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JOINT MODELLING OF HEAT AND MASS TRANSFER AND AERODYNAMIC PROCESSES IN EVAPORATIVE WATER COOLERS

Statement of the problem. The implementation of the mathematical model of heat and mass transfer in water evaporative coolers shows that a decrease in the cross-section of the nozzle results in a more intense heat and mass transfer. On the other hand, it leads to an increasing aerodynamic resistance of the air path. Therefore joint modeling of heat and aerodynamic processes is needed.

Results and conclusions. The model connecting pressure and flow characteristics of ventilator units and aerodynamic characteristics of evaporative water coolers is examined. This model makes it possible to determine the air flow rate depending on the geometrical parameters of a specific nozzle. The conjunctive use of this model and the heat and mass transfer model allows one, in conjunction with the heat and mass transfer model, to define these parameters in order to obtain the maximum cooling performance.

Keywords: water evaporative nozzle, cooling performance, mathematical model, heat and mass transfer, aerodynamic resistance.

Introduction

As was shown in the paper [5], water evaporative coolers have proven best at attaining specified temperature and humidity parameters of the air.

The basic factors to affect the performance of evaporative water coolers are the parameters of the outside air (temperature, humidity), factors to be reckoned with in this particular climate zone (air flow determined by the type of a ventilator unit and resistance of the air path which is a dependent factor), geometrical parameters (the length of nozzles and cross-section of the nozzles) [3]. The result of the geometrical parameters is the secondary indicators: cooling flow and intensity. Their values are crucial to the cooling performance of the installation [1]:

$$Q = c \cdot \rho \cdot G \cdot \Delta t ,$$

which is affected by the volumetric air flow in the nozzles of the evaporative water cooler G, m^3/sec, and cooling intensity, Δt, ^oC, which is the difference between the inlet and outlet air temperature. In the Formula C is the specific air thermal conductivity, J/(kg·K); ρ is the air density, kg/m^3.

The dependence of the value of cooling performance on the air flow and cooling intensity is pretty much obvious. The air flow rate under specific characteristics depends on the air path resistance which reveal its geometrical features (length and cross-section of the nozzles, narrowing, expansion). Taking into account the resistance and thereby pressure losses in the air cooler induces us to look at the peculiarities of how the air moves from the heat exchange tower standpoint.

The greatest value of the function Q corresponds to the maximum efficient operation mode of the cooler. Until recently there has been a speculation that [4] the cooler should optimally be configured to have the outlet air temperature t_k equal to that of the outside air as read from a wet bulb thermometer t_{nm}. This being the case, the cooling intensity is maximum and relative humidity of the cooled air is 100 %.

The performed research suggests that the dynamics of temperature drops of the cooling air along the nozzles is not constant. At the start of the area the air temperature drops sharply owing to the intense air evaporation off the surface of the baffle plates. Then the intensity of heat and mass transfer processes goes down as the air is saturated with moisture and ultimately these are not of much significance (Fig. 1).

At the same time transport resistance of the nozzle which causes the air flow rate and thereby the cooling performance to drop is directly proportional to the length of the nozzles of the cooler. We have thus suggested the length of the cooling nozzle be reduced, which will lead to a lower cooling intensity but allow a higher air flow rate by reducing aerodynamic resistance. In order to perform quantitative assessment of these changes in the suggested mathematical model [5] it is necessary that aerodynamic resistances are taken into consideration.

