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Thermal-flow Study of Closed Cooling System with Cooling Towers

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1. Introduction

Although power plants are mainly seen as producers of electricity and heat, they are also significant customers of fresh water which is widely used in many technological processes including e.g.: water-steam cycle in power boiler, cooling systems cooperating with condensers or cooling installations of boiler's auxiliary equipment. The daily consumption of fresh water by large power units is usually accounted, at least, at the level of a several thousand m^3.

Large demand of fresh water by power energy sector is always connected with the issues of its proper management due to the unquestionable interference of the industry with local water ecosystems. It is worth mentioning that the water demands of power plant cooling systems may strain natural water reservoirs and make power production vulnerable, as well as natural environment, to water scarcity (Loew 2016). Moreover, meaningful increase of production of wastewater and its influence on environment, recently reported by e.g. Blog (Blog 2016), indicates on the need for new restrictive regulations for water environment protection like e.g. EU directive on industrial emissions from 2010. The attempts to the modelling of changes in chemical compounds in wastewater are presented in (Skoczko 2017, Regucki 2017).

Beside environmental aspects, proper management of fresh water in cooling systems has also significant influence on the overall efficiency of electricity and heat production in coal-fired power plants, which is gradual decrease with ages (Campbell 2013). Over the years, the working parameters of cooling installations are subjected to the changes due to e.g. reconstruction of these systems, unfavorable deterioration of the circulating pumps characteristics or deposition of mineral deposits on heat-exchange surfaces or pipelines. In order to improve the overall efficiency of currently in use power units, Ryabchikov suggested a number of specific actions, among which the retrofit of cooling water installations is one of the most important (Ryabchikov 2012). Nichols estimated that the modernization of cooling system performance in American power plants could lead to improving the overall efficiency of the unit by about 0.2-1.0% (Nichols 2008). Similar estimations, made for APEC countries, indicate that the improvement of feed water heaters and condensers could update the overall power unit efficiency by a value of approximately 0.8% (Boncimino 2005).

Taking into account the abovementioned facts, closed cooling systems cooperated with cooling towers have become, in natural way, the object of research and analysis of many scientists. The interests split into two main parts focusing on the numerical modeling of the cooling systems and research of influence of ambient conditions on working parameters of cooling towers. An example of numerical investigation of the cooling system as a whole is presented by Wróblewski in (Wróblewski 2013) where the comparison of two different variants of cooling system for 900 MW ultra-supercritical power unit is done. His analysis focuses on the impact of the thermal-flow properties of the cooling system on the lowest possible pressure achieved inside the condensers. Numerical modeling of the thermal-flow conditions inside the cooling towers is presented in (Blain 2016, Opris 2017). Blain in (Blain 2016) presents the results of CFD modeling of cooling tower basing on the Poppe and Merkel equations and splitting the interior of the cooling tower into three parts: fill, spray and rain zones. The model was validated on the small scale fill (7mx7m counter-flow section) and next on the real object achieving good agreement with measurements. Opris in (Opris 2017) presents other methodology of design the cooling towers using block-modules which takes into account division of the cooling tower into three main parts: spray and fill zone, rain zone and natural draft zone.

Another approach focuses on the influence of the ambient conditions, strength and direction of wind or air-water flow ratio on heat and mass processes inside the cooling towers. Li in (Li 2018) shows that cold inflow at the top of the cooling tower could have unfavorable influence on the natural drift blocking heat rejection. Her experiments, done on 20 m test cooling tower, indicate that the undesired turbulence inside the cooling tower could cause the increase of outlet cooling water temperature even up to 3°C. Similar consideration presents Weiliang in (Weiliang 2016) analyzing the influence of wind on the operational conditions of cooling tower. The vortices, created by the wind in outer and inner space of cooling tower, affect air flow decreasing the efficiency of heat and mass processes. He considers the implementation of windbreaks to prevent the deterioration of the cooling performance inside the natural draft dry cooling tower. The influence of air water flow ratio on the operating performance is studied by Liu in (Liu 2017). She shows that various meteorological parameters have a significant impact on the large variation of air water flow ratio what causes unstable operation of cooling tower. Martin in (Martín 2017) evaluates the influence of weather conditions and cooling tower localization on its operation parameters. He shows that weather conditions have an significant influence on non steady production of energy by power units. Among others, he confirms that the extreme temperatures in summer time reduce the electricity production capacity due to limitations in the heat transfer capacity inside the cooling tower.

