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Simulating power efficiency of heat transfer agent cooling recirculation systems at power plants

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Abstract

The article provides results of simulating processes of aerodynamic and heat-mass exchange interaction of air and water flows in evaporative cooling towers. It is found that use of the systems regulating direction and intensity of air flows cooling recirculating water of power plant turbine capacitors can be used commercially in order to enhance thermal efficiency and rise of electric power generation.

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Keywords: power efficiency, electric power engineering, evaporative cooling towers, effectiveness ratio, simulating;

1. Introduction

The evaporative cooling towers are widely used for heat extraction in the circulating water supply systems at modern power plants running both on fossil fuel and on renewable power sources. Main structural elements of the cooling tower are presented schematically in Fig. 1.

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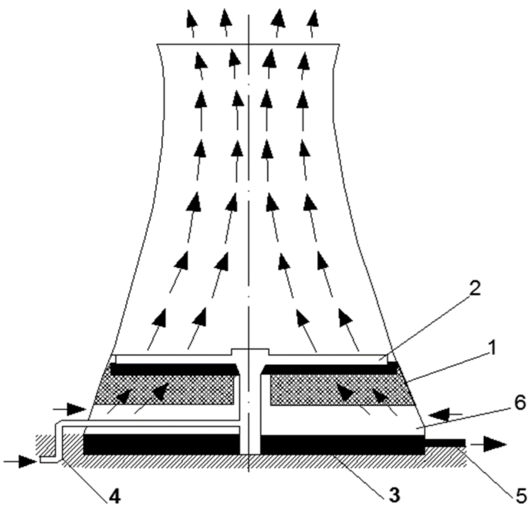
It is worth to note that the cooling towers currently operative are often far from modern requirements of energy efficient production of electric power. According to recent statistic data, in average among all power plant in Russia, the limitation of generated power related to lack of their capacity caused by reduced cooling efficiency of the towers, reaches the values of up to 2.5 % of total volume of generated power [4]. Sufficient reduced production of electricity is caused first of all because of climate changes which led to noticeable increase of duration and intensity of high Summer temperatures of the cooling air, sufficiently reducing depth of circulating cooling water which returns into turbine capacitors [1, 7, 9]. Deterioration of thermal efficiency characteristics of evaporating cooling towers in certain degree is related to ignoring influence of the structure of inner aerodynamics of air flows over thermal-and-mass exchange processes [2, 3, 5, 6, 8]. The article reviews the issues of efficient application of the inner air flows regulating in the cooling towers at power plants of alternative power engineering with use of mathematic model of water cooling process when evaporating in contact heat-and-mass exchange.

The technology of air control in flows inside counterflow cooling towers with natural draught presumes functioning of rotating units at the input windows [10, 11]. It is presumed that the rotating units, by changing direction and intensity of air flows incoming into the cooling tower, allow to provide cooling of circulating water down to sufficiently low temperatures by means of providing its contact with air at large area, with uniform filling sub-spray space with minimum length of the dead zones. In evaluation of technological efficiency of air regulating units in operation of circulating systems of water supply for turbine capacitors at power plants, the value of thermal efficiency coefficient of tower's cooling capacity is used

$$\eta = \frac{T_{1w} - T_{2w}}{T_{1w} - \tau} \tag{1}$$

where T_{1w} -water temperature at the cooling tower entrance , T_{2w} - water temperature at the exit , τ -wet bulb thermometer temperature. In order to analyze thermal efficiency of the cooling tower, the dependence of the effectiveness ratio from relative air mass consumption Q_a to water flow rate Q_w is reviewed in form of a parameter $\lambda = Q_a / Q_w$.



1	- water-catcher;	4	- hot water inlet;
2	- water distribution system;	5	- cold water output;
3	- cold water basin;	6	- air feeding.

Fig.1. Structure of the cooling tower.

