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Simulation of a cross-flow cooling tower performance

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Abstract

The new two-dimensional mathematical model of the performance of a cross-flow cooling tower is presented. Our model includes a positive feedback between aerodynamics of cooling tower and a rate of evaporative cooling. The self-consistent iterative algorithm of its solution is proposed and discussed. The simulation results, which included profiles of air temperature at the rain zone, are displayed. It is shown that the main parameter, affecting on the thermal efficiency of cross-flow cooling tower, is the average droplet radius. The range of change of final droplet temperatures is calculated.

1. Introduction

Cooling towers are huge and important parts of modern power plants [1] with water flow rates 30 000 tons per hour and more. Therefore better understanding of their performance is the goal of many engineers and researchers at the field of heat and mass transfer [2–7]. Evaporative cooling of water in the cooling tower depends on atmospheric conditions (temperature, humidity and wind conditions), design and geometric parameters of the tower, and total mass flow rate of water [7–10]. High accuracy simulation of cooling tower performance can help correctly to choose many parameters of cooling tower. Additionally, the simulation of cooling tower performance helps significantly reduce its long and expensive testing at variable atmospheric conditions.

New approach to simulation performance of several types of cooling towers was developed in our works [7–9]. In particular, we applied this approach to a natural draft cooling tower with the pack [7], to the simulation of performance of mechanical draft cooling towers [8,9]. We give here the expansion of our approach to simulation of the performance of natural draft cross-flow cooling towers [1,2,11].

We discuss the new advanced mathematical model of natural draft cross-flow cooling tower performance, the iterative algorithm of its solution and the results of its numerical simulation. It is worth to mention that the simulation of cross-flow cooling towers is most difficult task in compare with considered before ones because we have to combine two-dimensional description of heat and mass transfer in air flow with falling water droplets. The sketch of the cross-flow cooling tower is shown in Fig. 1. We assume that a wind velocity is smaller than the inlet velocity of air [10], so we can use the cylindrical symmetry for the description of the air flow.

Our mathematical model is the boundary value problem for two partial differential equations (PDE) and the coupled system of ordinary differential equations (ODE). The system of PDE describes large-scale turbulent heat transfer and water vapor transfer of diffusion type at vertical direction in the rain zone of the cooling tower. The system of ODE describes the process of evaporative cooling of the monodisperse ensemble of water droplets, which falling in cross-flow of air. Limits of use of monodisperse approximation for description of an ensemble of droplets were investigated at our paper [8].

We substantially use our earlier results [7–9], verified with experimental ones [10]. Following our general approach we describe the aerodynamics of a cooling tower at the integral approximation. Air flow is turbulent one

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Nomenclature

C	coefficient of aerodynamic drag, dimensionless
c	specific heat (J kg^{-1} K^{-1})
D	diffusion coefficient (m^2/s)
g	gravity (m/s^2)
H	height (m)
h	height of air window (m)
I_n	diffusion source
I_t	heat source
K	coefficient, dimensionless
L	length of air window (m)
l	droplet trajectory (m)
m	mass of droplet (kg)
N	the number of subzones
N_d	number of droplets (m^{-3})
n	the number density of droplets (m^{-3})
Q_a	specific air mass flow rate (kg/m^2 s)
Q_w	specific water mass flow rate (kg/m^2 s)
R	inlet cooling tower radius (m)
R_0	outlet cooling tower radius (m)
R_d	droplet radius (m)
r	coordinate in horizontal direction (m)
T	temperature (°C)
T_{pool}	water temperature in the pool (°C)
U_a	velocity of air (m/s)
U	latent heat of phase transitions (J kg^{-1})
u_0	inlet air velocity (m/s)
U_{or}	outlet air velocity (m/s)
z	coordinate in vertical direction (m)
Re	Reynolds number for droplet
Re_h	Reynolds number for air window

Greek symbols

α	heat transfer coefficient (W m^{-2} K^{-1})
Δ	difference
γ	mass transfer coefficient (m s^{-1})
η	thermal efficiency, dimensionless
λ_a	thermal conductivity of air (W/m °C)
μ_a	dynamic viscosity of air (kg m^{-1} s^{-1})
ρ	density (kg/m^3)
υ	droplet velocity (m/s)
Ψ	relative humidity of air, dimensionless

Subscripts

0	initial
a	air
d	droplet
lim	limiting
m	molecular
s	saturated
w	water

therefore we use the effective values for the diffusion coefficient of water vapor in air and for the heat conductivity coefficient of moist air, which are based on experimental results [12].

To obtain the solution of our non-linear mathematical model we use the iterative procedure. Usually it takes only three iterations for the convergence of the heat and aerodynamic parts of the solutions.