The paper presents the thermal-flow study of a closed cooling system with special emphasis on the operating parameters of the cooling tower. The uniqueness of the analysis lays in measurements done inside the working cooling tower to identify the heat and mass transfer processes across its radius. Next, the analysis of a cooling water temperature drop, in the cooling system cooperated with a set of cooling towers, is considered. As an example of these studies there are calculated the optimal cooling water flow rates through the cooling towers to achieve the highest possible cooling water temperature drop in the installation.

2. Study of thermal-flow processes inside cooling tower

Cooling systems are very important parts of the power units and, as it was mentioned previously, they have a significant influence on the overall efficiency of electricity and heat production. The scheme of a closed cooling water system cooperated with natural draft cooling tower is presented in Fig. 1.

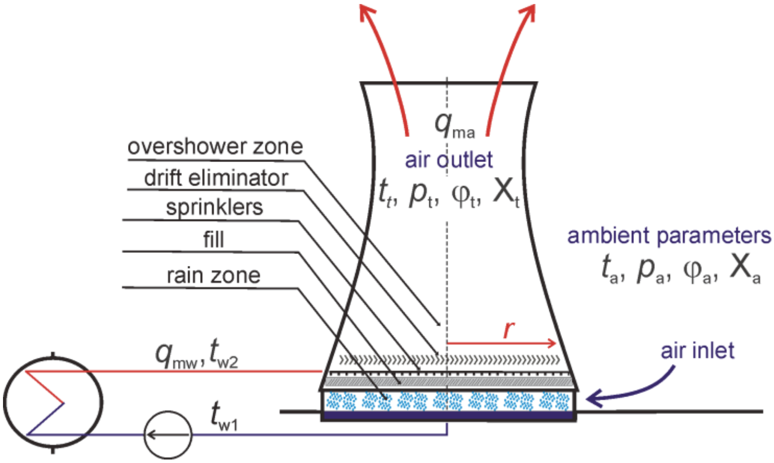

Fig. 1. Scheme of closed cooling water system with natural draft cooling tower. The symbols used in the drawing: t_a, t_t – temperature of the air; p_a, p_t – pressure of the air; φ_a, φ_t – relative humidity of the air; X_a, X_t – absolute humidity of the air at inlet and outlet respectively; q_{ma}, q_{mw} – mass flow rate of air and cooling water respectively; t_{w2}, t_{w1} – temperature of cooling water at inlet and outlet respectively

Rys. 1. Schemat zamkniętego układu chłodzenia z chłodnią kominową. Oznaczenia przestawione na rysunku: t_a, t_t – temperatura powietrza; p_a, p_t – ciśnienie powietrza; φ_a, φ_t – wilgotność względna powietrza; X_a, X_t – wilgotność bezwzględna powietrza odpowiednio na wlocie i wylocie z chłodni; q_{ma}, q_{mw} – strumień masy powietrza i wody chłodzącej; t_{w2}, t_{w1} – temperatura wody chłodzącej na wlocie i wylocie z chłodni kominowej

Circulating water is heated in a condenser by condensing steam from the low pressure turbine, and it is then cooled down in a cooling tower. The main mechanism of water temperature decrease inside the cooling tower is by its partial evaporation in contact with counter flowing air (Berman 1961). This phenomenon causes a continuous decrease in the mass of the circulating water in the installation which is replenished from natural resources. In the natural draft cooling tower the heat and mass transfer processes take place in two ways (notations according to Fig. 1):

– an increase in the temperature of the flowing air $\Delta t_t = t_t - t_a$
– an increase of the humidification of the flowing air $\Delta X = X_t - X_a$ (what results in changes of relative humidity of the air $\Delta\varphi_t = \varphi_t - \varphi_a$),

where:
t_a, t_t – air temperature,
X_a, X_t – absolute humidity of air,
φ_a, φ_t – relative humidity of air at inlet and outlet of the cooling tower, respectively.

