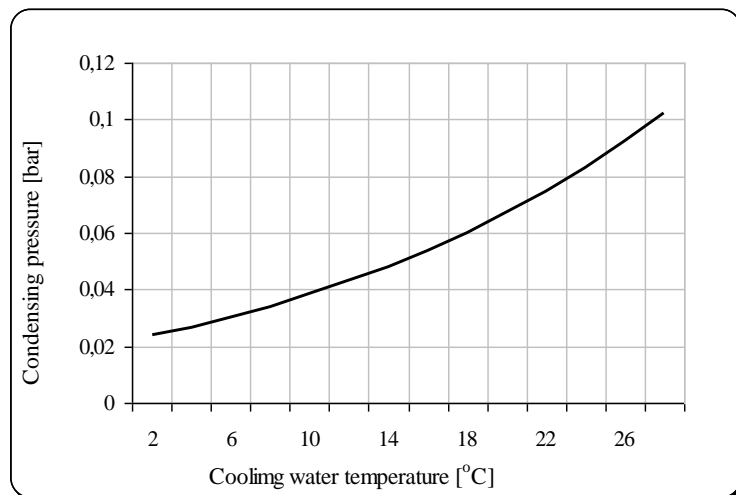


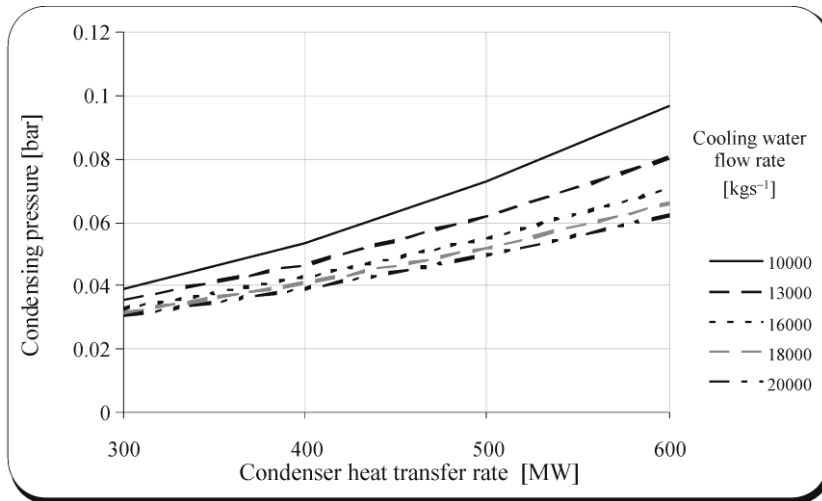
Figure 4 shows the correlation between condenser heat transfer rate and condensing pressure. In this figure, each curve shows the change of condensing pressure at a constant cooling water temperature, for different values of cooling water flow rate. This correlation was used also for model performance verification since the same dependence is given by manufacturer of this condenser, obtained from plant management [13].

