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INVESTIGATION OF THE INTERNAL AERODYNAMICS OF THE CHIMNEY-TYPE EVAPORATIVE COOLING TOWER

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We propose a technique for determining the hydrodynamic drag of the chimney-type evaporative cooling tower on the basis of temperature measurements and the mathematical model developed by us. The presence of return vortex flows in the upper part of the subsprinkling compartment of the cooling tower is shown. The air velocity in the sprinkler and the radius of the stagnant zone in the central part have been estimated.

Introduction. Chimney-type evaporative cooling towers are designed for cooling circulation waters at thermal and nuclear power plants [1, 2]. The present-day cooling towers have the following characteristic parameters: tower height 100 m, base diameter about 70 m, and water discharge ~30,000 m^3/h or more. In such a counterflow cooling tower, water cooling occurs as it runs down in sheets by the sprinkler shields or trickles downward in the super- and subsprinkling compartments (Fig. 1).

Obviously, the ascending air velocity in the vicinity of the cooling-tower sprinkler and its distribution play an important role in the evaporative cooling and depend on: 1) the state of the surface air layer; 2) the initial temperature of the water being cooled; 3) the height of the air-blowing windows; 4) the radius of the cooling-tower sprinkler; 5) the radius of the cooling-tower chimney; and 6) the chimney height over the sprinkler.

The heating of incoming cool air and its saturation with water vapors as it goes upwards, passing through the region where water runs down in sheets, influence the efficiency of evaporative cooling of water droplets above the sprinkler. Additional heating of air in the supersprinkling compartment due to the heat exchange on the droplets increases the velocity of convective air flows in the cooling tower. This intensifies the processes of evaporative cooling in the sheet flows on the sprinkler shields where the basic cooling of the circulation water occurs. The various cooling zones in the cooling tower are interconnected through both the gas and water phases moving in opposite directions.

It is common to estimate the efficiency of a chimney-type cooling tower by the quantity η:

$$\eta = \frac{T_{w0} - T_{w.out}}{T_{w0} - T_{lim}} . \tag{1}$$

The limiting temperature of water cooling in evaporative cooling T_{lim} is found from the equation

$$\rho_s (T_{lim}) = \varphi \rho_s (T_a) . \tag{2}$$

In the mathematical model of the cooling-tower functioning [3], its internal aerodynamics is described in the one-dimensional approximation. Our experimental data thereby are used to correct the one-dimensional aerodynamic calculations. Comparison of the experiment with the calculation has shown that the mathematical model describes with a high enough accuracy all experimental data obtained for the thermal efficiency of the cooling tower. Moreover, this model permits good calculation of the air temperature over the sprinkler inside the cooling tower [4].

It has been shown by experiment that the aerodynamics of the cooling-tower input [5, 6] produces a significant effect on the efficiency of its operation, including the influence through the action of the ground wind [7]. The aim of the present paper is to present the results of the investigation of the internal aerodynamics of cooling towers necessary for gaining a deeper insight into the functioning of a cooling tower and optimizing its operation [5, 8].

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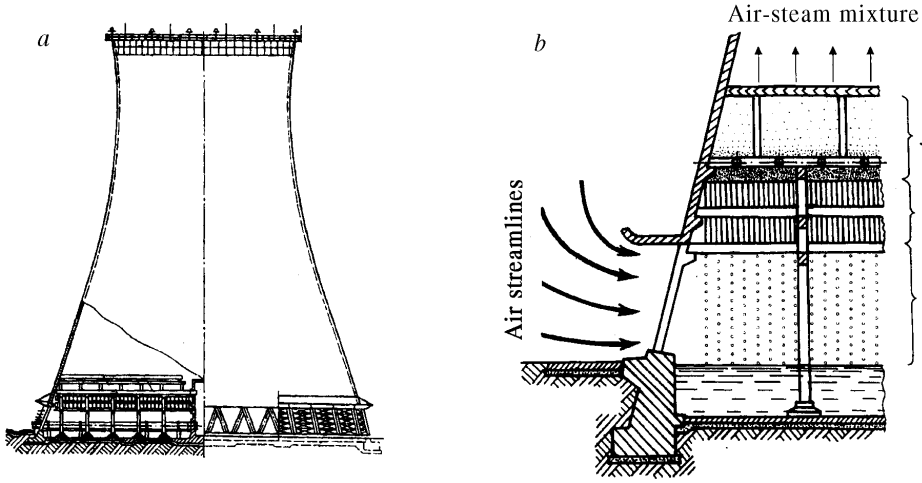

Fig. 1. General view of the chimney-type evaporative cooling tower (a) and lower part of the cooling tower in the region of the air-blowing windows (b) [1) subsprinkling compartment; 2) sprinkler shields; 3) supersprinkling compartment].