For description of cooling tower performance we use effective dimensionless parameter, so called the thermal efficiency of cooling tower η [1,7,8]:

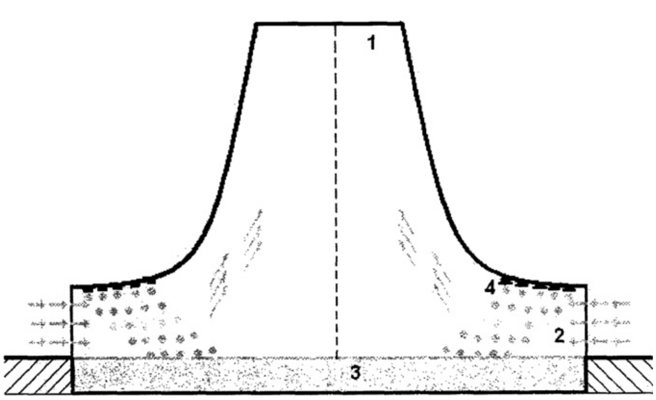

Fig. 1. The sketch of the cross-flow cooling tower. 1 is a cooling tower orifice; 2 is a cooling tower window: 3 is a water collecting pond; 4 are cooling tower nozzles.

$$\eta = \frac{T_{w_0} - T_{pool}}{T_{w_0} - T_{lim}}, \tag{1}$$

where T_{w_0} is the inlet water temperature, T_{pool} is the averaged temperature of water in the cooling tower pool, T_{lim} is the limiting temperature of evaporative cooling, which is determined from the equation:

$$\rho_s(T_a) \cdot \psi = \rho_s(T_{lim}).$$

It is worth to note that the limiting temperature of evaporative cooling T_{lim} is equal to the temperature of the wet-bulb thermometer, and it is important that the thermal efficiency of cooling tower η practically does not depend on the humidity surrounding cooling tower air.

Some preliminary results of our simulation have been presented at [13].

2. Mathematical model of cross-flow cooling tower

The system of PDE, describing heat and mass transfer in the rain zone of the cross flow cooling tower includes the equations for the temperature field of gaseous mixture and for the filed of the number density of water vapor. The equation for the temperature field $T_a(r,z)$ of the moist air is

$$u_a \frac{\partial T_a}{\partial r} = \frac{\lambda_a}{\rho_a c_a} \frac{\partial^2 T_a}{\partial z^2} + I_t, \tag{2}$$

where u_a is the average velocity of air flow, T_a is a temperature of the vapor–air mixture, λ_a, ρ_a, and c_a are, correspondingly, the coefficients of the heat conductivity of air, density of the vapor–air mixture and specific thermal capacity of air; r is the radial coordinate, z is the vertical coordinate, I_t is the heat source.

For moist air the efficient (turbulent) heat conductivity λ_a, based on experimental data, is expressed as [12]

$$\lambda_a = 0.04\lambda_{am}Re_h^{0.8},$$

where λ_{am} is the molecular heat conductivity coefficient of the moist air, the Reynolds number Re_h for air flow is

$$Re_h = \frac{\rho_s h u_s}{\mu_s},$$

where h is the height of inlet window of the cooling tower.

The equation for the number density of water vapor $n(r,z)$ in the rain zone of cooling tower is

$$u_a\frac{\partial n}{\partial r} = D\frac{\partial^2 n}{\partial z^2} + I_n, \tag{3}$$

where D is the effective (turbulent) coefficient of diffusion, I_n is a diffusion source.

For moist air the Lewis number Le has the constant value about one for all the Reynolds numbers [12], where

$$\mathrm{Le} = \frac{\lambda}{\rho c D}.$$

Therefore for self-consistency we accept that the effective diffusion coefficient D is determined by the following expression, which has the same structure as the expression for the turbulent heat conductivity:

$$D = 0.04 D_m Re_h^{0.8},$$

where D_m is the molecular diffusion coefficient of water vapor in air.

We have two Reynolds numbers at our model. For droplets the Reynolds number Re is defined as follows:

$$Re = \frac{2\rho_a R_d v_1}{\mu_a},$$

where ρ_a is a density of air, R_d is a droplet radius, μ_d, is air viscosity, v_1 is the absolute velocity of falling droplets which is defined by the expression:

$$v_1 = \left((v_r - u_a)^2 + v_z^2\right)^{0.5},$$

where v_r and v_z are radial and vertical components of the droplet velocity.

The equation for the droplet growth, based on mass conservation law for droplet, is as follows [8]:

$$\frac{dR_d}{dl} = -\frac{\gamma(Re)(\rho_s - \rho)}{\rho_w v_1}, \tag{4}$$

where dl is the element of a droplet trajectory, ρ and ρ_s are mass densities of the vapor and saturated vapor at given temperature, ρ_w is a mass density of liquid water, $\gamma(Re)$ is the mass transfer coefficient, depending on the Reynolds number:

$$\gamma(Re) = \frac{D_m(2 + 0.5Re^{0.5})}{2R_d}$$

The equation for the changing of the temperature of falling droplets along their trajectory is

$$\frac{\partial T_w}{\partial l} = \frac{3[\alpha(Re)(T_a - T_w) + \gamma(Re)(U - c_w T_w)(\rho_s - \rho)]}{c_w\rho_w R_d v_1}, \tag{5}$$

where T_w is a temperature of the water droplets, U is the specific latent heat of the evaporation of water, c_w is the specific heat capacity of water, $\alpha(Re)$ is the heat transfer coefficient, depending on the Reynolds number:

$$\alpha(Re) = \frac{\lambda_{am}(2 + 0.5Re^{0.5})}{2R_d}.$$

The equation for change of droplet velocity along the trajectory is

$$\frac{dv_1}{dl} = \frac{g}{v_1} - C(Re)\frac{\rho_a\left(v_z^2 + (v_r - u_a)^2\right)^{0.5}}{2v_1}\frac{\pi R_d^2}{m}, \tag{6}$$

where m is the mass of the droplet, $C(Re)$ is the drag coefficient [8].

