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# Numerical modeling of the thermal characteristics of a fan spray cooling tower

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# Numerical modeling of the thermal characteristics of a fan spray cooling tower

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**Abstract.** The work constructs a mathematical model of the heat and mass transfer process in a spray cooling tower. The mathematical model consists of the equations of motion and heat and mass transfer of water droplets and the equations of thermal and material balance of moist air, considers the polydispersity of the droplet flow and allows to calculate the cooling capacity of the cooling tower depending on the parameters of the droplet flow. The mathematical model is confirmed by the results of industrial tests of irrigation-free cooling towers. The simulation results show that the cooling capacity of a spray fan cooling tower significantly depends on the atomization dispersion and the height of the water cutting nozzles. Reducing the average diameter of the droplet flow to less than 2 mm and increasing the height of the nozzles $H > 4...5$ m make it possible to achieve the cooling performance of known cooling towers with nozzle. The mathematical model allows to calculate the cooling capacity of the cooling tower and determine the necessary individual parameters of the droplet flow for each type of equipment and consumers for which the specific energy costs for cooling is minimal.

## 1. Introduction

Cooling towers are widely used in mining, in particular for cooling water that takes heat from compressed air in the compression stages of centrifugal compressors and refrigeration dryers [1]. The most efficient type of cooling towers is fan cooling tower with a film nozzle, providing the greatest cooling capacity. Spray cooling towers are traditionally recommended for applications with lower heat loads and chilled water temperature requirements.

Experimental data on frame and tower cooling towers, which were tested using cast iron tangential nozzles B-10 with a pressure of 0.09 MPa and cast iron involute nozzles, are summarized in [2]. According to the author’s conclusions, water cooling rates in spray cooling towers reach the level of cooling towers with a film sprinkler at an irrigation density of $q = 3...4$ m$^3$/(m$^2$h), using B-10 nozzles and increasing the water pressure for spraying to 0.1…0.12 MPa. However, in the implemented spray cooling towers the cooling performance is lower than that of film cooling towers [3].

The study of the swirling flow hydrodynamics [4] made it possible to create centrifugal nozzles with the optimal shape and size of the vortex chamber and improve the quality of atomization of liquids at low pressures. Reducing energy consumption for spraying to 9…12 J/m$^2$ of droplets surface area allows improving the technical and economic performance of heat and mass transfer equipment with inexpensive reconstruction.

Such centrifugal nozzles are promising for cooling water in cooling towers. It is known [5] that the heat transfer coefficient $\alpha$ of drops is 10…16 times greater than the film on sprinkler shields, for example, for film – 11.6 W/(m$^2$K), for drops with a diameter $\delta$ = 3 mm – 138.7 W/(m$^2$K). The specific surface area $f$ of the shield sprinkler is 27 m$^2$/m$^3$, and the specific surface area of the droplets is 2.7 m$^2$/m$^3$ at $q$ = 6.9 m$^3$/(m$^2$h). Then $\alpha \cdot f$ for droplets and film are close in value and comparable water cooling is achievable at the same aerodynamic resistance.

Obtaining droplets with an average diameter of $\delta \sim$ 1.5…2 mm in nozzles with a spray pressure of 0.03...0.04 MPa improve the characteristics of spray fan cooling towers to the level of film cooling towers and reduce the cost of reconstruction by abandoning sprinklers.

The purpose of the work is construction of a mathematical model of the heat and mass transfer process in a spray cooling tower and determination of its cooling capacity.

## 2. Mathematical model of air-droplet flow

Schematic diagram of the cooling tower is shown in figure 1. Hot water is sprayed into the counter flow of the ambient air by spray nozzles (1). Ambient air enters the cooling tower through the inlet windows (2). Heated moist air exits the cooling tower through the diffuser (3). If the cooling tower is equipped with the fan it is located in the exiting air stream.