Some tentative results were published in [9]. Other experimental data on internal aerodynamics of chimney-type cooling towers are unknown to the authors.

Qualitative Assessment. Following the continuity equation of [10], the mean velocity of air ascent through the cooling-tower sprinkler shields v_a can be calculated by the formula

$$v_a = v\,\frac{\pi R_{t.h}^2}{\pi R^2 b}\,. \qquad (3)$$

where $\pi R^2 b$ is the area of the flow section of the cooling-tower sprinkler. The coefficient b takes into account the difference between the flow and geometric sections of the sprinkler. For cooling towers of the BG-3200 type used at the Minsk TÉTs-4 thermoelectric plant, the value of $b \approx 0.8$. In turn, the mean velocity of incoming cooling air through the air-blowing windows of the cooling tower v_w is related to the quantity v_a by the relation

$$v_w = v_a\,\frac{Rb}{2h}\,. \qquad (4)$$

The dimensionless complex $Rb/2h$ determines many properties of the internal aerodynamics of the cooling tower. It can be shown that the closer its value to 1, the better it is. Using formulas from the theory of free convection [12], the velocity of the steam-air mixture v can be given in the form

$$v = ((2gH\Delta\rho_m)/\rho_m)^{0.5}\,k\,. \qquad (5)$$

The empirical coefficient k takes into account the total hydrodynamic drag of the cooling tower, including the turbulent friction loss in the cooling-tower chimney and sprinkler shields, the "rain" drag in the subsprinkling compartment, and the input drag. The change in the steam-air mixture density $\Delta\rho_m$ in (5) is calculated in the process of the self-consistent solution of the system of differential equations describing the processes of heat and mass exchange in the cooling tower [3]. In so doing, $\Delta\rho_m$ is largely determined by the change in the temperature of the steam-air mixture passing between the sprinkler shields. In [3, 11], it was assumed that the value of $k \approx 0.5$. It should be remembered that the size of the zone of intensive evaporative cooling approximately equal to the height of the cooling-tower sprinkler is much smaller than the height of the cooling-tower chimney, which permits speaking of a constant velocity of motion of air through the sprinkler.

Expressions (3)–(5) describe the internal dynamics of the cooling tower in the one-dimensional approximation. For a whole number of cooling-tower designs the inequality $v_w > v_a$ holds, which inevitably leads to the appearance of aerodynamic and thermal effects associated with the nonuniform distribution of incoming cooling air flows. Indeed, there are a number of two-dimensional effects in the subsprinkling compartment revealed during the laboratory modeling of the cooling tower [5].

We estimate the size of the nonuniformity zone. In the approximation of cylindrical symmetry, denote the horizontal component of the incoming air-flow velocity by $w(r, z)$ and its vertical component by $u(r, z)$. Let the ordinate axis pass through the center of the cooling tower and the origin of coordinates coincide with the water surface in the drainage basin. From the continuity equation [12] we have

$$\frac{\partial w}{\partial r} + \frac{\partial u}{\partial z} = 0 . \tag{6}$$

For calm the following boundary conditions in the center of the subsprinkling compartment hold true:

$$\frac{\partial w (0, z)}{\partial r} = 0 , \quad u (r, 0) = 0 . \tag{7}$$

For air incoming at a distance h_1 from the upper section of the air-blowing window, we determine the value of the breaking length of the horizontal velocity as $R - R_s(h_1)$ [10]. Here the characteristic radius of the zone where the horizontal velocity component is absent is denoted by $R_s(h_1)$. Then we obtain the following estimates:

$$\frac{\partial u}{\partial z} \sim v_a / h_1 , \quad \frac{\partial w}{\partial r} \sim v_w / (R - R_s (h_1)) .$$

From (6) and the above formulas we have

$$R_s (h_1) \sim R \left(1 - \frac{b h_1}{2h} \right). \tag{8}$$

As follows from (8), the higher the entrance point of air, the smaller the path it will cover in the radial direction. For the air flow entering in the vicinity of the lower section of the air-blowing window, $R_s(h) \sim 0.6R$. Thus, the following practically important conclusion can be drawn: at least about 36% of the total area of the sprinkler operates at a lower density of the mass flow of air than follows from the one-dimensional aerodynamic estimates, and this fact decreases the thermal efficiency of the cooling tower. "Fresh" air necessary for evaporative cooling is conveyed into the stagnation zone by means of secondary flows and turbulent diffusion. In this connection, the development of designs with smaller sizes of the stagnation zone is deemed to be a topical problem [6].