The analysis shows that the highest temperatures of the returning cooling water t_{w1} are in summer time when the ambient air temperature t_a is high (in Poland ambient temperatures varies then between 20-30°C). Worse heat exchange conditions in the closed cooling system causes seasonal deterioration of the overall power unit efficiency (due to the fact of higher achievable pressure inside steam condensers), what leads to the increase of fossil combustion and, therefore, higher emission of GHG and other pollutants (Martín 2017, Campbell 2013). Table 1 presents an example of measurements done in June inside the natural draft cooling tower along the radius r (where $r = 0$ is at the center line of cooling tower – see Fig. 1). The investigated cooling tower has a height of 105 m and is made in the form of a hyperboloid reinforced concrete structure with natural draft. Its hydraulic load varies in the range of q_{vw} = 20000-35000 m^3/h and thermal output is up to 300 MW. It is worth mentioning that the data presented in Table 1 for each position (r) are calculated from hundred of measurements collected in 30-minute time interval.

The measurement was done in the overshower zone above drift eliminators, where the radius of the cooling tower is R = 35 m. Pressure inside the cooling tower was uniformly distributed along the radius and equal p_t = 977.7 hPa, when the ambient pressure was p_a = 980.0 hPa. Additionally the profile of air velocity was measured using Testo 416 anemometer and the mean velocity u_t inside the cooling tower was 1.06 m/s.

Table 1. An example of measurements done inside the natural draft cooling tower along the radius r ($r = 0$ is at the center line of cooling tower – Fig. 1)

Tabela 1. Przykład pomiarów wykonanych wewnątrz chłodni kominowej wzdłuż jej promienia r ($r = 0$ jest na osi symetrii chłodni – rysunek 1)

Time, June hh:mm	radius r, m	**air**, ambient parameters		**cooling tower**, air at outlet		**cooling tower**, parameters of cooling water		
		t_a, °C	φ_a,%	t_t, °C	φ_t, %	t_{w2}, °C	t_{w1}, °C	q_{vw}, m^3/h
15:55	2	26.4	59.0	35.3	99.5	38.5	30.7	34988
15:25	14	26.5	55.0	35.5	98.3	38.5	30.8	35585
14:55	20	27.4	53.0	35.4	97.6	38.5	30.7	34541
14:25	26	27.7	53.0	34.3	99.1	38.4	30.8	34435
13:55	29	27.9	52.0	36.0	97.3	38.4	30.8	34493
13:20	32	27.4	52.0	35.6	99.5	37.9	30.4	34628
mean value ± standard dev.		**27.2**±0.6	**54.0**±2.7	**35.4**±0.6	**98.6**±1.0	**38.4**±0.2	**30.7**±0.1	**34778**±441

The presented measurement data are the starting point for the analysis of mass and heat exchange processes inside a cooling tower. The authors' intention is to quantitatively analyze the data and to assess which processes play a leading role in obtaining the lowest possible outlet temperature of cooling water (especially in the summer, when the efficiency of blocks is the lowest one).

2.1. Heat and mass transfer processes inside a cooling tower

Collected data allows precisely to follow the heat exchange processes inside the cooling tower if one assumes steady thermal-flow conditions (Baker 1961):

$$c_{\mathrm{w}} \cdot \rho_{\mathrm{w}} \cdot q_{\mathrm{vw}} \cdot (t_{\mathrm{w2}} - t_{\mathrm{w1}}) = q_{\mathrm{ma}} \cdot \big(h_2(t_{\mathrm{t}}, \varphi_{\mathrm{t}}) - h_1(t_{\mathrm{a}}, \varphi_{\mathrm{a}})\big) \quad (1)$$

where:
$c_{\mathrm{w}} = 4189.9$ J/kgK – specific heat of water,
$\rho_{\mathrm{w}} = 994.23$ kg/m^3 – density of water at mean temperature equals 34.5°C,
q_{vw} – volumetric flow rate of circulating water,
q_{ma} – mass flow rate of air flowing inside the cooling tower,
t_{w2}, t_{w1} – inlet and outlet temperature of circulating water, respectively,
h_2, h_1 – enthalpy of moist air leaving and entering cooling tower, respectively.