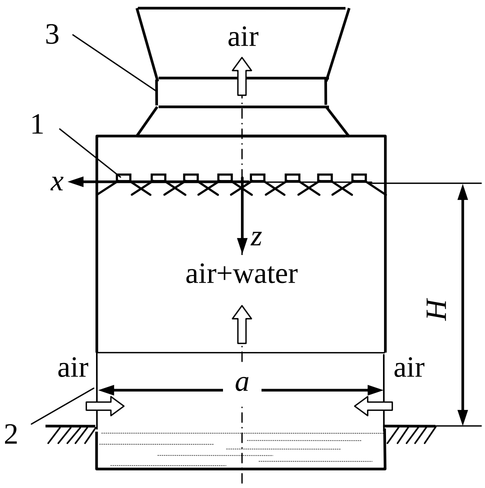


**Figure 1.** Spray cooling tower: 1 – spray nozzles, 2 – windows, 3 – diffuser, $a$ – cooling tower width, $H$ – nozzle height.

The governing equations of the air and water flow as well as heat and mass transfer in the cooling tower are:

$$\frac{\mathrm{d}V_{\mathrm{x}i}}{\mathrm{d}\tau} = -\frac{3}{4}\cdot\frac{C_{\delta i}\psi_i}{\delta_i}\cdot\frac{\rho_{\mathrm{g}}}{\rho}\cdot V_{\mathrm{x}i}\cdot|\mathbf{V}-\mathbf{U}|_i\,, \tag{1}$$

$$\frac{\mathrm{d}V_{zi}}{\mathrm{d}\tau} = -\frac{3}{4}\cdot\frac{C_{\delta i}\psi_i}{\delta_i}\cdot\frac{\rho_{\mathrm{g}}}{\rho}\cdot\left(V_{\mathrm{z}i}-U_{\mathrm{z}}\right)_i\cdot|\mathbf{V}-\mathbf{U}|_i+g\,, \tag{2}$$

$$\frac{\mathrm{d}x_i}{\mathrm{d}\tau} = V_{\mathrm{x}i}\,, \tag{3}$$

$$\frac{\mathrm{d}z_i}{\mathrm{d}\tau} = V_{\mathrm{z}\,i}\,, \tag{4}$$

$$\frac{\mathrm{d}\delta_i}{\mathrm{d}\tau} = -2\frac{D}{\delta_i\rho}\left(\rho_{\mathrm{s}i}^{\vartheta} - \rho_{\mathrm{v}}\right)\mathrm{Sh}_i\,, \tag{5}$$

$$\frac{\mathrm{d}\vartheta_i}{\mathrm{d}\tau} = -\frac{6}{\delta_i^2\rho C_{\mathrm{P}}}\left[\lambda_{\mathrm{g}}\left(\vartheta_i - t\right)\mathrm{Nu}_i + L_i D\left(\rho_{\mathrm{s}i}^{\vartheta} - \rho_{\mathrm{v}}\right)\mathrm{Sh}_i\right], \tag{6}$$

$$\frac{\mathrm{d}t}{\mathrm{d}z} = -\frac{S}{G_{\mathrm{g}}C_{\mathrm{Pg}}}\sum_i f_i\left[\alpha_i\left(\vartheta_i - t\right) + \beta_i\left(\rho_{\mathrm{s}\,i}^{\vartheta} - \rho_{\mathrm{v}}\right)C_{\mathrm{Pv}}\left(\vartheta_i - t\right)\right], \tag{7}$$

$$\frac{\mathrm{d}\rho_{\mathrm{v}}}{\mathrm{d}z} = -\frac{\rho_{\mathrm{g}}S}{G_{\mathrm{g}}}\sum_i f_i\beta_i\left(\rho_{\mathrm{s}i}^{\vartheta} - \rho_{\mathrm{v}}\right), \tag{8}$$