Estimates show that the Reynolds number for air flows in the subsprinkling compartment $\mathrm{Re} \sim 10^6$; therefore, air flows in it are turbulent. In this case,

$$\mathrm{Re} = \frac{h \rho_m v_w}{\mu} .$$

Experimental results. The investigations were carried out at the Minsk TÉTs-4 thermoelectric plant, where the chimney-type hyperbolic evaporative cooling towers have the following parameters: sprinkling area — 2300 m^2, tower height — 80 m, base diameter — 70 m, air-blowing window height — 5 m, and a maximum circulation water discharge of 30,000 m^3/h. Experiments were performed in summer, during daylight hours, close to noon, when the state of the atmosphere is maximally stabilized.

The temperature and humidity of ambient air and the velocity and direction of wind were measured by metrologically verified instruments. Water temperature was registered by TL-4 mercury thermometers with a scale factor of 0.1°C. The air-flow velocity was measured by an ASO-3 propeller anemometer. The discharge of water flowing into the cooling tower was determined in two ways: by differential manometers and by an ultrasonic flow meter.

TABLE 1. Experimental Data for Calculating the Incoming Air-Flow Velocity

Experimental number	T_{w0}	$T_{w.out}$	T_a	$T_{a.t}$	φ	v_w (calculated)	Q_w	η
1	30.9	24.0	20.0	27.4	50.4	1.7	9270	31.15
2	31.7	24.7	17.4	27.1	60.0	1.9	9270	31.16
3	31.5	24.3	20.8	28.1	51.4	1.7	9270	33.2
4	32.3	25.5	21.6	28.9	51.0	1.7	9270	31.1
5	33.1	26.1	23.0	29.8	45.8	1.7	9270	30.37
6	31.0	26.1	24.2	28.8	44.7	1.68	9270	23.92

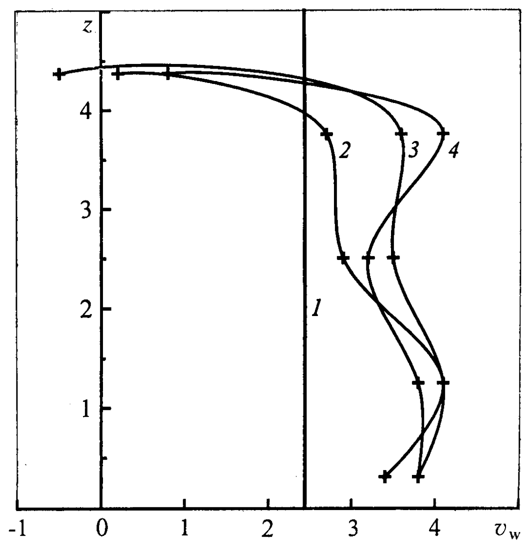

Fig. 2. Profiles of the horizontal air velocity at the section of the air-blowing windows: [1) calculated value of the velocity with regard for the experimental hydrodynamical drag coefficient *k*; 2, 3, 4) experimental values of the velocity on the window height *z* for the windward, leeward, and lateral sides of the cooling tower, respectively].

Measurements were taken at 30-min intervals by the following technique. The temperature T_a, the relative humidity φ, the velocity v_w, and the direction of wind were determined. Parallel measurements of the volume flow Q_w and of the inflowing water temperature T_{w0} and outflowing water temperature $T_{w.out}$ in the cooling tower were made. Simultaneously, the air temperature at 6–8 points above the sprinkler inside the cooling tower was measured; its averaged values $T_{a.t}$ are given in Table 1. For four positions relative to the external wind, on the windward, leeward, and lateral sides of the cooling tower, we determined the velocity of the air flow into the subsprinkling compartment at the air-blowing window section v_w at five characteristic height points. Experiments were carried out under various hydraulic and thermal loads on the cooling tower. Moreover, we carried out visualization of the current lines of incoming air flows into the subsprinkling compartment of the cooling tower [5].

We have taken unique measurements of the air temperature over the sprinkler of the industrial cooling tower under a small hydraulic pressure when the contribution of the evaporative cooling of water droplets over the sprinkler can be neglected.

Table 1 shows the calculated value of the mean velocity of the incoming air flow v_w at the air-blowing window section obtained by formula (5) for $k \cong 0.5$. It should be noted that at this value of the adjustable constant the calculated final temperature of water and the air-steam mixture temperature over the sprinkler are close to the obtained experimental results.

Analysis of a large file of thermal measurements made with the aid of the mathematical model of [2] shows that the value of the hydrodynamic drag is within the range of $0.46 < k < 0.53$ depending on the hydraulic load and wind velocity.

Figure 2 shows the profiles of the horizontal velocity component in the air-blowing windows on the windward and leeward sides of the cooling tower at a hydraulic pressure of Q_w = 27,000 m^3/h.

