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CALCULATION OF TEMPERATURE OF ROUTINE WATER COOLED IN IRRIGATED LAYERS

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Abstract

The article describes the developed mathematical model, algorithm and program for calculating the process of cooling the water leaving the evaporative cooler and the final temperature of humid air. The compilation of a mathematical model is based on the analysis of literature data. Practically at all industrial enterprises, technological equipment is cooled by means of circulating water supply systems equipped with evaporative coolers. The article made a choice of a cooling system for air conditioning systems of residential premises. The developed basic design scheme of the evaporative water and air cooler with the irrigated layer is presented, as well as the estimated thermal and material balance. One of the main elements of these devices is a heat-mass transfer nozzle - sprinkler. This article presents the results of mathematical modeling of processes occurring in the volume of the sprinkler evaporator chamber, Raschig rings composed of vertical polymeric materials. Expressions are obtained for determining the values of air temperature based on the calculation of thermal modeling of the process of cooling circulating water in evaporative coolers of the type in question.

Key words: *evaporative cooler, fluidized bed, sprinkler, heat and mass transfer, relative humidity, wet thermometer*

A lot of time and high financial costs on the basis of field experiments are required in order to specify the thermo-technical excellence of evaporative coolers of circulating water, including those with a fluidized bed. Another more modern method of solving this problem is a numerical calculation method that takes into account any changes in the parameters of the cooled water , the environment, structural, thermo-technical and operational characteristics of individual elements and components of the cooler. At the present stage of computer technology development, this method allows to obtain and justify more reliable initial data for the development of appropriate engineering solutions to ensure the efficient operation of coolers of this type in various sectors of the energy, processing, petrochemical and food industries. The basis of numerical calculation method for the temperature regime of recycled water cooled in evaporative coolers with a three-phase fluidized bed, as a rule, is mathematical modeling, i.e. compilation and solution of systems of equations of thermal balances for cooled water, evaporative agent (air) and elements of a cooler of the analyzed type.

For the implementation of the evaporative cooling process of circulating water in air conditioning systems, injector and irrigation devices are most widely used [1, 2, 3, 8]. In injector-type devices, the development of surface between air and water is constant and is achieved by spraying water through the injectors; the latter ensures the formation of many small droplets in the injector chambers. The injector chambers, as a rule, have large air capacities and are used in industrial facilities where it is necessary to reduce the temperature and maintain high relative air humidity in such textile enterprises, etc. [3].

The following requirements are imposed on the materials of the irrigation layer in the water and air coolers of this type:

- the greatest development of a constant surface in the same volume of filling;
- the most complete surface wettability or the material itself with water at the lowest energy cost for water supply;
- high efficiency of evaporative cooling with minimal aerodynamic drag;
- the material's resistance against rotting, corrosion, weathering, the harmful odors;
- resistance against infection by bacteria;
- low cost and availability of source material.

Synthetic fibers, wood fibers and chips, fiberglass, palm tree bark fibers, metal wires, plates of porous plastics, thin metal sheets are used to fill the irrigated layers [1, 14].

In our work, the evaporative cooling of water and conditioned air in irrigated layers in which it is permissible to take the parameters of moist air equal to the corresponding parameters of saturated air at the temperature of the irrigated water (t_w) is considered.

Table 1 shows the surface values by the constant of heat and mass transfer in a unit volume of particles of some filling materials of the irrigated layer (a) according to [1, 4].

Filling material and average size of particles, mm	The ratio of the contact surface of heat and mass transfer (M) to unit volume
Nylon fiber with a diameter of 0.25 mm and a length of 200 mm.	16000
Fiberglass with a diameter of 0.135 mm and a length of 200 mm.	29629
Corrugated fiberglass with a diameter of 0.04 mm and a length of 50 mm.	100000
Corrugated aluminum tape 0.225 mm thick, 51 mm wide and 600 mm long.	8900
Brass mesh of corrugated brass wire with a diameter of 0.25 mm and a length of 600 mm.	16000

The local evaporative water and air coolers with irrigated layers of lower air capacity installed in the most served rooms or in their surroundings are more economical and convenient for public and residential buildings [1, 18].