2. Mathematic model description

Thermal design of cooling towers with use of air control technology and analysis of thermal-and-mass exchange processes is conducted on mathematic model. Evaporating and heat-exchanging cooling of water moving in the air flow is described in the model by system of regular and differential equations, as follows:

$$\frac{\partial \vec{U}}{\partial t} + (\vec{U}\nabla)\vec{U} = -\frac{1}{\rho}\nabla p + \nu\Delta\vec{U} + \vec{g} - 2\left[\vec{\Omega}\times\vec{U}\right]$$

$$\frac{\partial \rho}{\partial t} + div(\rho\vec{U}) = 0$$

$$\frac{\partial T}{\partial t} + (\vec{U}\nabla)T = \kappa\Delta T$$

$$\rho = \rho(T)$$

$$\frac{dG_w(x_3)}{dx_3} = -\gamma B\cdot\left[\rho_s(x_3) - \rho_v(x_3)\right].$$

$$\frac{dG_v(x_3)}{dx_3} = \gamma B\cdot\left[\rho_s(x_3) - \rho_v(x_3)\right]. \tag{2}$$

$$\frac{d}{dx_3}J_a(x_3) = \alpha b\cdot\left[T_w(x_3) - T_a(x_3)\right].$$

$$\frac{d}{dx_3}J_v(x_3) = -b\cdot\begin{Bmatrix}\alpha\cdot[T_w(x_3) - T_a(x_3)] \\ -r\cdot\gamma\cdot[\rho_s(x_3) - \rho_v(x_3)]\end{Bmatrix}$$

Where $\vec{U}$ – velocity vector; T – temperature; p – pressure; ρ – density; g – acceleration of gravity; t – time; ν and **κ** – respectively, the coefficients of molecular viscosity and thermal conductivity; Ω – angle velocity of flow rotation in the cooling tower; J – specific enthalpy (heat content), J/kg (ccal/kg); Q – mass flow rate (kg/s); F – irrigated area, m^2; b – height of the irrigator, m. B – width of the irrigator, m; f – relative humidity, %; r=2493 kJ/kg (595 ccal/kg) – specific heat of evaporation; α – heat exchange ratio, $W/(m^2\cdot°C)$; γ – mass transfer coefficient, $kg/(m^3\cdot s)$.

Indexes: a – (air); s – (saturated); v – (vapor); w – (water); 1 – at the cooling tower entrance, 2 – at the cooling tower exit.

Having transformed the equations using Boussinesq approximation and the equation for air humidity condition in the line via function of current and velocities potential, the rate setting was performed on the obtained equations system by scale numbers: for length – tower height *H*, for temperature temperature ox external air- T_a, for velocity V - $\sqrt{gH}$, for time t - $\sqrt{H/g}$.

The border conditions for the equations system are presented as follows:

- the temperature of external air T_{1a}, its humidity φ and pressure P_a, thermal flow $-C_a\rho_a\lambda_a\frac{\partial T_a}{\partial x_3} = Q(x_2)$, air enthalpy J_{a1}, velocity of the flows incoming the tower, flow vortex and air flow rate $U_1 = U_1^{in}, U_2 = U_2^{in}$, $\Omega = \frac{U^{in}}{R}\sin\chi, G_a$, where χ – angle of flow entrance into sub-irrigating space, calculated from radial direction, R- radius of supporting ring of the tower, are set in the lower section of the cooling tower, at the entrance in its sub-irrigating space;
- in the upper section of the irrigator, the initial water flow rate G_w, water temperature T_{1w}, steam enthalpy J_{a1}.for initial water temperature, are set.

The mathematic model of the cooling tower was effected in mathematic package Matlab. Check of verification and justification of the numerical model for homogeneous liquid flow was conducted by comparing the calculation results with data of laboratory measurements. Analysis of calculated and experimental data allowed to select empiric parameters. Good coincidence which was achieved in wide range of Froude and Reynolds numbers leads to conclusion that the proposed mathematic model provides adequate description of heat-and-mass exchange processes and air-hydrodynamics. The model's sensitivity towards change of stated parameters was researched as well. In relatively wide range of changing parameters of turbulent viscosity, ratio of turbulent heat exchange and ration of turbulent mass exchange, the results of numerical computations do not differ sufficiently. It confirms that the model is stable and not sensitive to error in parameters determination. Number of nodes was changed from 11 to 81. At that, it was found that the results of computations do not depend strongly of computation mesh density. For practical computations quite small number of mesh nodes is sufficient. The model simulations revealed that the conditions of stationary state are stabilized in the simulated area after 4 hours.