In this case, the ratio of the external wind velocity to the section-mean horizontal velocity of the air flowing into the subsprinkling compartment $a/v_w \approx 0.1$, where a = 0.3 m/sec and v_w = 2.8 m/sec. The calculated value of the mean velocity is approximately 2.5 m/sec. According to [7], under these conditions such a wind can be considered to be weak. In this case, the contribution of the evaporative cooling on droplets was taken into account and the effective radius of droplets was chosen to be 0.6 mm.

The window-section-mean air velocity calculated on the basis of the experimental points (curves 2–4, Fig. 2) differs from the calculated value of the velocity (curve 1). However, this difference between the averaged velocities for any of curves 2–4 is no more than 0.2–0.3 m/sec, i.e., it does not exceed 10%. It should be noted that the hydrodynamic drag coefficient k = 0.5 gives a satisfactory agreement between calculated and experimental values of the averaged air velocities at the air-blowing window sections.

Note that the air has the maximum velocity in the ground region at $z/h \sim 0.25$. Naturally, because of the condition of flow adhesion to a solid surface the velocities are equal to zero at z = 0 and z/h = 1. In the upper part of the window, the air velocity is very small. Therefore, the structure of the air flows in this zone is unstable and the appearance of back flows where the horizontal velocity is negative is possible (Fig. 2, curve 4). This points to the presence of return vortex flows and is additionally confirmed by experimental studies on the visualization of incoming air flows [5]. Apparently, in this case, the air flow in the subsprinkling zone of the cooling tower is vortex and the scale of vortices is $\sim 0.2h$.

As the analysis of the visualization of the air-flow patterns has shown, under a strong wind the structure of the flows becomes much more nonuniform with height than is shown in Fig. 2, and the velocity amplitude of the return vortex flows thereby increases. Vortex structures are also formed between the water surface, the lower edge of the air-blowing window, and the vertical wall of the drainage basin. The scale of these vortex structures depends on the water level in the basin and can reach 1 m.

CONCLUSIONS

On the basis of experimental measurements of the thermal efficiency of a full-scale (BG-3200 type) cooling tower under a small hydraulic load and a weak wind, using the mathematic model of evaporative cooling developed in [3], a method for determining the hydrodynamic drag of a chimney-type evaporative cooling tower (parameter k) is proposed. It has been found experimentally that the air flow in the upper part of the subsprinkling compartment of the cooling tower is return-vortex.

It has been shown that in the subsprinkling compartment of the cooling tower a stagnation zone leading to a worsening of evaporative cooling on the sprinkler shields should exist and its size has been estimated. An experimental technique for determining the stagnation zone is under development and a number of engineering methods for decreasing its size have been proposed [6, 13].

There are a number of unsolved aerodynamic problems connected with the investigation of the operation of cooling towers. We can note such problems as the modeling of the operation of a cooling tower in winter (Russia, Baltic countries, the CIS, and Scandinavia), where the aerodynamics of the cooling tower radically changes, as well as the investigation of the operation of a cooling tower under a strong wind load.

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NOTATION

h, height of air-blowing windows of a cooling tower, m; R, radius of the cooling-tower sprinkler, m; $R_{t.h}$, radius of the cooling-tower chimney, m; H, height of the cooling-tower chimney over the sprinkler, m; η, thermal efficiency of the cooling tower, %; T_{w0}, temperature of water flowing into the cooling tower, ^oC; $T_{w.out}$, temperature of cooled water leaving the cooling tower, ^oC; T_{lim}, limiting temperature of water cooling (wet thermometer temperature), ^oC; ρ_s, saturated water vapor density, kg/m^3; φ, relative air humidity, %; T_a, ambient air (entering the cooling tower) temperature, ^oC; v, mean convection rate of the air-steam mixture in the cooling-tower chimney, m/sec; v_a, mean velocity of air ascent through the sprinkler shields, m/sec; b, coefficient taking into account the difference between the "flow" and geometric sections of the cooling tower because of its being piled with the sprinkler shields; v_w, mean velocity of the cooling air entering the cooling tower, m/sec; k, empirical coefficient taking into account the total hydrodynamic drag in the air; ρ_m, ambient air density in the vicinity of the cooling tower, kg/m^3; $\Delta\rho_m$, change in the air-steam mixture density; w, horizontal velocity component of the incoming air flow, m/sec; u, vertical component of the incoming air flow, m/sec; $R_s(h)$, radius on which the air flow entering the air-blowing window at height h penetrates into the sprinkler shields, m; μ, dynamic viscosity coefficient. kg/(m·sec); Q_w, cooling water discharge in the cooling tower, m^3/h; $T_{a.t}$, mean air temperature over the sprinkler, ^oC; α, external wind velocity, m/sec. Subscripts: s, saturated; a, air; lim, limit; w, water; m, mixture; t, tower; out, water at the outlet; h, height; 0, initial, zero.

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