where $V_{\mathrm{x}i}$, $V_{\mathrm{z}i}$ – projections of the droplet velocity vector on the coordinate axes $xz$, m/s; $U_{\mathrm{z}}$ – projections of the air velocity vector on the coordinate axis $z$, m/s; $\vartheta_i$ – temperature (°C) of water droplet with a diameter $\delta_i$ (m); $t$ – air temperature in the cooling tower, °C; $\rho_{\mathrm{v}}$ – absolute humidity (density) of water vapor, kg/m$^3$; $\tau$ – time, s; $S$ – cooling tower irrigation area, m$^2$; $G_{\mathrm{g}}$ – mass air flow rate in the cooling tower, kg/s; $\rho = 1000$ kg/m$^3$ – water density; $L_i = [0.25 - 0.00023(\vartheta_i - 273.2)]\cdot 10^7$ – specific heat of water vaporization, J/kg; $C_{\mathrm{P}} = 4180$ J/(kg·K) – specific heat capacity of water; $C_{\mathrm{Pg}} = 1007$ J/(kg·K) – specific heat capacity of air; $C_{\mathrm{Pv}} = 1970$ J/(kg·K) – specific heat capacity of water vapor; $\lambda_{\mathrm{g}} = (2.9 - 0.78\cdot t)\cdot 10^{-2}$ W/(m·K) – thermal conductivity of air; $\mu_{\mathrm{g}} = \mu_0\frac{273.2 + C_{\mathrm{s}}}{t + 273.2 + C_{\mathrm{s}}}\left(\frac{t + 273.2}{273.2}\right)^{1.5}$ – dynamic viscosity of air, Pa·s, $C_{\mathrm{s}} = 112$, $\mu_0 = 1.71\cdot 10^{-5}$ Pa·s; $D = \frac{0.0805}{P_{\mathrm{a}}}\left(\frac{t + 273.2}{273}\right)^{1.8}\frac{1}{3600}$ – diffusion coefficient (diffusivity), m$^2$/s; $\left|\mathbf{V} - \mathbf{U}\right|_i = \sqrt{V_{\mathrm{x}i}^2 + \left(V_{\mathrm{z}i} - U_{\mathrm{z}}\right)^2}$ – the difference between the droplet and air velocity vectors in the cooling tower, m/s; $P_{\mathrm{a}}$=101325 Pa – ambient air pressure; $C_{\delta i} = \frac{24}{\mathrm{Re}_i} + \frac{4.4}{\sqrt{\mathrm{Re}_i}} + 0.32$ – drag coefficient for droplet; $\mathrm{Re}_i = \frac{\rho_{\mathrm{g}}\delta_i\left|\mathbf{V} - \mathbf{U}\right|_i}{\mu_{\mathrm{g}}}$ – Reynolds number; $\psi_i = \exp(0.03We_i^{1.5})$ – deformation coefficient for droplet in a stream; $We_i = \frac{\rho_{\mathrm{g}}\delta_i\left|\mathbf{V} - \mathbf{U}\right|_i^2}{\sigma_i}$ – Weber number for a droplet, $\sigma_i = (124.11 - 0.17\cdot\vartheta_i)\cdot 10^3$ – surface tension of water, N/m.

Sherwood number as well as Nusselt number are $\mathrm{Sh}_i = \beta_i\cdot\delta_i/D = 2 + 0.55\mathrm{Re}_i^{0.5}\mathrm{Sc}^{0.33}$, $\beta$ – mass transfer coefficient, $\mathrm{Nu}_i = \alpha_i\cdot\delta_i/\lambda_{\mathrm{g}} = 2 + 0.55\mathrm{Re}_i^{0.5}\mathrm{Pr}^{0.33}$ respectively, where Prandtl number $\mathrm{Pr} = \frac{\mu_{\mathrm{g}}C_{\mathrm{Pg}}}{\lambda_{\mathrm{g}}}$, Schmidt number $\mathrm{Sc} = \frac{\mu_{\mathrm{g}}}{D\rho_{\mathrm{g}}}$. The number of droplets per unit volume $n_i = \frac{6}{\pi\delta_i^3}\frac{q}{V_{\mathrm{z}\,i}}$.

Density of the moist air:

$$\rho_{\mathrm{g}} = \frac{PM_{\mathrm{g}}}{R(t + 273.2)}\left(1 - 0.378\frac{\varphi P_{\mathrm{s}}}{P}\right), \tag{9}$$

where $M_{\mathrm{g}} = 0.029$ kg/mol; $R = 8.314$ J/(mol·K) – universal gas constant; $P$ – air pressure in the cooling tower, Pa; $P_{\mathrm{s}}$ – the saturation pressure of water vapor at ambient air temperature, Pa; $\varphi$ – relative humidity of air, %.