**Figure 2.**  
 Condenser heat transfer rate due to cooling water temperature and flow rate change



**Figure 3.** Condensing pressure change due to cooling water temperature change

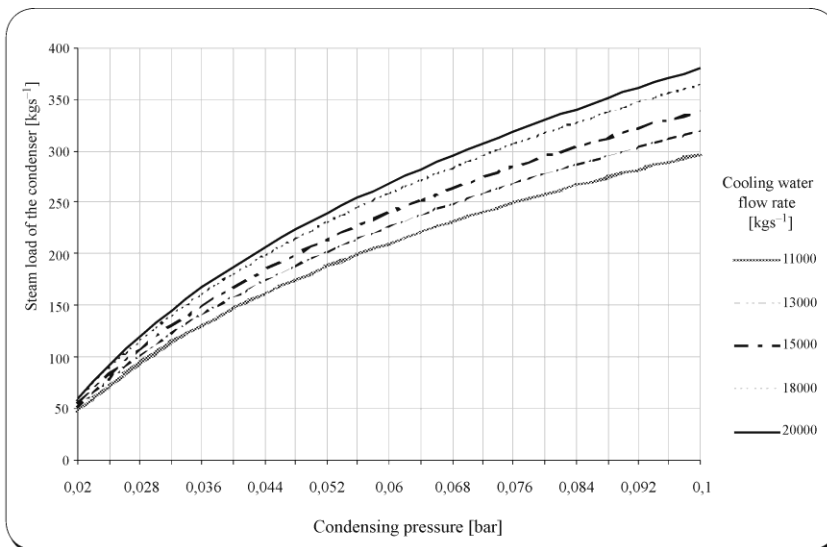




**Figure 4.**  
**Condenser**  
**characteristic**  
**curves**

### ***Steam load of the condenser***

With cooling water temperature increasing, in order to maintain designed heat transfer rate of the condenser, condensing pressure will increase. As this plant is working under turbine-follow mode, 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, which will be discussed in section 4 of this paper. Figure 5 shows the steam load of the condenser dependence on condensing pressure, at a constant cooling water temperature as for different values of cooling water flow rate.



**Figure 5.** Steam  
load of the  
condenser for  
cooling water  
temperature  
12 °C

## Condenser operating conditions influence on the plant performances

As it can be seen in figure 5, with increasing of condensing pressure and flow rate of the cooling water, steam flow through the condenser, and thus through the low-pressure turbine is increasing. This will increase net power output. This increasing of the net power output, however, is correlated to the increasing of the heat rate.

### Specific heat rate

Specific heat rate change due to condensing pressure change is shown in figure 6. Those results are identical to experimental results for similar plant given in literature [14]. With decreasing of the pressure in the condenser, specific heat rate decreases. With pressure decreasing to below the point where the exhaust annulus becomes choked, excessive condensate sub-cooling will result, tending to reduce the improvement in heat rate resulting from the lower condensing pressure. Condensate sub-cooling is defined as the saturation temperature corresponding to the pressure in the condenser minus the condensate temperature in the hot well [2]. This is evident in figure 6 especially for 337 MW curve. Condensate sub-cooling is undesirable since more heat has been removed from the cycle without generating any additional power, but still has to be replaced by adding fuel.

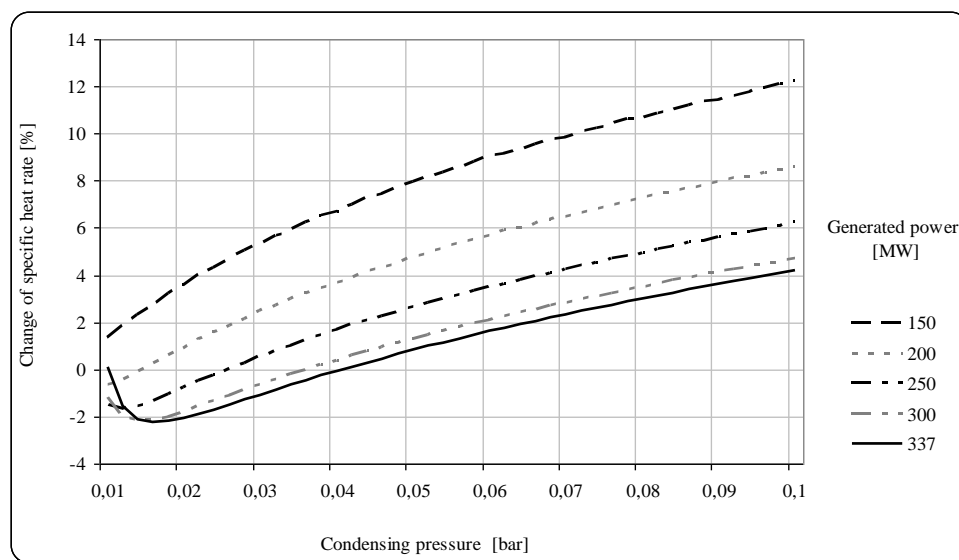


Figure 6. Specific heat rate change due to condensing pressure and generated power change

Having the condensing pressure change due to the cooling water temperature change (figure 3) and dependence given in figure 6, the change of the specific heat rate dependence on the cooling water temperature is obtained, and it is shown in figure 7. In order to present this dependence more clearly, it was chosen to keep the plant power production constant. Power consumption of the cooling water pumps was not considered.