Basing on values from Table 1, left hand side of (1) gives the heat flux released into the flowing air by cooling water (if one neglects the loss of water mass due to its partial evaporation process) $Q_{\mathrm{w}} = 308.5$ MW. On the other side, the value of heat flux taken by the air is equal $Q_{\mathrm{a}} = 315.4$ MW, where q_{ma} is calculated for air parameters inside the cooling tower: $u_{\mathrm{t}} = 1.06$ m/s and $\rho_{\mathrm{t}}(t_{\mathrm{t}}, \varphi_{\mathrm{t}}, p_{\mathrm{t}}) = 1.08$ kg/m^3.

The discrepancy between obtained values of heat flaxes $e = (Q_{\mathrm{w}}-Q_{\mathrm{a}})/Q_{\mathrm{w}}$ is at the level of 2.2% what is a very good result. This difference could be corrected if one takes into account a correction connected with the loss of circulating water Δq_{vw} due to its partial evaporation:

$$Q_{\mathrm{w}} = c_{\mathrm{w}} \cdot \rho_{\mathrm{w}} \cdot \big(q_{\mathrm{vw}} \cdot t_{\mathrm{w2}} - (q_{\mathrm{vw}} - \Delta q_{\mathrm{vw}}) \cdot t_{\mathrm{w1}}\big) \quad (2)$$

Amount of evaporated water is calculated from the difference of absolute humidity of air entering and leaving the cooling tower X_{a} and X_{t} respectively:

$$\Delta q_{\text{mw}} = q_{\text{ma}} \cdot \left(X_{\text{t}}(t_{\text{t}}, \varphi_{\text{t}}) - X_{\text{a}}(t_{\text{a}}, \varphi_{\text{a}}) \right) \tag{3}$$

For analyzed data, this value is Δq_{mw} = 394.0 m^3/h. This is approximately 1% of total mass flow rate of circulating water and is in very good agreement with literature (Berman 1961). Correction of heat flux released by cooling water due to its partial evaporation is ΔQ_w = 14.0 MW. Finally, corrected value of total heat flux transferred from the cooling water is now equal Q_w = (308.5+14.0) MW = 322.5 MW but the discrepancy e is still at the level of 2%. On the other hand, the calculation of relative humidity φ_t inside the cooling tower bases on the measurement of dry- and wet-bulb temperatures of flowing air (using two A class Pt100 thermometers). If one assumes that the leaving air has φ_t = 100% then the total heat flux transferred to the air will be Q_a = 321.6 MW and final discrepancy between corrected heat fluxes Q_w and Q_a is less than 1%.

Summarizing the above calculations, it seems that to balance correctly heat and mass transfer processes inside the cooling tower it is necessary to take into account the loss of circulating water as well as to assume that the air above the overshower zone is fully saturated and is at its dewpoint (what means that the relative humidity of leaving air φ_t equals 100%). It is worth mentioning that the source of potential discrepancies in analyzed results may be, on the one hand, the accuracy of the measurements made, and on the other, the use of an 0-dimensional heat and mass exchange model.

It is interesting how the total heat flux Q_a is divided between increase of temperature Δt_t and changes of humidification (expressed by changes of relative humidity $\Delta\varphi_t$) of the flowing air:

$$\mathrm{Q_a} = \mathrm{Q_{a\Delta t}} + \mathrm{Q_{a\Delta\varphi}}, \tag{4a}$$

$$\mathrm{Q_{a\Delta t}} = q_{\text{ma}} \cdot \left(h(t_{\text{t}}, X_{\text{a}}) - h_1(t_{\text{a}}, X_{\text{a}}) \right) \tag{4b}$$

$$\mathrm{Q_{a\Delta\varphi}} = q_{\text{ma}} \cdot \left(h_2(t_{\text{t}}, X_{\text{t}}) - h(t_{\text{t}}, X_{\text{a}}) \right) \tag{4c}$$

For analyzed data, $Q_{a\Delta t}$ = 37.9 MW and $Q_{a\Delta\varphi}$ = 277.5 MW (for the case of φ_t = 100%: $Q_{a\Delta t}$ = 37.9 MW and $Q_{a\Delta\varphi}$ = 283.7 MW, respectively). It means that the main heat exchange process between the cooling water and air is through partial evaporation of circulating water (approximately 88%) and only 12% of total heat flux Q_a is utilized to warm up the flowing air.