The basic design scheme of an evaporative water and air cooler with an irrigated layer is shown in Fig. 1 [6, 7, 9, 13, 20].

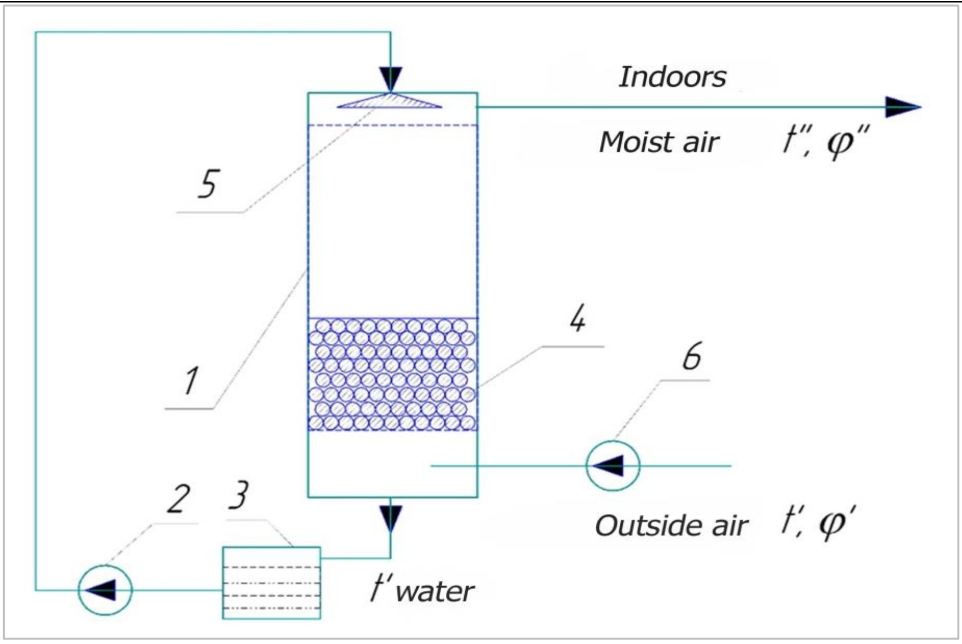

Fig. 1. The basic design scheme of an evaporative water and air cooler with an irrigated layer:

1 - evaporative cooler, 2 - circulation (water) pump, 3 - sump, 4 - injector of the Rashig ring, 5 - injector, 6 - air fan, t_{ah} and φ_{ah} - air temperature and humidity, tw - temperature of irrigating water ('- means at the inlet and "- at the exit of the irrigation chamber)

According to the basic design scheme shown in Fig. 1, the differential equation of stationary heat balance for the considered water and air cooler has the form.

