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Water-air aerosol cooling of rows of cylindrical elements under conditions of natural convection

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Abstract. The article notes the disadvantages of application of dry cooling towers and air-cooled condensers for heat elimination. The cooling air humidifying is considered as a prevalent method of these heat removers efficiency increasing. The way of heat-transfer coefficient rise is proposed using the water mist which enables the latent heat of water evaporation. The physical process of cooling with water mist is analyzed. The effect of water mist rate on the heat transfer enhancement is investigated experimentally in conditions of natural convection. The experimental facility with a heated tube bundle and transversal flow is described. The results of the experiments are given in the form of Nusselt number depending on Rayleigh number and water mist rate.

1 Introduction

In late decades the so-called dry cooling towers and air-cooled condensers find wide application in energy field. These heat exchangers dissipate heat in the atmosphere by heat transfer through heat-exchange surface without evaporation [1]. This way of heat removal makes it possible to save water resources, that is especially important in water-short regions [2-5].

Despite the advantages of dry cooling towers and air-cooled condensers, dry (air) cooling is less efficient than convenient water cooling. The increase in the temperature of the surrounding air in summer months leads to significant reduction of the efficiency of cooling in dry cooling towers and air-cooled condensers and, consequently, to the decrease of power plants capacity [6].

The design of the dry cooling towers and air-cooled condensers differs according to the way of cooling air supply. It can be forced circulation in a mechanical-draft tower (Fig. 1a and 1b) and natural circulation in a chimney-type cooling tower (Fig. 1c).

The majority of published researches of enhancement of dry cooling methods thermal efficiency considers the variants with forced air movement and humidifying by water micro-drops (aerosol) [7-13]. This way of cooling increases significantly the efficiency of heat exchange, at the same time reducing the amount of water fed to the cooling air [14-16], but it demands the additional expenditure of energy for air-supply blowers' drive.

The way of enhancing heat transfer in dry cooling towers by creating the conditions of natural convection of the aerosol flow over heated surface is much less investigated.

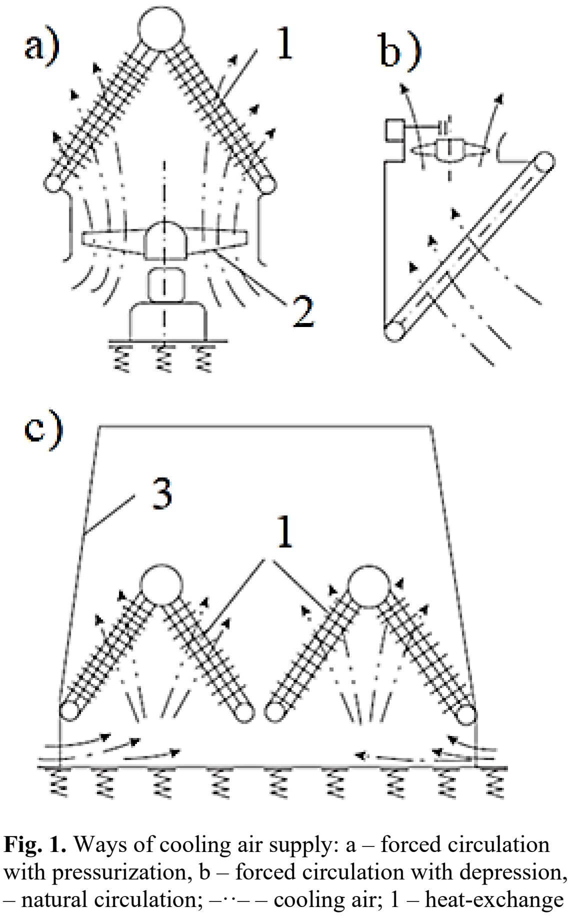

Fig. 1. Ways of cooling air supply: a – forced circulation with pressurization, b – forced circulation with depression, c – natural circulation; –·· – – cooling air; 1 – heat-exchange surface; 2 – fan; 3 – tower.

The results of experimental research of water and air aerosol cooling of rows of heated cylinders (a model of a heat exchanger) in conditions of natural convection of

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air and aerosol flow are given in this work. The research had two main objectives. The first was to measure the actual heat transfer coefficient and to estimate the heat removal efficiency. The other one was to obtain an empirical formula to predict the degree of heat exchange intensification effect of the water and air aerosol flow that arises due to natural convection in a channel with heat-dissipating tubes.