Above results show how significant are ambient conditions for the proper working parameters of the cooling system. This observation is especially important because power unit operation is characterized by non-steady production of energy that strongly depends on current weather conditions.

2.2. Thermal–flow characteristic of cooling tower

The heat flux balance (1) is a starting point to much more detailed analysis of the heat and mass transfer processes inside the cooling tower which bases on enthalpy potential as the driving force (Baker 1961). The heat transfers to air from a tin film of cooling water flowing down along the fill is considered in control volume dV:

$$dQ = \alpha(t_w - t)dV + h' dG_{mw} \tag{5}$$

where: α – volume heat transfer coefficient; t_w, t – local cooling water and air temperature, respectively; h' – enthalpy of water evaporation at local bulk water temperature; dG_{mw} – mass flow rate of evaporated water, in the control volume dV. Amount of evaporated water depends on the mass transfer coefficient between the water film and air β and difference of local absolute humidities on the interface water-air in control volume dV:

$$dG_{mw} = \beta(X'_w - X'_a)dV \tag{6}$$

The equations (5) and (6) describe the amount of heat transferred from the cooling water to the counter-flowing air in control volume dV of the fill. In this approach dV is equal $S{\cdot}dx$ where S is a surface of the cross-section of the cooling tower (perpendicular to the direction of air flow) and dx is a small section of the height of the fill. Due to the complexity of the arrangement of the fill inside the cooling tower there is impossible to consider the heat and mass transfer processes separately for individual components of the fill but rather simplified approach is applied. This is the reason why heat and mass balance equations (5) and (6) base on overall heat and mass transfer: α, β respectively.

Basing on (5), (6) and (1), Berman in (Berman 1961) shows that the total heat transferred to the flowing air is described by equation:

$$Q = \beta \cdot \Delta h_{\mathrm{m}} \cdot V = G_{\mathrm{ma}} \cdot (h_t - h_a) \tag{7}$$

where: V – is volume of fill; G_{ma} – mass of flowing air; Δh_{m} – mean enthalpy increase of flowing air expressed by formula:

$$\Delta h_{\mathrm{m}} = \frac{(t_{w2} - t_{w1})}{\int_{t_{w1}}^{t_{w2}} \frac{dt}{(h' - h)}} \tag{8}$$

where: h' is enthalpy of water evaporation at local bulk water temperature and h – local enthalpy of moist air flowing through the fill (Baker 1961).

On the other side, the heat released from water is calculated from (2):

$$Q = \frac{1}{K} \cdot c_{\mathrm{w}} \cdot G_{\mathrm{mw}} (t_{\mathrm{w2}} - t_{\mathrm{w1}}) \tag{9}$$

where: K – correction factor due to partial evaporation of cooling water, ($K \cong 0.947$). Combining together (7) and (9) one can obtain the formula for the temperature of cooling water at outlet of cooling tower, t_{w1} (Berman 1961):

$$t_{\mathrm{w1}} = t_{\mathrm{w2}} - \frac{K \cdot V \cdot \beta \cdot \Delta h_{\mathrm{m}}}{c_{\mathrm{w}} \cdot G_{\mathrm{mw}}} \tag{10}$$

Equation (10) is a non-linear one, because value of Δh_{m} depends, among other, on t_{w1}. Values of V and β are calculated individually for particular cooling tower because they depend on its geometry and current ratio of air and water mass flow rates $q_{\mathrm{ma}}/q_{\mathrm{mw}}$.

Basing on the measurement data from Table 1, there was possible to calculate the characteristic of the cooling tower showed at Fig. 2.