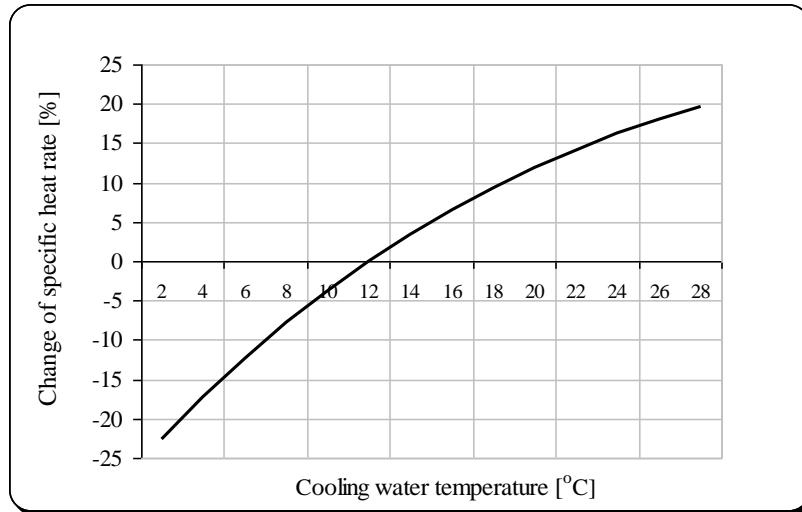


Figure 7. Specific heat rate due to cooling water temperature change

### ***Energy efficiency***

Using the known assumption, from the literature [14], that with increasing pressure in the condenser of 1 kPa, efficiency decreases to 1.0-1.5% and considering that in this particular case the reduction is 1.2%, dependence of the energy efficiency (generated power divided by an amount of energy consumed) in the function of the cooling water temperature rise is obtained and is shown in figure 8.

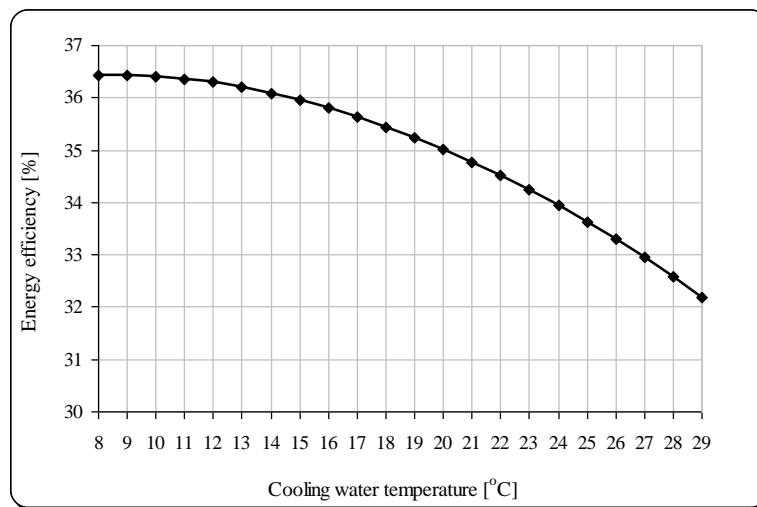


Figure 8. Energy efficiency of the plant due to cooling water temperature change

Cooling water temperature rise causes the reduction of energy efficiency and power of steam power plants (in Germany the production of electricity in summer months due to the increase of cooling water temperature is decreased by 18%). The problem is more severely set for plants with closed-cycle cooling system in comparison to the plant with an open cycle cooling system. Dry bulb temperature, the wet bulb temperature, the atmospheric pressure, flow rate of the circulating water, the characteristic of cooling tower fill, the resistance coefficient of the parts in a tower, and the working condition of a water distribution system and so on, can affect the outlet water temperature of the cooling tower [15]. It means that cooling water temperature is changing in a much wider range in one day in comparison to once-through cooling system.