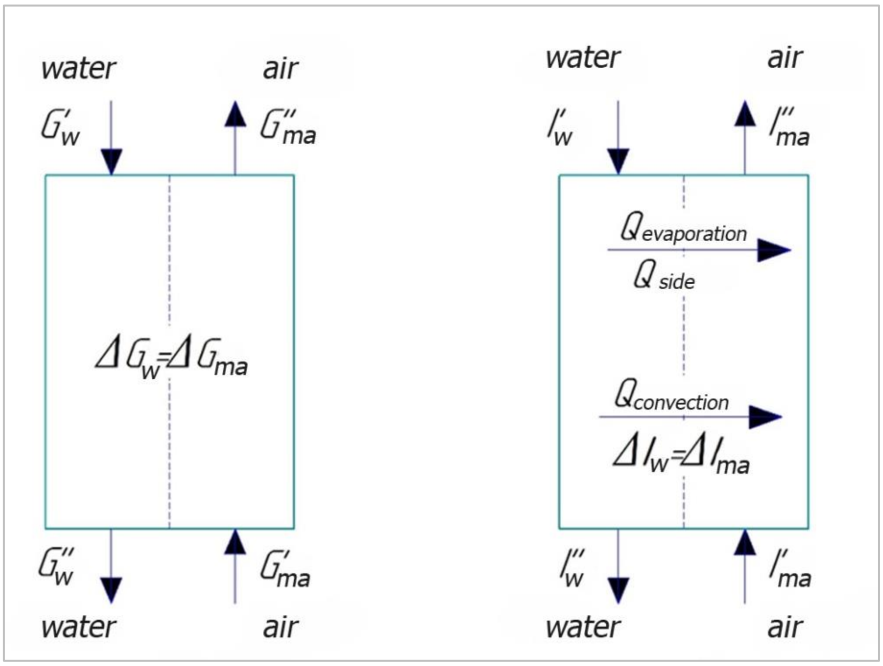

Fig. 2. a) The basic design scheme of the material and heat balance of the evaporative cooler: $G_w^{'}$and $G_w^{''}$- respectively, flows cooled (at the inlet) and cooled (at the outlet) from the evaporative cooler of the circulating water, $G_{ha}^{'}$and $G_{вв}^{''}$- accordingly, the flows of humid air at the inlet to the evaporative cooler and at the outlet from it, $\Delta G_{fw}$is the flow of moisture (water vapor) passing from the cooled water to the humid air. b) $I_w^{'}$and $I_w^{''}$- respectively, the enthalpy of

the cooled (at the inlet to the evaporative cooler) and cooled (outlet of it) circulated water, $I'_w$and I''_{hw}- respectively, the enthalpy of humid air at the inlet and at the outlet of the evaporative cooler, $Q_{ev}$and Q_{conv}- respectively, heat flow passing from the cooled water to moist air by evaporation and convective heat transfer, Q_{hf}^{tank}- heat flow through the side wall of the cooler.

$$dQ_{water} = dQ_{conv} + dQ_{ev} \quad (1)$$

Where

$$dQ_{water} = G_w * Cp_w * dt_w \quad (2)$$

dQ_{water} - heat flow given off by the cooled water

$$dQ_{conv} = \alpha_c * (t_{ha} - t_a) * dF_{he} (3)$$

dQ_{conv} - explicit (convective) heat flow from humid air to water;

$$dQ_{ev} = \beta_P * r * (P_в - P_0) * dF_{he} \quad (4)$$

dQ_{ev} – latent (by evaporation) heat flow from water to humid air;

$\alpha_к$- convective heat transfer coefficient between humid air and water in the irrigated layer; β_P – mass transfer coefficient between water and moist air in the irrigated layer, referred to the difference between the partial pressures on the evaporation surface ($P_в$) and far from it (iP_0); r - the latent heat of vaporization of water;

$$dF_{he} = a * dV = a * F_{nc} d\delta (5)$$

dF_{he} – heat and mass transfer surface of the irrigated layer in its elementary volume dV;

F_{nc}- surface area of the transverse irrigation chamber; δ is the thickness of the irrigated layer.

Substituting (2) - (5) in (1), we obtain

$$\frac{G_{wd}}{a} * Cp_в * dt_в = \left[\alpha_k * (t_{вв} - t_в) - \beta_P * r * (P_в - P_0)\right] d\delta \quad (6)$$

Where

$$G_{wd} = \frac{G_w}{G_{nc}} \quad (7)$$

G_{wd-} irrigation water flow density, referred to the unit area of the transverse (i.e. live) section of the irrigation chamber.

To solve equation (6), we first express the temperature of the humid air using a dry thermometer t_{ha} through the temperature of the humid air using a wet thermometer (t_w). To do this, we use the conditions of the thermodynamic equilibrium set between the requisitions of apparent and latent heat

$$\alpha_k * (t_{ha} - t_w) = \beta_P * r * \left(P_{t_w} - P_0\right) \quad (8)$$

Suggested in [2,3,10,15,19].

In (8), P_{wt} is the partial pressure of water vapor above the surface of the evaporated water having a saturation temperature, i.e., wet thermometer.