Water vapor concentration within the boundary layer of the droplet:

$$\rho_{s\ i}^{\vartheta} = \frac{M_v P_a}{R(\vartheta_i + 273.2)} \exp\left(14.07 - \frac{5221}{\vartheta_i + 273.2}\right). \quad (10)$$

Water vapor concentration in ambient air:

$$\rho_v = \frac{M_v P_s}{R(t + 273.15)} \frac{\varphi}{100}, \quad (11)$$

where $M_v$=0.018 kg/mol.

Initial conditions for the integration of the equations (1)–(8):
at $\tau=0$: $z_i=0$, $x_i=0$, $\delta_i=\delta_{i0}$, $\vartheta_i=\vartheta_{i0}$, $V_{xi}=V_{i0}\cdot\sin(\alpha_t/2)$, $V_{zi}=V_{i0}\cdot\cos(\alpha_t/2)$,
where $\alpha_t$ – full nozzle torch opening angle;
at $z=H$: $\rho_v=\rho_{va}$, $t=t_a$,
where $\rho_{va}$, $t_a$ – humidity and air temperature.

Moist air velocity for arbitrary cross-section of the cooling tower:

$$U = U_d \frac{S_d}{S}, \quad (12)$$

where $U_d$ – air velocity (m/s) in a diffuser with an area $S_d$ (m).

Moist air velocity at the diffuser's throat:

$$U_d = \sqrt{\frac{2\Delta P}{\zeta \rho_g}}, \quad (13)$$

where $\zeta$ – total coefficient of aerodynamic resistance of the cooling tower; $\Delta P$ – pressure drop, which is determined by the aerodynamic characteristics of the cooling tower fan.

The summation of the corresponding components for droplets of all calculated diameters, that are formed by the nozzles, occurs in equations (7)-(8).

Minor loss coefficient for cooling tower to the dynamic pressure at the diffuser's throat ratio:

$$\zeta = \sum_i \zeta_i, \quad (14)$$

where minor loss coefficients $\zeta_i$ are defined by [3]. For droplets in cross-flow air [3]:

$$\zeta_{04} = (0.1 + 0.025 \cdot 3600q)\frac{a}{6}, \quad (15)$$

For droplets in counter-flow air [6]:

$$\zeta_{05} = \frac{3}{2} C_\delta \frac{qH}{V_z \delta}\left(1 + \frac{V_z}{U_z}\right)^2. \quad (16)$$

## 3. Results and discussion

Industrial tests of spray-free cooling towers of standard design 901-6-19 using nozzles with a capacity of ~ 10 m$^3$/h with an average Sauter drop diameter of 3.2 mm, as well as other spray cooling towers, confirm the results of calculations according to the mathematical model [6].

Mathematical modeling allows to determine the thermal characteristic of a spray cooling tower with known initial droplet flow parameters, heat load and atmospheric air parameters. The droplet size distribution function was divided into 10 intervals, choosing the average diameter for each. The

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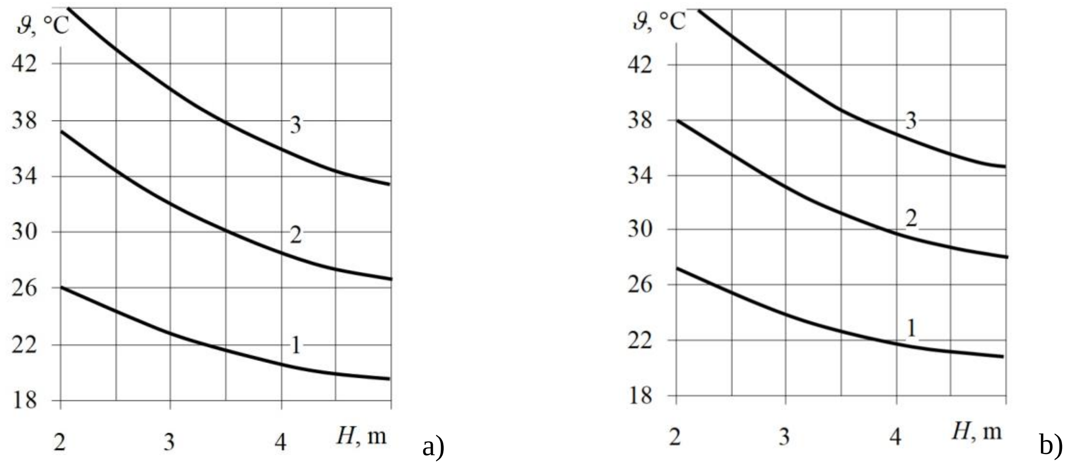
calculation results at initial droplet velocity $V_0 = 4.7$ m/s, torch angle $\alpha_t = 120°$, water temperature difference $\Delta\vartheta = 15$ °C, air velocity in the cooling tower $U_z = 3$ m/s are shown in figures 2-3. From the graphs it follows that the cooling capacity of a spray fan cooling tower significantly depends on the dispersion of the atomization and the height of the nozzles. Reducing the average diameter of the droplet flow to less than 2 mm and increasing the height of the nozzles $H > 4...5$ m (figure 2), allows to significantly reduce the outlet water temperature, achieving and exceeding the cooling performance of known cooling towers with a nozzle.