## Conclusions

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. The pressure in the surface steam condenser will depend on condenser design, the 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 the combined status of these operating conditions will determine the relationship between heat rate and power output.

In this paper, influence of cooling water temperature and flow rate is considered for power plant "Kostolac B" with once-through cooling system. The thermodynamic model of the plant is described, together with the mathematical model. Numerical integration was done by using the Microsoft Excel programming platform and Visual Basics for Applications, including the water/steam thermodynamic properties simulator based on IAPWS-IF97. As the result of calculation, set of different dependencies is obtained and described herein: the condenser heat transfer rate and pressure in the condenser are given for variable cooling water temperature and flow rate, the specific heat rate change due to the change of condensing pressure, and the specific heat rate change due to the cooling water temperature change. Finally, the energy efficiency for the reference plant is given as the function of the change in condensing pressure.

Having those dependences, it is clearly that the increasing in cooling water temperature, inevitable in the summer, will lead to the decrease of energy efficiency of the power plant for the given power production. The increasing of cooling water flow rate to maintain the same heat transfer rate at higher vacuum in the condenser, and thus, to increase energy efficiency of the plant, is a good way in optimizing plant operation.

## Nomenclature

$A$	– heat transfer surface, [m <sup>2</sup> ]	$h_1$	– enthalpy of the steam leaving steam boiler [kJkg <sup>-1</sup> K <sup>-1</sup> ]
$B$	– fuel mass flow rate [kgs <sup>-1</sup> ]	$K$	– heat transfer coefficient [kWm <sup>-2</sup> K <sup>-1</sup> ]
$c_w$	– water specific heat, [kJkg <sup>-1</sup> K <sup>-1</sup> ]	$\dot{m}_s$	– steam flow rate through condenser [kgs <sup>-1</sup> ]
$D_1$	– tube inner diameter [m]	$\dot{m}_w$	– cooling water flow rate [kgs <sup>-1</sup> ]
$H_i$	– lower calorific value of the fuel [kJkg <sup>-1</sup> ]		
$h_{fw}$	– feed water enthalpy [kJkg <sup>-1</sup> K <sup>-1</sup> ]		
$h_c, h_s$	– steam and condensate enthalpy [kJkg <sup>-1</sup> K <sup>-1</sup> ]		

$n_t$	– number of condenser tubes	<i>Greek letters</i>	
$P$	– power output [kW]	$\beta$	– heat transfer surface fouling rate
$Q$	– specific heat rate [ $\text{kJkW}^{-1}\text{s}^{-1}$ ]	$\eta_{SB}$	– steam boiler efficiency [–]
$\dot{Q}_c$	– condenser heat transfer rate [kW]	$\eta$	– overall energy efficiency of the power plant [–]
$\dot{Q}_{SB}$	– steam boiler heat transfer rate [kW]	$\Phi_d$	– factor which takes into account influence of the condenser steam load [–]
$t_c$	– condensing temperature [°C]	$\Phi_z$	– factor which takes into account number of water passes through the condenser [–]
$t_{w1}$	– cooling water inlet temperature [°C]	$\rho_w$	– water density [ $\text{kgm}^{-3}$ ]
$t_{w2}$	– cooling water outlet temperature [°C]		
$W$	– water velocity in tubes [ $\text{ms}^{-1}$ ]		
$z$	– number of water passages through the condenser		

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