Presented curve depends on the current inlet parameters of air (t_{a}, p_{a}, φ_{a}) and cooling water (t_{w2}, q_{mw}). It is worth noting that value of $\Delta t_{\mathrm{w}}=(t_{\mathrm{w2}}-t_{\mathrm{w1}})$ increases (what means that t_{w1} decreases) if the volumetric flow rate of cooling water decreases. The difference between the highest and lowest value of Δt_{w} is about 3.3°C and shows that the proper selection of the hydraulic load of cooling tower could increase its cooling properties.

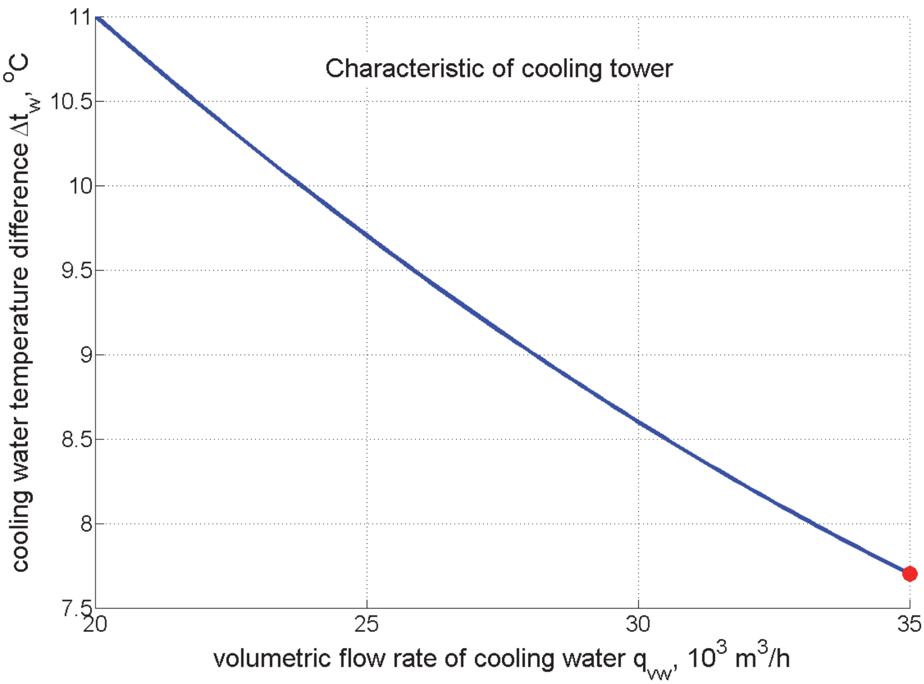

Fig. 2. Characteristic of natural draft cooling tower obtained for measurement data from Table 1. Red circle indicates actual working point of cooling tower
Rys. 2. Charakterystyka chłodni kominowej otrzymana dla danych pomiarowych z Tabeli 1. Czerwonym kołem oznaczono aktualny punkt pracy chłodni kominowej

3. Optimization of cooling process in closed cooling system

The idea of closed cooling system bases on the cooling towers, which are usually arranged in a group of few units and cooperate with several condensers of power units by common hydraulic installation. Such arrangement allows to model the flow rates to individual cooling towers in order to optimize the cooling process (Regucki 2018).

As an example of such approach one can consider the situation when two cooling towers, with different thermal characteristics show at Fig. 3, have to cool down fixed volumetric flow rate of water q_{vt}.