2 Materials and methods

A schematic diagram of the facility for investigation of heat exchange of cylindric heaters (tube bundle) and water and air flow in conditions of natural convection is shown in Fig. 2.

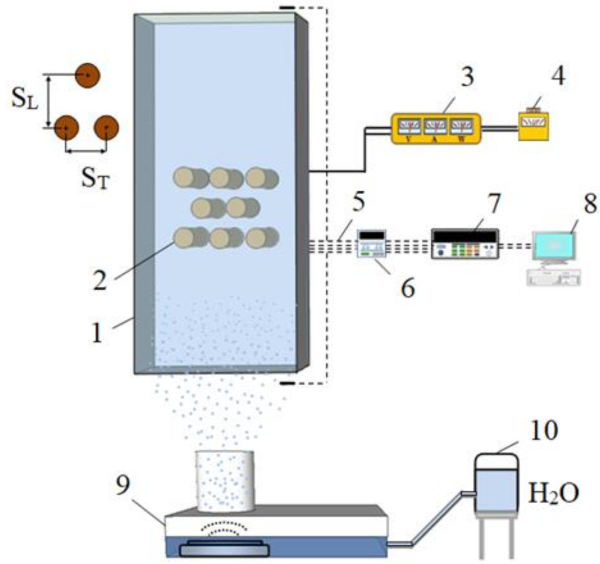

Fig. 2. The schematic diagram of the experimental facility: 1 – the heat-insulated channel; 2 – the tubes of the heat exchanger; 3 – the multimeter; 4 – the voltage regulator; 5 – the thermocouples; 6 – the analog output module; 7 – the data acquisition module; 8 – the computer; 9 – the ultrasonic water mist generator;10 – the water tank.

The main part of the experimental facility is the heat exchanger which contains three staggered rows of cylindric heaters in a rectangular channel. The facility also includes an air blower, a water mist generating system, and a system for data handling.

The bundle of the heat-exchanger consists of copper tubes with cartridge heaters that are located inside the tubes. The thermocouples are placed at the back side of the tubes. Each heating element is controlled with an autotransformer to obtain the heat flux demanded. Before the heaters were mounted, the inner side of each heat exchanger tube was covered with a thin layer of heat-conducting paste in order to reduce the contact thermal resistance between the heater and the copper tube.

The heat exchanger tubes have outer diameter d_t = 14 mm and wall thickness 3 mm. The construction of a tube is shown in Fig. 3. The assembled tubes are fastened in a vertical heat-insulated channel, as it is shown in Fig. 4. The channel of rectangular cross-section measuring 117 mm x 55 mm has the length of 1 m. The heat-exchange tubes are fixed inside the channel at both ends. The textolite rods of 4 mm diameter are used to fix the tubes in the channel. The material of these rods has the heat transfer coefficient equal to 0.023 $W \cdot m^{-1} \cdot K^{-1}$.

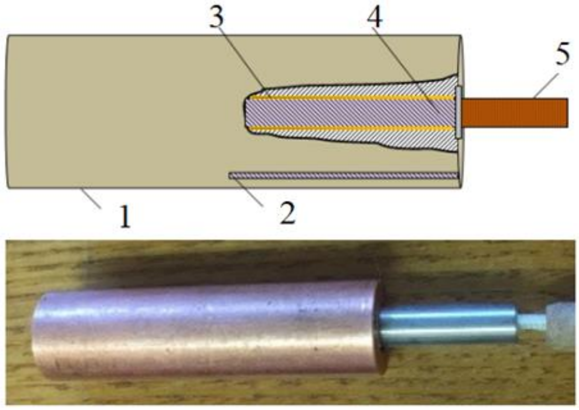

Fig. 3. The scheme and the photograph of a heat-exchange tube: 1 – the cylindric heating element; 2 – the thermocouple; 3 – the heat-conducting paste; 2 – the electric heater; 5 – the textolite rod.