3. Discussion of numeric simulation results

Discussing possibilities of airflow control influence over thermal efficiency, let us provide the results of numeric model computation at various angles χ of rotating units installation in air flow conducting windows of the cooling tower. The calculations were conducted for standard conditions of operation mode with the water flow rate, supplied to the tower, Q_w=1 m³/h; incoming water temperature T_{w1},=40°C; external air temperature T_a=25°C; air relative humidity f=50% and values of air mass flow rates Q_a=840 m³/h , Q_a=268 m³/h.

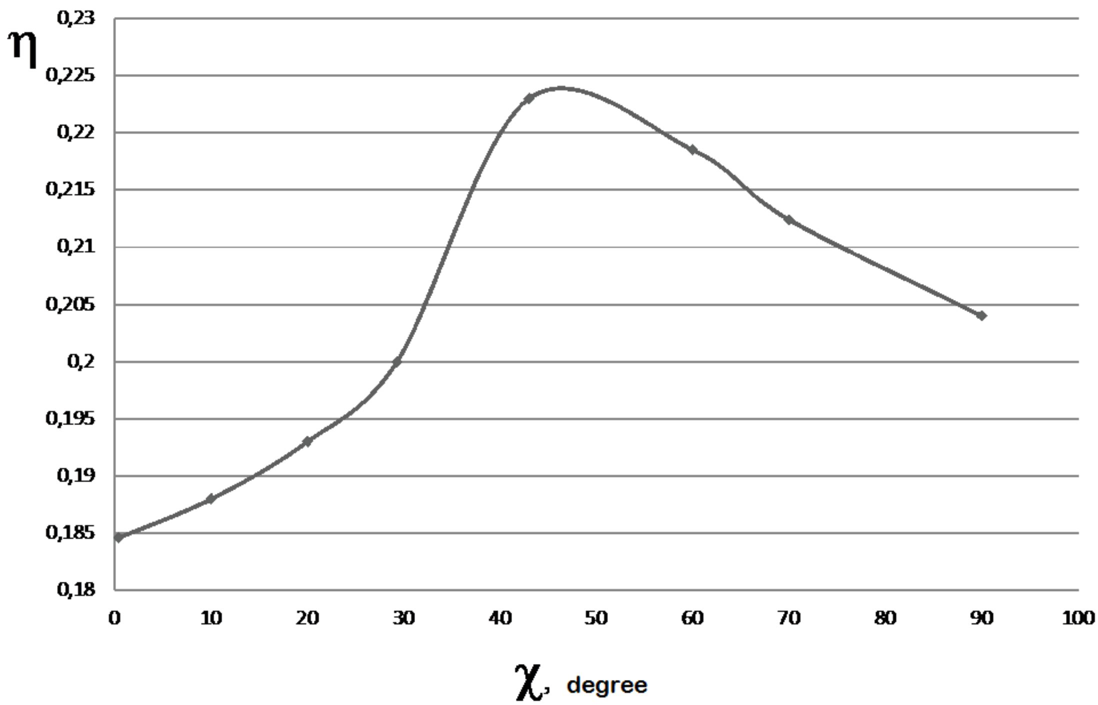


Fig. 2. Plot of the efficiency ratio dependence at given air flow input angle χ from 00 to 900 and relative flow rate Qa/Qw=1.1.

Quite indicative dependence of thermal efficiency from the flow input angle into the tower is found. Fig.2 shows that maximum efficiency ratio of the cooling tower operation at fixed relation between air and water flow rates Q_a/Q_w=1,1 is achieved at values of air flow input angle into laboratory cooling tower in range from 30 to 60 degree. The points represent data of numeric calculation for laboratory model of cooling tower.