At the cylindrical systems of coordinates, related with the symmetry axes of the cooling tower, the equation for change of droplet velocity along radial direction is

$$\frac{dv_1}{dr} = -C(Re)\frac{\rho_a(v_r - u_a)}{2}\frac{\pi R_d^2}{m}. \tag{6'}$$

The equation for change of droplet velocity along vertical coordinate z is

$$\frac{dv_1}{dz} = \frac{g}{v_1} - C(Re)\frac{\rho_a v_z}{2}\frac{\pi R_d^2}{m}. \tag{6''}$$

The boundary conditions for the system of the differential equations (2)–(6) are: for the droplets at $z = h$ we have the following values:

inlet radiuses:

$$R_d|_{z=h} = R_{d0}, \tag{7}$$

inlet water temperature:

$$T_w|_{z=h} = T_{w_0}, \tag{8}$$

inlet velocity of droplets:

$$v|_{z=h} = v_0. \tag{9}$$

For gaseous mixture the boundary conditions are: for $z = h$ we have the condition that air temperature is equal to inlet water temperature:

$$T_a|_{z=h} = T_{w_0}, \tag{10}$$

the condition that the density of water vapor is equal to the density of saturated vapor at the inlet water temperature:

$$\rho|_{z=h} = \rho_s(T_{w_0}). \tag{11}$$

where ρ_s is the density of saturated water vapor for given temperature. The boundary conditions Eqs. (10) and (11) are valid only for high enough of water flow rate. It is worth to emphasize that their use simplifies calculations significantly. More rigid consideration of this problem will be presented at another our publication.

For $z = 0$ we have

$$T_a|_{z=0} = T_{wp}, \tag{12}$$

where T_{wp} is a surface temperature of water in the pool,

$$\rho|_{z=0} = \rho_s(T_{wp}). \tag{13}$$

Inlet air is characterized by the temperature T_0 and by the relative humidity Ψ:

$$T = T_0, \tag{14}$$

$$\Psi = \Psi_0, \tag{15}$$

where Ψ_0 is the relative humidity of air, measured at the edge of inlet window. For simulation we convert relative humidity to the corresponding the number density of water vapor.

We describe a field of air velocity by means of average value, which is determined in the approximation of one-dimensional description of cooling tower aerodynamics. Important detail of aerodynamics of a cross-flow cooling tower is that the converging character of air flow leads to increasing average velocity $u(r)$ of moist air. This effect intensifies the heat and mass transfer processes during evaporative cooling. We have the formula for average velocity:

$$u(r) = u_0 \frac{R}{r}, \tag{16}$$

where u_0 is a initial air velocity, R is the radius of the inlet windows of the cross-flow cooling tower (see Fig. 1).

The thermal source for Eq. (2) is

$$I_t = 4\pi R_d^2 N_d \cdot \alpha(Re)(T_w - T_a),$$

where N_d is the number of droplets per unit of volume, depending on droplet velocities and their initial density.

The source term at Eq. (3) is

$$I_n = 4\pi R_d^2 N_d \cdot \gamma(Re)(n - n_s),$$

where n and n_s are, correspondingly, the number density of water vapor and the number density of the saturated vapor for given temperature.

We neglect the processes of breaking and coagulation of droplets; therefore there is the law for the conservation of the number of droplets along the trajectory:

$$N_0 \upsilon_0 = N_d \upsilon_d, \tag{17}$$

where N_0 and υ_0 are, correspondingly, initial number of droplets per unit of volume and droplet velocity, N_d is a number of droplets per unit of volume at the arbitrary point of flow, where droplet velocity is equal to υ_d. It follows from the expression (17) that increasing droplet velocity leads to decreasing the number density of droplets. Immediately we have important conclusion that at lower part of the rain zone air flow should be less heated and less moistened.