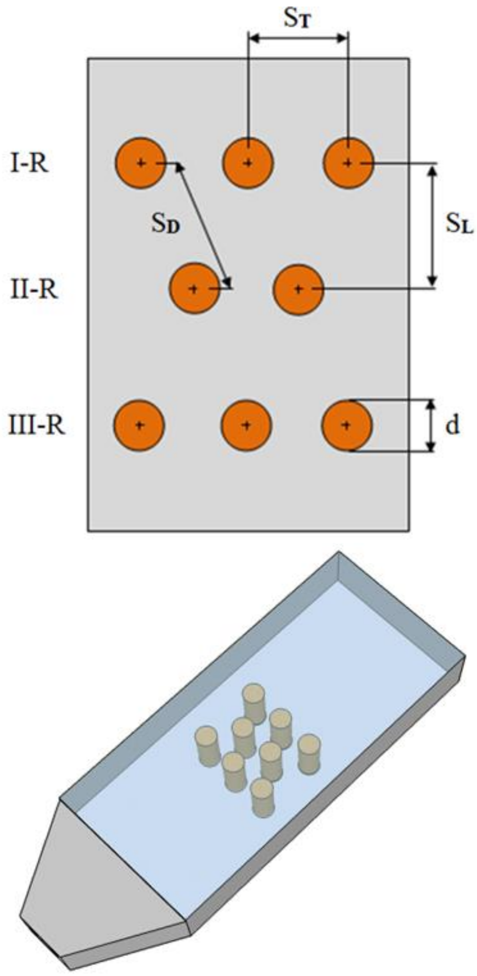

Fig. 4. The arrangement of tubes in the channel.

The temperature of the tubes' surface was measured directly by chromel-copel thermocouples, which were installed at the back side of each tube at an angle of 180 degrees from the frontal point. A thermocouple was placed at the area of flow entrance. Two other thermocouples were put where the flow leaves the channel. In addition, the temperatures of the surrounding air and the flow at the channel entrance and exit were measured constantly, as well as the time of the tube bundle cooling. All thermocouples are connected to a data acquisition system.

The water and air aerosol flow was created by injection of water micro-drops in the air flow. The system of drops generation consisted of an ultrasonic mist generator with piezoelectric converter and generator which worked at 1.7 MHz frequency. The ultrasonic generator created water drops with the diameter range from 1 to 10 micrometers.

The steady-state measurements were conducted by atmospheric pressure. The rate of water injection into air (water concentration) j varied from 23.39 to 111.68 kg·m^{-2}·hr^{-1}. The values of heat-flux density q varied from 373.8 to 1843.7 W·m^{-2}.

The values of heat transfer coefficient α were determined by the given heat-flux density on the surface of the cylindric heaters and the difference between the temperature of the cylinder surface Ts and the temperature of the incoming water and air flow T_f:

$$\alpha = \frac{P_e - Q_w}{A_c \cdot \left(T_s - T_f\right)}, \tag{1}$$

where P_e is the electric power of the heater, W; A_c is the side surface area of the tubes, m2; Q_w is the thermal loss in the tube and channel wall attachment point, W.

To analyze the process of heat exchange of the cylindric heaters and aerosol flow in the channel, the following parameters were calculated:

- The flow Reynolds number

$$\mathrm{Re} = U \cdot d_t / \nu_f, \tag{2}$$

- The Nusselt number

$$\mathrm{Nu} = \alpha \cdot d_t / \lambda_f, \tag{3}$$

- Water concentration at the channel entrance

$$j = G / A_{ch}, \tag{4}$$

where U is the flow velocity in the narrowest cross section of the channel, m/s; d_t is the tube diameter, m; ν_f is the flow kinematic viscosity, m^2/s; λ_f is the flow thermal conductivity coefficient, W·m^{-1}·K^{-1}; G is the mass of water sprayed in unit time, kg/s; Ach is the area of the channel cross-section, m^2.

While analyzing the heat exchange between the flow of fine-air water mist and the heated surface, it is usually considered the total quantity of heat which is transferred by the air and water mixture, and the heat which is expended for the water heating and evaporation [16, 17]:

$$Q = \alpha_c \cdot A_c \cdot (T_s - T_f) + j \cdot r \cdot A_c = \\ = \alpha_c \cdot A_c \cdot [(T_s - T_f) + r \cdot n \cdot h / \alpha_c (x_s - x_f)], \tag{5}$$

where α_c is the heat transfer coefficient from the tube surface to the water and air mixture, W·m^{-2}·K^{-1}; x_s and x_f is the moisture content of saturated air, relative units, at the temperatures T_s and T_f; r is the latent heat of vaporization, J/kg; h is the mass transfer coefficient, kg·m^{-2}·s^{-1}; n is the share of the moisten tube surface in relative units.