We define the value of from (8) t_{ha}

$$t_{ha} = t_w + \frac{\beta_P * r}{\alpha_k} * (P_{wt} - P_0) \quad (9)$$

Substituting (9) in (6) we get

$$\frac{G_{wd}}{a} * Cp_в * dt_в = \left[\alpha_k * t_w + \beta_P * r * \left(P_{t_w} - P_0\right) - \alpha_k * t_w - \beta_P * r * (P_w - P_0)\right] d\delta$$

or

$$\frac{G_{wd}}{a} * Cp_в * dt_в = \left[\alpha_k * (t_w - t_w) + \beta_P * r * \left(P_{t_w} - P_0\right)\right] \quad (10).$$

To solve equation (10) with relation to the desired temperature t_w- based on the dependence of the resilience of saturated steam over water, we establish an approximation dependence between P_{wt} and t_w , as well as Pw and t_w. As calculations show to establish such a dependence, in the range of t_w from 10 0C to 25 0C, approximations P and P_{wt} through t_w and t_w can be represented as

$P_{wt} = t_w - 1{,}5\ ℃$ (11),

$P_w = t_w - 1{,}5\ ℃$ (12).

Substituting (11) and (12) in (10) we get

$$G_{wd} * Cp_w * dt_w = -(\alpha_k + \beta_P * r) * (t_w - t_w) * a * d\delta$$

From which

$$\frac{d_{wt}}{t_w - t_w} = e^{-\frac{(\alpha_k+\beta_P*r)*a*d\delta}{G_{wd}*Cp_w}} \quad (13)$$

Or

$$t_w^{''} = t_w + (t_w^{'} - t_м) * e^{-\frac{(\alpha_k+\beta_P*r)*a*\delta}{G_{wd}*Cp_w}} \quad (14)$$

As follows from the analysis of solution (14), for $(\alpha_k + \beta_P * r)a\delta \ll G_{wd} * Cp_w$ that is, large values of the specific flow rate of the cooled water with a small thickness of the irrigated layer (δ) $t_w^{''} \to t_w^{'}$, and vice versa, $(\alpha_k + \beta_P * r)a\delta \gg G_{wd} * Cp_w$ then there will be $t_w^{''} \leftarrow t_w^{'}$.

To determine the value of $t_w^{''}$, as follows from (14), ceteris paribus (meaning $t_w^{'}$, G_{wd}, $t_м$ and δ, the values α_k and $\beta_P$are required).

According to the results of experimental studies performed by O.Ya. Kokorin [1, 11] when changing the value of the irrigation coefficient

$$\mu = \frac{G_{water\ d}}{G_{air\ d}} \quad (15)$$

from 0.015 to 0.15 kg of water / kg of air, the values of α_k and $\beta_P$for the irrigated layer of fiberglass with a diameter of 0.135 mm, a length of 200 mm and $a = \frac{F_{mo}}{V} = 29629, \frac{1}{m}$ (Table 1) can be principles, respectively, $\alpha_k = 58{,}15\ \frac{Watt}{m^2*℃}$ и $\beta_P * r = 120\ \frac{kg}{m^2*hour*atm}$.

If we take into account that the value of the specific flow rate of humid air through the irrigated layer (G_{had}) (15) is related to the linear velocity of the flow of humid air (ϑ_{ha}) in expression, then solution (14) can be represented as

$$t_w^{''} = t_w + (t_w^{'} - t_w) * e^{-\frac{(\alpha_k+\beta_P*r)*a*\delta}{\mu*\vartheta_{ha}*\rho_{ha}*Cp_w}} \quad (16)$$

According to the results of calculations by the definition of $t_в^{''}$according to formula (16) with $\alpha_k = 58{,}15\ \frac{Watt}{м^2*℃}$, $\beta_P * r = 120\ \frac{kg}{m^2*hour*atm}$, $a = 16000\ \frac{м^2}{м^3}$, $\delta = 0{,}002\ м$, $\mu = 0{,}1$, $\vartheta_{ha} = 2\ ^{м}/_{c}$, $\rho_{ha} = 1{,}25\ ^{kg}/_{m^3}$ и $Cp_{ha} = 4186{,}8\ \frac{Joules}{kg*℃}$ the value of $t_в^{''}$ при $t_в^{'} = 25$ ℃ and $t_м = 18$ ℃ equals to $t_w^{''} = 18 + (25 - 18) * e^{\frac{-(58+120)*16000*0{,}002}{0{,}1*2{,}0*1{,}25*4186{,}8}} = 18 + 7 * e^{-2{,}0407} = 18{,}91$ ℃.