For the cross-flow cooling tower the outlet velocity of air U_{or} can be calculated using standard formula of the natural convection [7]:

$$u_{or} = \sqrt{gH\frac{\Delta\rho}{\rho_0}} \cdot K, \tag{18}$$

where $\Delta\rho$ is a change of gaseous mixture mass density, ρ_0 is the initial density of air at surrounding atmosphere, g is the gravity, H is the height of the cooling tower, K is empirical coefficient. For natural draft cooling tower with the pack K is about 0.5 [7]; for cross-flow natural draft cooling tower we use at simulation the value of $K = 0.8$. We expect that hydraulic resistance of cross-flow cooling tower is smaller than one of standard natural draft cooling tower [2]. We receive the expression fort the average inlet velocity of air u_0, using the integral form of the continuity equation and the expression (18). For the first iteration we apply the approximate expression for change of the density of gas flow $\Delta\rho = \rho_0\beta\Delta T$:

$$u_0 = \frac{R_0^2}{2Rh} \cdot \sqrt{gH\beta\Delta T} \cdot K, \tag{19}$$

for higher iterations we use more exact expression,

$$u_0 = \frac{R_0^2}{2Rh} \cdot \sqrt{gH\frac{\Delta\rho}{\rho_0}} \cdot K, \tag{20}$$

where R_0 is the orifice radius, R is the radius of cooling tower pool, the change of air density $\Delta\rho$ includes the change of its humidity. Our simulation shows that cross-flow cooling tower thermal efficiency η has quite weak dependence on the parameter K. Thus the expression (20) collects contributions of basic geometric parameters of the cooling tower to the value of inlet velocity of gas flow.

Before numerical simulation let us compare the contributions of different terms at right-hand side of Eqs. (2) and (3). We can neglect mass transfer of diffusion type at Eq. (3) if the following inequality is valid [14]:

$$\frac{4\pi R_d D_m h^2}{D} N_d \gg 1. \tag{21}$$

Correspondingly, for Eq. (2) we can neglect heat transfer of diffusion type if the following inequality is valid:

$$\frac{4\pi R_d \lambda_{am} h^2}{\lambda_a} \rho_a c_a N_d \gg 1. \tag{22}$$

To mention that laminar flow with vapor condensation on droplets have been considered at our paper [15].

2.1. Iteration procedure

The specific of any natural draft cooling tower is a positive feedback between evaporative cooling and averaged velocity of air flow. In order to overcome this trouble we use the self-consistent iterative procedure.

For the first iteration we suppose that the final temperature of moist air at the outlet of the rain zone is equal to temperature of inlet water. Using this assumption inlet velocity u_0 is calculated using expression (19). Also we consider that the temperature and humidity on interfacial border of the cooling tower pool are equal to corresponding inlet parameters. After these assumptions we solve the boundary problem (2)–(6). For solution of boundary problem we divide the irrigation zone on arbitrary number of subzones, each of them have the same length at the radial direction of cooling tower. For each subzone we make subsequent calculations: final results of the calculation of air parameters from previous subzone are the entry conditions of the subsequent subzone. Also we take into account an important circumstance that the air flow has the converging character in the rain zone of cooling tower. According to our algorithm, first of all we solve the equations for water droplets in our model with averaged parameters of air flow, and then results of this calculation we use for reckoning of source terms in Eqs. (2) and (3). In agreement with our previous results [8], evaporative of cooling of droplets take place mainly in upper part of rain zone where droplets velocity is relative small. It is obvious that if we have a larger number of such subzones the more accurate is the description of heat and mass transfer processes in the rain zone of cooling tower.

As a result of numerical solution of mathematical model we have the following parameters: average final temperature of the droplets, average temperature of moist air and the average the number density of water vapor. Using these data we recalculate average velocity of a vapor–air stream using the expression (20). Then we again solve the boundary problem (second iteration). For the second iteration we put correct calculated value of the surface temperature of the pool, which is equal to the averaged final temperature of droplets. We solve again the boundary problem with new boundary conditions and new inlet velocity. Usually we need only three iterations for convergence calculations of the heat efficiency of evaporative cooling.

3. Simulation results

We present simulation results for the cross-flow cooling tower with following parameters: height of the cooling tower $H = 80$ m, inlet cooling tower radius $R = 40$ m, outlet radius $R_0 = 20$ m, height $h = 10$ m and length $L = 10$ m of the cooling tower window, $K = 0.8$. For these parameters our mathematical model gives that the inlet velocity of air is about 2.6 m/s, so we assume the velocity of a ground wind is smaller than this value.

For all results below we assume that initial air temperature is 25 °C, and relative humidity is equal to 60%. The influence of atmospheric condition was investigated carefully at our previous publications [7–10]; therefore below we conduct simulation for the same atmospheric conditions.

It is worth to note that for one millimeter size droplets and parameters of the cooling tower, described above, right-hand sides of inequalities (21) and (22) are larger than 10^3 and contributions of turbulent heat and mass transfer are weak in compare with direct impact of evaporative cooling of droplets.

We see the change of droplets radius in Fig. 2. Curves 1 and 2 show the evaporation droplet versus its vertical position at the inlet of rain zone. The initial droplet radius was used as the scale at the transition to dimensionless values.