Using the Lewis relation,

$$\alpha_c / h = C_p = 1{,}005 + 1{,}8 \cdot 10^{-3} \cdot x_f, \tag{6}$$

where C_p is the water and air mixture heat capacity, the equation (5) can be written as

$$Q = \alpha_c \cdot A_c \cdot [(T_s - T_f) + r \cdot / C_p \cdot n \cdot (x_s - x_f)]. \tag{7}$$

If the velocity of fine-air water mist is high, a part of the drops can bed in on the surface of a tube avoiding vaporization in the boundary layer. In this case the heat is removed due to the heating to the surface temperature of the water mist which beds in in the form of separate drops or water film. The resulting heat from the cooled surface can be defined as

$$Q = A_c \cdot [\alpha_c \cdot (T_s - T_f) + j \cdot r \cdot + j \cdot C_p \cdot (T_s - T_f)]. \tag{8}$$

So, the effective heat transfer coefficient α can be found as a sum of the heat transfer coefficient from the surface to the water and air mixture and the heat transfer coefficient that concerns evaporation $\alpha_r = j \cdot r / (T_s - T_f)$:

$$\alpha = \frac{Q}{A_c \cdot \left(T_s - T_f\right)} = \alpha_c + \alpha_r \left(1 + \frac{C_p\left(T_s - T_s\right)}{r}\right). \tag{9}$$

The Rayleigh number Ra = Gr·Pr was found as follows:

$$\mathrm{Ra} = \frac{\beta \rho^2 g \left(T_s - T_f\right) d_t^3}{\mu^2} \cdot \frac{\mu C_p}{\lambda_f}, \tag{10}$$

where Gr is Grashof number, Pr is Prandtl number, β is the coefficient of volumetric expansion, 1/K; ρ is the water and air mixture density, kg/m^3; g is the gravitational acceleration, m/s^2; μ is the dynamic viscosity of the mixture, Pa·s.

The Weber number is

$$\mathrm{We} = j^2 \cdot d_d / [2 \rho_w \sigma], \tag{11}$$

where d_d is a drop diameter, m; ρ_w is water density, kg/m^3; σ is water surface tension coefficient, N/m.

According to estimation conducted, the errors of determining the Nu and Ra values were 5 and 5.42 percent respectively.

3 Results and discussion

The experimental study of heat exchange between the staggered cylindric heaters and the flow was conducted in conditions of natural convection for both dry air and water and air aerosol. The results of experimental data processing are shown in figures from 5 to 6.

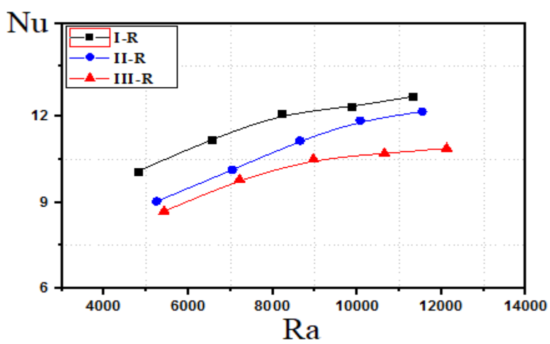

Fig. 5. The cooling of the cylindric elements in the channel in conditions of natural convection for the 1st, the 2nd and the 3rd rows (see Fig. 2).

Fig. 5 shows the dependency of the row averaged Nusselt number on the Rayleigh number for dry air cooling in conditions of natural convection. When the forced convection is absent, the cylindric heaters have uniform heat release, and the coolant moves due to natural convection. A thin boundary layer is formed in vicinity of each heated cylinder, where the temperature varies from the maximum temperature of the cylinder surface (T_s) to the flow temperature (T_f). The heat transfer coefficient between the heated cylinder and the environment is determined by steady-state natural convection. The thermophysical properties of the coolant were assumed to be constant.

According to expectations, the average Nusselt number grows significantly as the Rayleigh number increases. As follows from Fig. 5, the Nusselt number of the first row is larger than that of the second and the third rows for the whole range of Ra numbers. The cause of this is that the air surrounding the second and the third rows is heated preliminary.