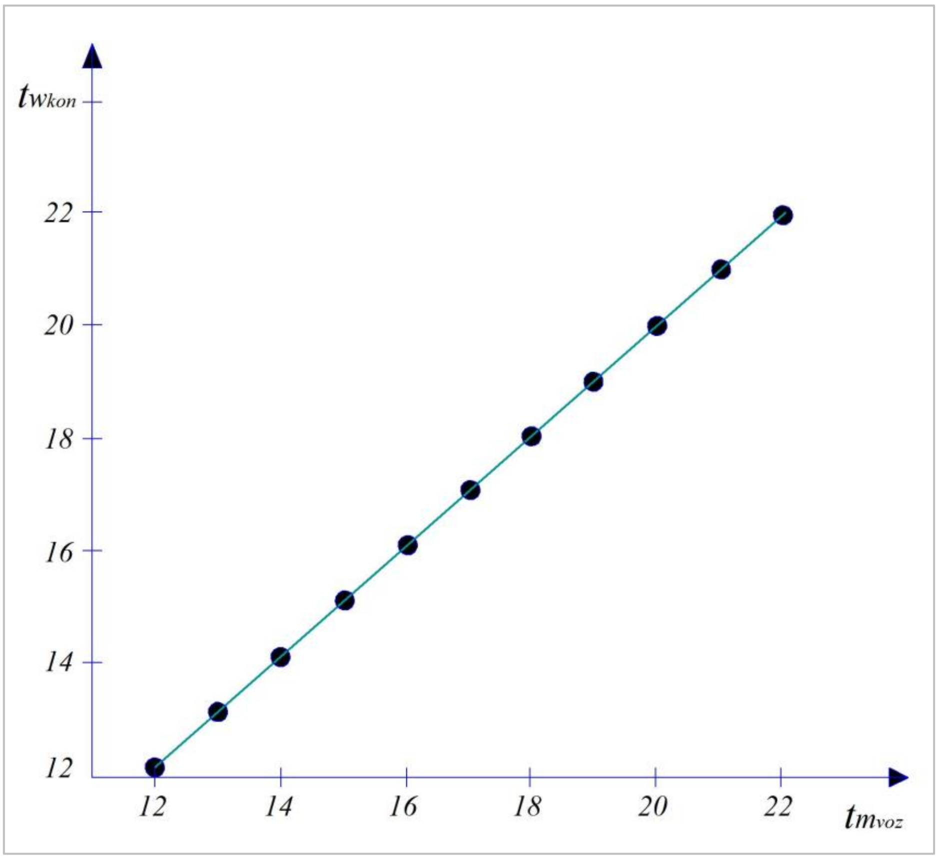

Fig. 3. Dependence of the final water temperature ($t_{w_{kon}}$) of the evaporative circulating water cooler on the temperature and humidity of the wet thermometer ($t_{m_{voz}}$) at φ = 0.25–0.45 [12].

The thermal efficiency value of the evaporative water cooler is determined by the ratio $E = \frac{t_w^{'} - t_w^{''}}{t_w^{'} - t_w}$, while

$$E = \frac{t_w^{'} - t_w^{''}}{t_w^{'} - t_w} = \frac{25 - 18{,}91}{25 - 18{,}0} = 0{,}87$$

and little effect on the values of the coefficients of heat and mass transfer, which is explained by the complete wettability of the material of the irrigated layer even at low values of the irrigation density (G_{waterd}).

The values of $t_w^{'}$ и $t_w^{''}$in (16), in turn, can be determined from the calculated expressions [5] obtained on the basis of thermal modeling of the process of cooling circulating water in evaporative coolers of the considered type.

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