The change of temperature of falling droplet is shown in Fig. 3. Curves 1 and 2 describe droplets at the inlet of the rain zone; curves three and four describe droplets at the outlet of the rain zone. We clear see two important effects: effect of droplet size and effect of position of evaporative cooling at the rain zone. Indeed, qualitative estimation, based on the Eq. (5) gives for the temperature difference of droplet during evaporative cooling ΔT:

$$\Delta T \sim R_{\mathrm{d}}^{-3/2},$$

(see also [8]). The numerical results, presented in Fig. 3, confirm this conclusion. The minimal droplets radius is limited by the spray design, water pressure at supplying system and conditions that all droplets will fall to collecting pond of cooling tower. Conditions for evaporative cooling of droplets differ significantly in the rain zone of cross-flow cooling tower. The main reason is heating air at top part of rain zone and increasing of its humidity.

For three different heights temperatures of moist air at the rain zone are shown in Fig. 4. We presented here the results obtained for droplets with initial radius 1 mm. The fast heating of upper part of air flow, where the temperature and residence time of droplets are much larger than at the bottom, is obvious.

Also we present the change of the number density of water vapor at the rain zone of the cooling tower. For three different heights at the rain zone the data are plotted in Fig. 5. The value of the inlet number density of water vapor is used as the scale for transition to dimensionless variable. Behavior of air temperature and of the number density of water vapor at the lower part of the rain zone is the

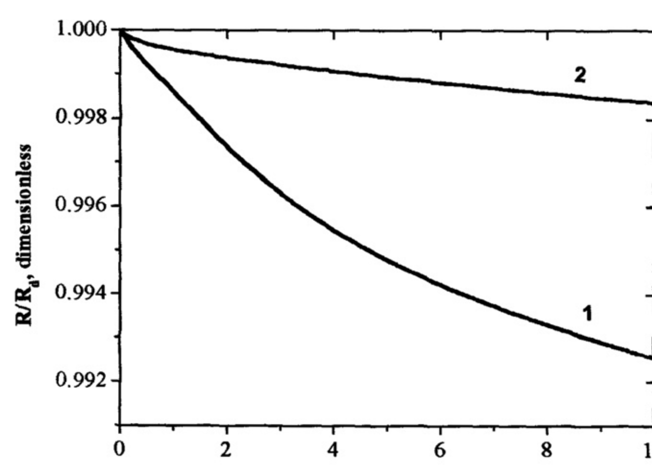

Fig. 2. Change of dimensionless droplets radius along vertical position. Curve 1 is for $R_{\mathrm{d}} = 1$ mm; curve 2 is for $R_{\mathrm{d}} = 1.5$ mm (first zone).

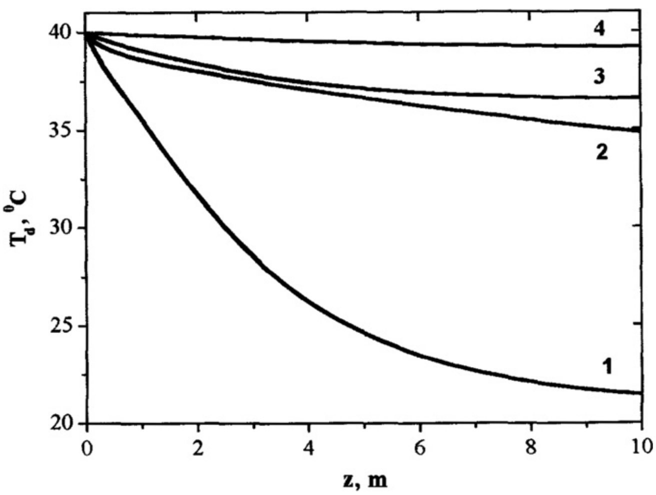

Fig. 3. Change of droplets temperature T_d along vertical position. Curve 1 is for $R_d = 1$ mm; curve 2 is for $R_d = 1.5$ mm (first zone).

manifestation of decreasing of contribution from falling droplets. The main physical reason is that the increasing of droplet velocity leads to decreasing of the number density of droplets, according to the continuity equation. The second factor is decreasing of residence time according to Eq. (4). Increasing of transfer coefficients does not compensate these factors. The simulation data are obtained for initial droplet radius 1 mm. To note that increasing of droplet radius leads to decreasing of steepness of all profiles for air.

The thermal efficiency η is the most interesting parameter at our problem from engineering point of view. The averaged temperature T_{pool}, entering into expression (1), is calculated by the formula

$$T_{\text{pool}} = \frac{\sum_{i=1}^{N} T_{wi} \cdot Q_{wi}}{\sum_{i=1}^{N} Q_{wi}}, \tag{23}$$

where T_{wi} is the final temperature of droplet from the ith subzone of the irrigation system of the cooling tower, Q_{wi}

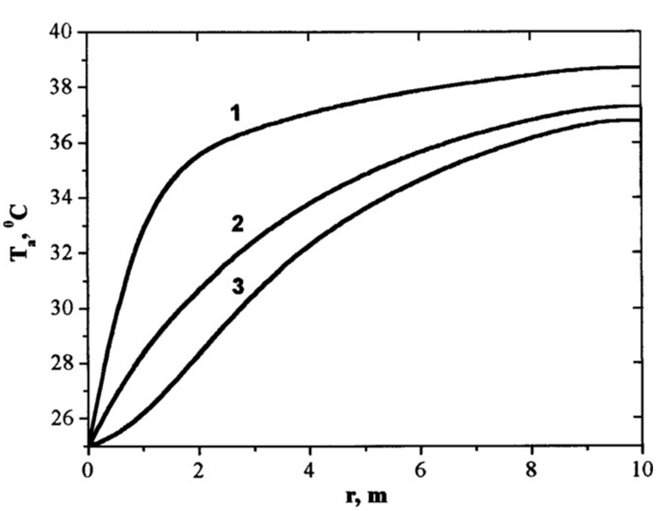

Fig. 4. Air temperature T_a along the rain zone. Curve 1 is for $h = 1$ m; curve 2 is for $h = 4$ m; curve 3 is for $h = 7$ m.