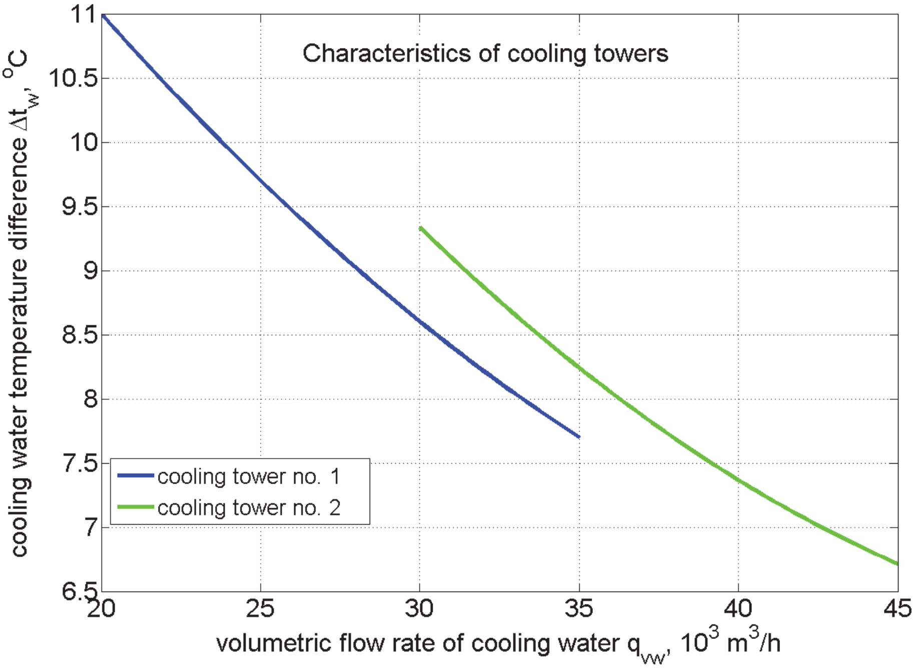

Fig. 3. Example of characteristics of two natural draft cooling towers
Rys. 3. Przykładowe charakterystyki dwóch chłodni kominowych

The total hydraulic load of these two cooling towers is between 50 and 80 thousand m^3/h. Let assume that one would like to cool down fixed flow rate of water q_{vt} = 60,000 m^3/h. The optimal distribution of water between these two cooling towers could be achieved finding the maximum possible total cooling water temperature drop Δt_{w}:

$$\Delta t_{\mathrm{w}} = (\Delta t_{\mathrm{w1}}(q_{\mathrm{vw1}}) \cdot q_{\mathrm{vw1}} + \Delta t_{\mathrm{w2}}(q_{\mathrm{vw2}}) \cdot q_{\mathrm{vw2}})/q_{\mathrm{vt}} \qquad (11)$$

with constraint: $q_{\mathrm{vt}} = q_{\mathrm{vw1}} + q_{\mathrm{vw2}} = 60{,}000$; where q_{vw1}, q_{vw2} – volumetric flow rates through first and second cooling tower, respectively. Solution of this non-linear problem is showed at Fig. 4.

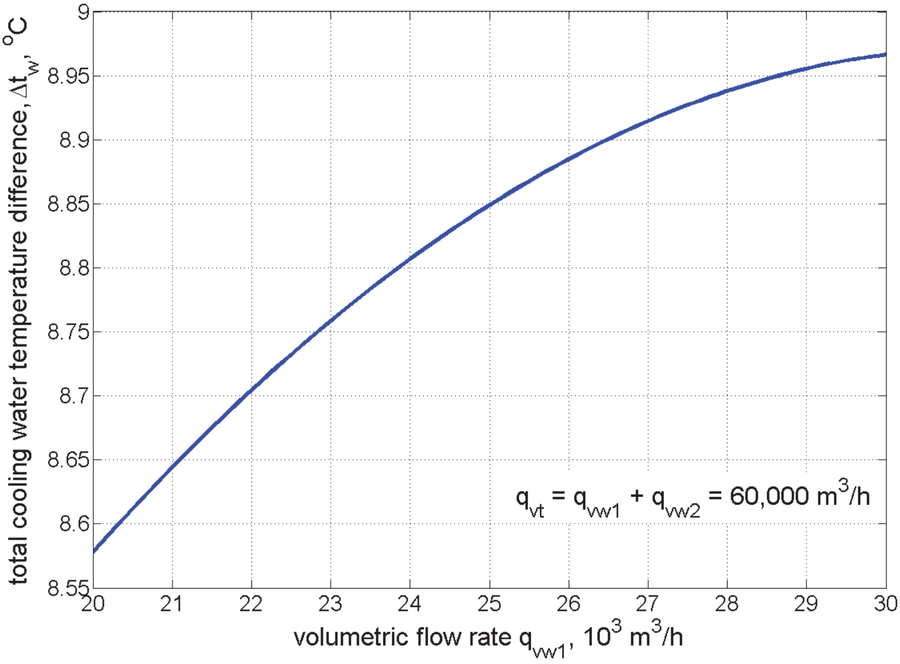

Fig. 4. Total cooling water temperature difference Δt_w calculated from (11) for characteristics presented at Fig. 3