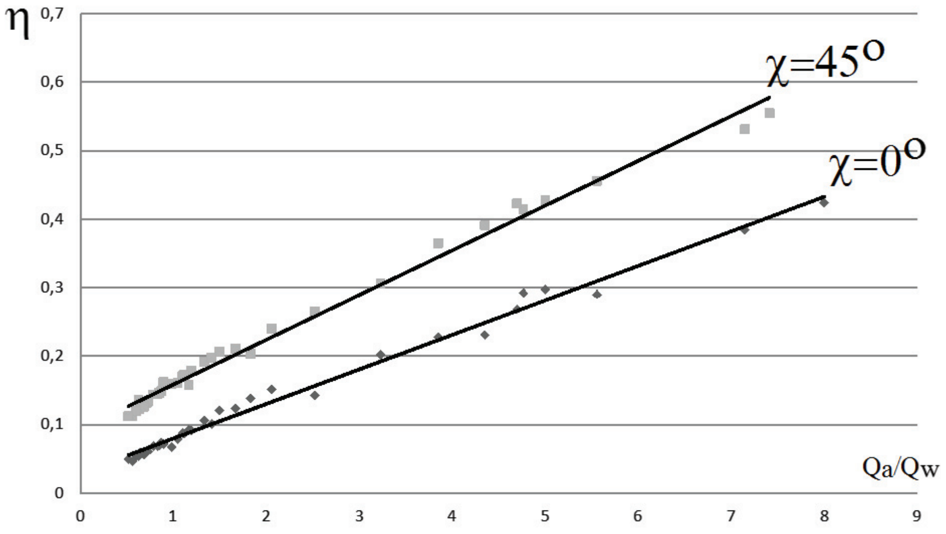


Fig. 3. Change of the tower thermal efficiency ratio against relation between air-dynamic and hydraulic loads Qa/Qw in case of open cooling tower and the tower with flow vortex along whole perimeter by air-regulating devices.

The difference in efficiency ratio between open cooling tower and cooling tower with air controlling units, providing flow tower input angle 45^0 is 3.9%, which is equivalent to increase of generated electric power by 1.5 MW. With change of relation between hydraulic and thermal load, the efficiency of cooling capacity of evaporating cooling tower is changed as well. Influence of both direction of incoming flows and their intensity over the efficiency and degree of water cooling in the tower was confirmed not only by numeric calculations but also by data received during measurements in the laboratory experiments. Fig. 3 provides characteristic dependencies of tower thermal efficiency from relative water-air flow rate in comparison with data of numeric calculation. Therefore, the value of tower thermal efficiency ratio can be controlled not only by adjusting direction of air flows incoming into tower sub-irrigating space. The tower thermal efficiency ratio can be changed also by using systems controlling intensity of air flows by means of air-regulating devices installed simultaneously at fixed angle along the whole perimeter of the tower base.

4. Conclusions

Among presented results one should note that representation of the thermal efficiency ratio in coordinates of water-air ratio reveals not equal to zero values of efficiency ratio in absence of air flows. The lines cross the Y-axis in points practically close to zero, but not at zero values. It means that inn these two cases, when there is a vortex in the tower, and when the flow is u-free, water cooling is stipulated only by evaporation which is equal for both cooling modes.

The analysis of simulating parameters distribution fields in the tower reveals that the atmospheric air flow during its movement, in addition to heat flow meets various system elements which also give thermal energy to the air flow, at that, the latter is losing its efficiency when interacting with the heat flow. It is necessary to design air flows input directly at the point of direct contact with water flows. This can be achieved by intaking the air directly from atmosphere (without its contact with heat transfer medium) and directing it straight into heat exchange with water zone, in order that the cooling air having passed the zone was capable to exit into atmosphere with minimized parasitical contacts and air-dynamic resistance.

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