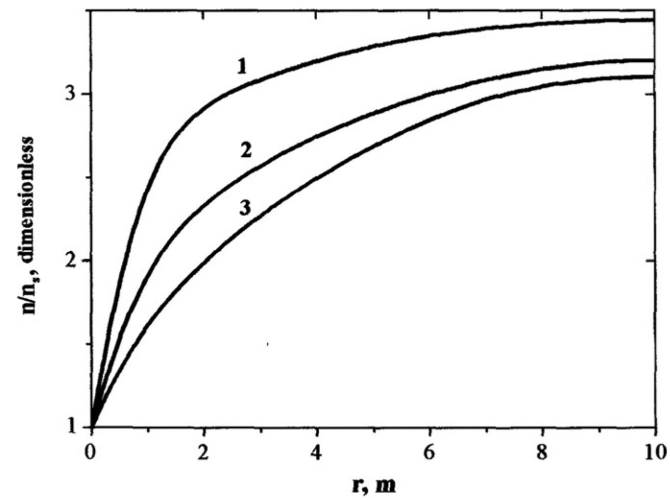

Fig. 5. Dimensionless number density of water vapor along the rain zone. Curve 1 is for $h = 1$ m; curve 2 is for $h = 4$ m; curve 3 is for $h = 7$ m.

is a relative mass flow rate through ith subzone of the irrigation system, where N is the number of subzones of the irrigation zone. All numerical results are presented here for $N = 5$.

The dependence of η versus the ratio of mass flow rates of water and air Q_w/Q_a is shown in Fig. 6. For the case of evaporative cooling of droplets with radius 1 mm (curve 1), there is practically no dependence on Q_w/Q_a. The explanation of this simulation results is based on two our observations. At first, it follows from Figs. 2–5; the evaporation rate is high enough for small droplets. Therefore air temperature and humidity reach their final values quite quickly at the upper part of the rain zone. At second, there is a great variation of final droplet temperature from different part of the rain zone (see Fig. 8). Indeed, the evaporation of droplet is much less effective except of inlet zone because they evaporate when they have big falling velocity. Therefore for small droplets if we increase the parameter Q_w/Q_a we simply change not

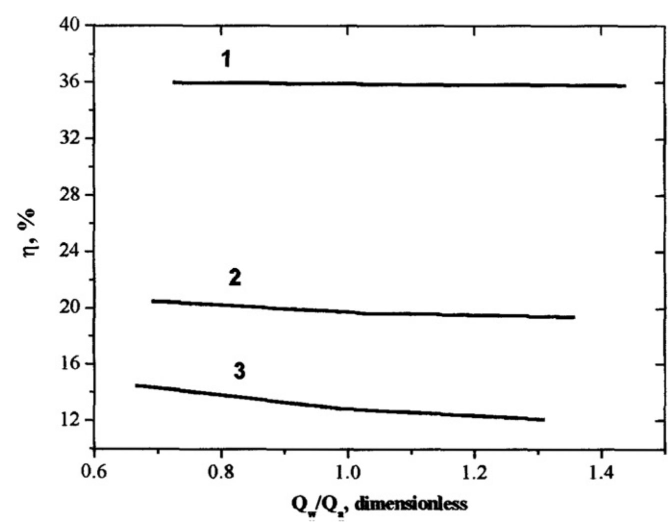

Fig. 6. Thermal efficiency η versus mass flow rates ratio Q_w/Q_a. Curve 1 is for $R_d = 1$ mm; curve 2 is for $R_d = 1.5$ mm.

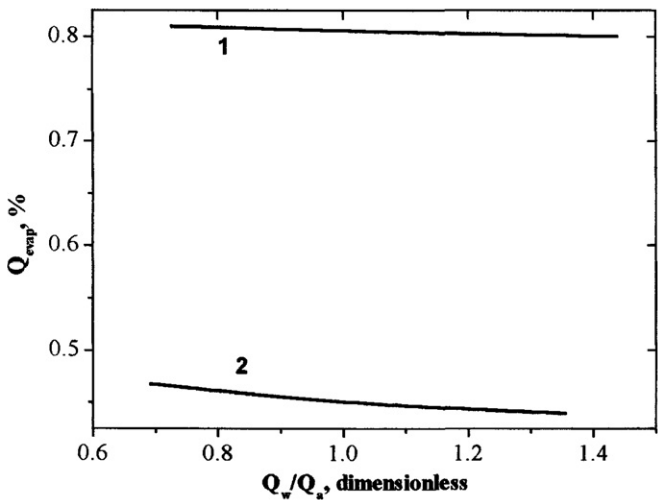

Fig. 7. Dimensionless flow rate of the evaporated water Q_{evap} versus mass flow rates ratio $Q_{\mathrm{w}}/Q_{\mathrm{a}}$. Curve 1 is for $R_{\mathrm{d}} = 1$ mm; curve 2 is for $R_{\mathrm{d}} = 1.5$ mm.