The effect of the water spray rate was explored in the range of *j* from 20.48 to 97.85 kg·m^{-2}·hr^{-1}. The Rayleigh numbers varied from 1800 to 12000. The evaporation of water drops on the surface of cylindric heating comes into being when the water and air aerosol flow washes the tube bundle.

Fig. 6 shows that the Nusselt number, therefore, the heat transfer rate, grows coupled with the water spray rate in the case of constant heat flux. It indicates that the air with suspended water drops can be an effective coolant for temperature regulation and heat removal from the heated surfaces.

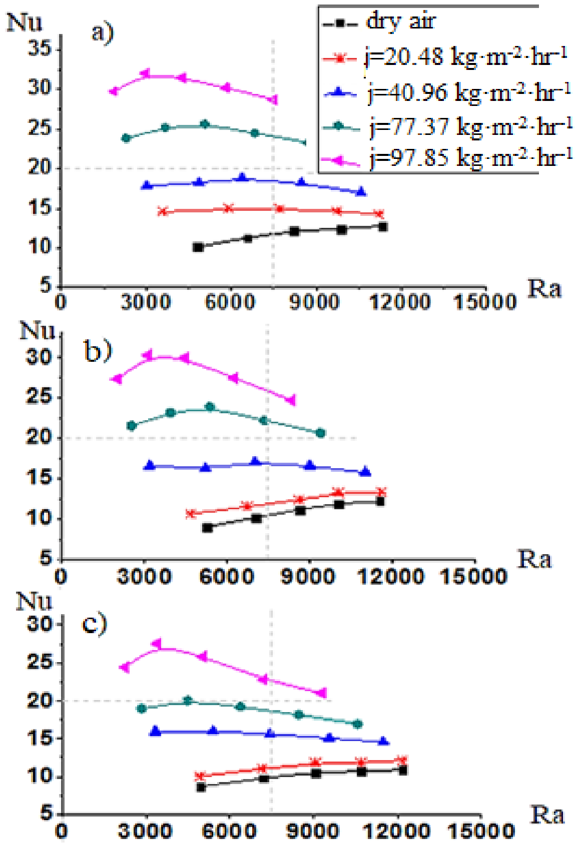

Fig. 6. The influence of water concentration j on the cylinders' cooling in the channel: a – the first row, b – the second row, c – the third row.

In addition, the average Nusselt number of each row grows till the maximum value for each water concentration, and then decreases along with the Rayleigh number increase. This can be explained with an increase in the temperature difference due to the water film that forms on most of the surface of the cylindrical elements and creates an increased thermal resistance. Therefore, Nu number reaches its maximum at a certain ΔT.

The Nusselt number increases by about 148%, 144% and 128%, respectively, for each row of cylindrical elements at j = 97.85 kg·m^{-2} hr^{-1}, compared to dry air cooling.

The intensification of heat transfer by water drops can include four main physical processes [16, 17]: evaporation of water microdroplets on the surface of cylinders, evaporation of a water film on the surface of cylinders, convective heat transfer, and a decrease in air temperature due to an increase in its humidity.

The processing of the results obtained made it possible to deduce a formula for estimating the heat transfer enhancement as a function of the Rayleigh (Ra) and Weber (We) dimensionless criteria:

$$Nu_m / Nu_0 = 1 + C\,Ra^{-0.79}\,We^{0.67}, \qquad (12)$$

where Nu_m is the Nusselt number for heat removal by water and air aerosol (mist) and Nu_0 is the Nusselt number for dry air cooling. Coefficient C is 1.95 10^7; 1.91 10^7 and 1.85 10^7 for the first, second and third row of tubes, respectively.

4 Conclusion

The heat exchange of water aerosol (mist) and a heated tube bundle was experimentally studied at low flow rates of finely dispersed water with water density j = 20.48 – 97.85 kg m^{-2} hr^{-1}. Studies have shown that the efficiency of heat removal from a tube bundle can be significantly increased by using microaerosol water mist. Nusselt number increases by about 148%, 144% and 128%, respectively, for the first, the second and the third rows of tubes compared to single-phase air flow at water density j = 97.85 kg m^{-2} hr^{-1}. The heat transfer efficiency of each tube is highly dependent on the temperature difference ΔT. This indicates the importance of evaporation of the water film from the wetted part of the tube surface to improve heat transfer. An empirical formula has been obtained for estimating the dimensionless heat transfer coefficient (Nussel number) as a function of the Rayleigh number and the Weber number.

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