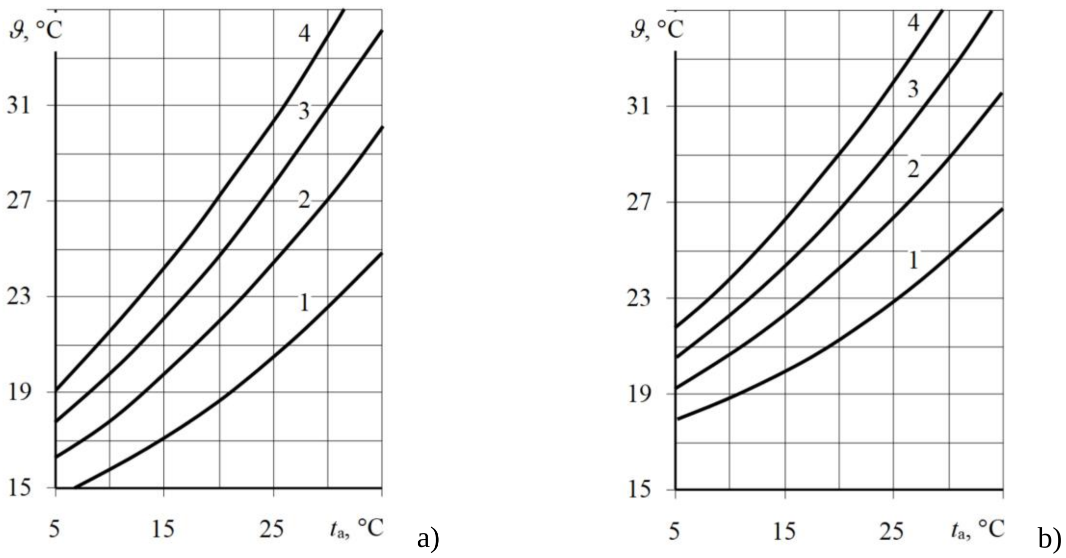
The mathematical model allows to calculate the cooling capacity of the cooling tower (figure 3) and determine the necessary individual parameters of the droplet flow for each type of equipment and consumers for which the specific energy costs for cooling will be minimal.




$q = 5$ m$^3$/(m$^2$h) $q = 7$ m$^3$/(m$^2$h)

1 – $\delta = 1$ mm, 2 – $\delta = 2$ mm, 3 – $\delta = 3$ mm

**Figure 2**. Cooling tower outlet water temperature.




$q = 5$ m$^3$/(m$^2$h) $q = 7$ m$^3$/(m$^2$h)

1 – $\varphi = 30$ %, 2 – $\varphi = 50$ %, 3 – $\varphi = 70$ %, 4 – $\varphi = 90$ %

**Figure** 3. Cooling tower outlet water temperature.

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## 4. Conclusions

The mathematical model of the heat and mass transfer process in a spray cooling tower, which takes into account the polydispersity of the droplet flow and allows to calculate the cooling capacity of the cooling tower, was built in the work.

Reducing the average diameter of the droplet flow to less than 2 mm and increasing the height of the nozzles $H > 4...5$ m, makes it possible to achieve the cooling performance of known cooling towers with a nozzle.

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