average final droplet temperature but increased the range of its values. Average final temperature is mainly determined by the evaporative cooling of droplets in the inlet zone. For larger droplets when evaporation rate is not so high we see standard dependence of η versus $Q_{\mathrm{w}}/Q_{\mathrm{a}}$ [7–10].

The second important engineering parameter is the mass flow rate Q_{evap} of evaporated water. Dimensionless flow rate of the evaporated water versus the ratio between the mass flow rates of water and air $Q_{\mathrm{w}}/Q_{\mathrm{a}}$ is shown in Fig. 7. The mass flow rate of water is used as the scale. We see that mass flow rate of evaporated water is smaller than one percent of mass water flow rate. If we decrease water flow rate, the mass flow rate of evaporated water slightly increases due to more favorable conditions for evaporative cooling.

To remind that we divided irrigation zone on several subzones with different flow rates, which are determined by the geometrical consideration. Let us introduce the temperature dispersion of the final temperature of falling droplet in the collecting pond of cooling tower:

$$\mathrm{D}T = \frac{\sum_{i=1}^{N} (T_{\mathrm{w}i} - T_{\mathrm{pool}})^2 \cdot Q_{\mathrm{w}i}}{\sum_{i=1}^{N} Q_{\mathrm{w}i}} \tag{24}$$

Variations of water temperature in the pond versus $Q_{\mathrm{w}}/Q_{\mathrm{a}}$ are shown in Fig. 8. We see that the conditions for evaporative cooling are worse at larger values of the ratio $Q_{\mathrm{w}}/Q_{\mathrm{a}}$, difference between final droplet temperatures at inlet zone and another parts of the rain zone of cooling tower increases. For larger droplets the dispersion is smaller.

4. Conclusions

The new advanced mathematical model of natural draft cross-flow cooling tower, which includes partial differential equations and ordinary ones, is presented. The specific of cooling towers of such type is two-dimensional character of heat and mass transfer processes in the rain zone. The characteristic spatial scale of the evaporative cooling is about the droplet radius R_{d}, the characteristic spatial scale of the aerodynamic process is about the height of inlet window H. Typical estimation of the ratio H/R_{d} is about 10^4. Therefore we use detailed description of evaporative cooling of falling droplets and only integral description of convective flow of moist air in the cooling tower. We took into account the converging character of air flow also.

The iterative procedure of finding the solution of our non-linear mathematical model is developed. In particular, the irrigation zone of the cooling tower was divided on five subzones of the same length at radial direction.

For weak winds our calculations shown that the dimensionless integral parameter the thermal efficiency of cooling tower η is mainly determined by the initial droplet radius if ratio of flow rates $Q_{\mathrm{w}}/Q_{\mathrm{a}}$ is about one. This conclusion agrees well with our previous result [7]. The mass flow rate of evaporated water is about 1% of total mass water flowrate. Two-dimensional nature of heat and mass transfer

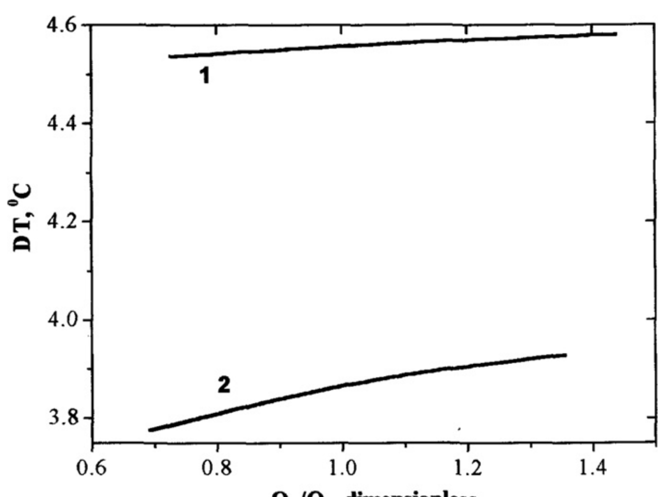

Fig. 8. Dispersion of the final droplets temperature DT versus mass flow rates ratio $Q_{\mathrm{w}}/Q_{\mathrm{a}}$. Curve 1 is for $R_{\mathrm{d}} = 1$ mm; curve 2 is for $R_{\mathrm{d}} = 1.5$ mm.