Rys. 4. Całkowity spadek temperatury wody Δt_w obliczony z formuły (11) dla charakterystyk przedstawionych na rysunku 3

The value of Δt_w changes between 8.58 and 8.97°C. Presented calculations show that its maximum value is achieved for q_{vw1} equals 30000 m^3/h (and q_{vw2} = 30000 m^3/h). It is worth mentioning that this division of flow rates is valid for analyzed cooling water and weather (air) conditions. If the ambient temperature and relative humidity change then the characteristics will change as well and the optimal total cooling water temperature drop Δt_w could appear for other values of q_{vw1} and q_{vw2}. The problem of optimizing cooling water redistribution in a closed cooling system is a problem that should be referred to the currently analyzed installation. The example presented above was supposed to draw attention to the fact that the proper water redistribution in the cooling system has a direct impact on the obtained mean cooling water temperature in the installation t_{w1}. The presented optimization method can be applied to a system consisting of a larger number of cooling towers, if only their characteristics can be determined.

4. Conclusions

The efficient optimization process always bases on the scientific background involving mathematical modeling and numerical approach which are more and more widely used in power engineering. Due to the huge daily amount of utilized fresh water, closed cooling systems are an important part of power plants and its proper operation could significantly reduce the usage cost as well as increase the overall efficiency of power unit. Presented calculations show that, despite the simple construction, cooling towers are still subjected to research and modeling. Thermal-flow measurements across radius of cooling tower allows to identify the heat and mas exchange mechanisms between cooling water and counter flowing air. After analyzing the data, it was indicated that the measurement of relative humidity inside the cooling tower is not necessary. The calculations show that the relative humidity the inside the cooling tower can be assumed to be 100%. In addition, the presented analysis shows that the decisive parameter affecting the process of water cooling is the relative humidity of the air sucked into the tower. These observations can be used as a starting point for attempts to modernize cooling towers. Data analysis can be used to indicate representative places of measurement of thermodynamic parameters inside a cooling tower in order to evaluate its operating parameters. The presented optimization shows that the proper regulation of the flow rates between two cooling towers can increase the total cooling water temperature difference about 0.4°C. It is very important result if one pays attention to the fact that improvement of Δt_w by 1°C has a direct impact on the increase of overall efficiency of power unit by approximately 0.5%.

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Badania cieplno-przepływowe zamkniętego układu chłodzenia z chłodniami kominowymi

Streszczenie

W pracy przedstawiono cieplno-przepływowe badania zamkniętego układu chłodzenia ze szczególnym uwzględnieniem parametrów pracy chłodni kominowych. Unikalność analizy polega na pomiarze wykonanym wewnątrz pracującej chłodni kominowej w celu identyfikacji procesów wymiany ciepła i masy wzdłuż jej promienia. Następnie przeanalizowano stopień schłodzenia wody cyrkulującej w układzie chłodzenia współpracującym z zespołem chłodni kominowych. Jako przykładowy wynik tych badań zaproponowano dobór optymalnych strumieni przepływu wody chłodzącej w układzie dwóch chłodni kominowych, pozwalający uzyskać możliwie najwyższy spadek temperatury (stopień schłodzenia) cyrkulującej wody.

Abstract

The paper presents the thermal-flow study of closed cooling system with special emphasis on the working parameters of the cooling tower. The uniqueness of the analysis lays in measurement done inside the working cooling tower to identify thermal-flow processes across its radius. Next, the analysis of a cooling water temperature drop, in the cooling system cooperated with a set of cooling towers, is considered. As an example of these studies there are proposed the optimal cooling water flow rates between two cooling towers to achieve the highest possible water temperature difference in cooling system.

Słowa kluczowe:
elektrownia, zamknięty układ chłodzenia,
przeciwprądowa mokra chłodnia kominowa

Keywords:
power plant, closed cooling system, counter flow wet cooling tower