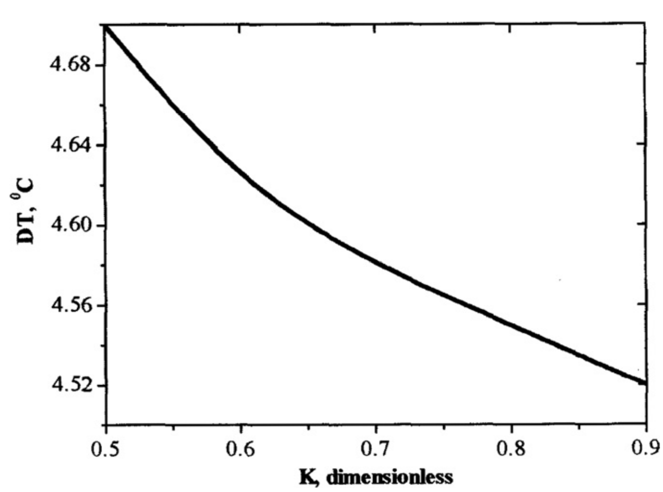

Fig. 9. Dispersion of the final droplets temperature DT versus parameter K.

processes leads to a broad range of final droplet temperatures, depending on their initial position and related with heating of air flow and its increased humidity. Using our iterative procedure we introduced the dispersion of final water droplet temperature. As a measure of water fluctuations in the water-collecting pond of the cooling tower, we have calculated the dispersion of water temperature $\mathrm{D}T$ (Fig. 9). It worth to note, that we plotted the square root of the dispersion $\mathrm{D}T$. It will be interesting to make measurements of the dispersion of water temperature on surface of collecting pond and compare with our calculations.

Majority of simulation results above have been obtained for $K = 0.8$. Calculation leads to conclusion that the change of parameter K at the range 0.5–0.9 lead to the change of thermal efficiency of cooling tower η is only about several percentages.

In accordance with our qualitative picture of evaporative cooling at cross-flow cooling tower, if we increase flow rate of air (increasing K), we decrease the dispersion of temperature.

We hope that presented simulation results will help engineers to increase efficiency of cross-flow cooling tower, optimizing its parameters and its design. In particular, to find the way to introduce slow water droplets at the lower part of air flow.

References

[1] L.D. Berman, Evaporative Cooling of Circulating Water, Pergamon, London, 1961, p. 710.

[2] J.R. Singham, Cooling towers, in: G.F. Hewitt, G.L. Shires, Y.V. Polezhaev (Eds.), International Encyclopedia of Heat Mass Transfer, CRC Press, Boca Raton, 1997.

[3] J.C. Kloppers, D.G. Kröger, A critical investigation into the heat and mass transfer analysis of counterflow wet-cooling towers, Int. J. Heat Mass Transfer 48 (2005) 765–777.

[4] R. Al-Waked, M. Behnia, CFD simulation of wet cooling towers, Appl. Therm. Eng. 26 (2006) 382–395.

[5] P. Naphon, Study on the heat transfer characteristics of an evaporative cooling tower, Int. Commun. Heat Mass Transfer 32 (2005) 1066–1074.

[6] A.S. Kaiser, M. Lucas, A. Viedma, B. Zamora, Numerical model of evaporative cooling processes in a new type of cooling tower, Int. J. Heat Mass Transfer 48 (2005) 986–999.

[7] S.P. Fisenko, A.I. Petruchik, A.D. Solodukhin, Evaporative cooling of water in a natural draft cooling tower, Int. J. Heat Mass Transfer 45 (2002) 4683–4694.

[8] S.P. Fisenko, A.A. Brin, A.I. Petruchik, Evaporative cooling of water in a mechanical draft cooling tower, Int. J. Heat Mass Transfer 47 (2004) 167–177.

[9] S.P. Fisenko, A.I. Petruchik, Towards to the control system of mechanical draft cooling tower of film type, Int. J. Heat Mass Transfer 48 (2005) 31–35.

[10] A.I. Petruchik, A.D. Solodukhin, N.N. Stolovich, S.P. Fisenko, Toward the analysis of experimental data on thermal efficiency of evaporative cooling tower, Appl. Energ.: Russ. J. Fuel Power Heat Systems 37 (6) (2000) 142–149.

[11] H. Inazumi, S. Kageyama, Successive graphical method of design of a cross-flow cooling tower, Chem. Eng. Sci. 30 (7) (1975) 717–721.

[12] A.A. Zhukauskas, High-Performance Single-Phase Heat Exchangers, Hemisphere Publishing Corporation, New York, Washington, Philadelphia, London, 1989.

[13] A.A. Brin, S.P. Fisenko, Simulation of a cross-flow cooling tower performance, 13th International Heat Transfer Conference, Sydney, Australia, 2006.

[14] V.P. Krainov, Qualitative Methods in Physical Kinetics and Hydrodynamics, American Institute of Physics, New York, 1992.

[15] S.P. Fisenko, A.A. Brin, Heat and mass transfer and condensation interference in a laminar flow diffusion chamber, Int. J. Heat Mass Transfer (49) (2006) 1004–1014.