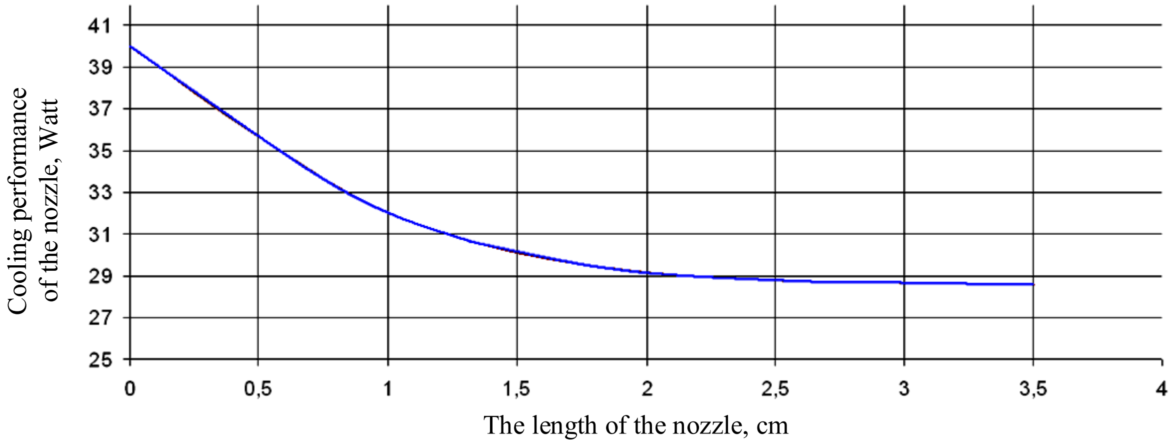

Fig. 1. Dynamics of the air temperature drops across the cooler

1. Aerodynamic resistance

During the operation of the evaporative water cooler as the air flow travels and faces the transport resistance $P_{тр}$, abrupt narrowing P_c, abrupt expansion P_r [2]. Considering the laminar mode of vapor and gas mix travelling inside the cooler as well as the shape of the nozzle and according to the Darcy Weisbah equation we get

$$\Delta P_{тр} = 217\frac{L \cdot V}{h^2}10^{-6},$$

where L is the length of the nozzles of the cooler; V is the air velocity; h is the cross-section of the nozzle.

Local changes in the geometry of the nozzle bring about local resistance which is invariably accompanied by pressure loss. The Darcy Weisbah equation is used to determine pressure loss incurred during the local resistance

$$\Delta P = \xi \cdot \frac{\rho \cdot V^2}{2},$$

where ξ is the coefficient of local losses.

One type of local resistance occurs during abrupt narrowing of the flow which takes place at the inlet of the nozzle

$$\Delta P_c = \xi_c \cdot \frac{\rho \cdot V^2}{2},$$

where the resistance coefficient ξ_c is defined using Table 1.

Table 1

Resistance coefficient during abrupt narrowing

F_0 / F_1	*Re*					
	10^2	**$2\cdot10^2$**	**$5\cdot10^2$**	**10^3**	**$2\cdot10^3$**	**$4\cdot10^3$**
0.1	1.30	1.04	0.82	0.64	0.50	0.80
0.2	1.20	0.95	0.70	0.50	0.40	0.60
0.3	1.10	0.85	0.60	0.44	0.30	0.55
0.4	1.00	0.78	0.50	0.35	0.25	0.45
0.5	0.90	0.65	0.42	0.30	0.20	0.40
0.6	0.80	0.56	0.35	0.24	0.15	0.35

Approximating the table data we conclude that this coefficient is as follows

$$\xi_c = 5.6 \cdot \mathrm{Re}^{-0.4} - 0.12 + \left(0.6 - \frac{F_0}{F_1}\right) \cdot 0.5 .$$

Here F_0 and F_1 are a bypass cross-section and air path cross section respectively. During abrupt expansion which occurs at the outlet of the nozzle

$$\Delta P_r = \xi_r \cdot \frac{\rho \cdot V^2}{2} ,$$

where the resistance coefficient ξ_r is defined using the Table 2.

Table 2

Resistance coefficient during abrupt expansion

F_0/F_1	*Re*					
	10^2	**$2\cdot10^2$**	**$5\cdot10^2$**	**10^3**	**$2\cdot10^3$**	**$3.3\cdot10^3$**
0.1	1.70	1.65	1.70	2.00	1.60	0.81
0.2	1.40	1.30	1.30	1.60	1.25	0.64
0.3	1.20	1.10	1.10	1.30	0.95	0.50
0.4	1.10	1.00	0.85	1.05	0.80	0.36
0.5	0.90	0.75	0.65	0.90	0.65	0.25
0.6	0.80	0.60	0.40	0.60	0.50	0.16

Approximating the table data we conclude that this coefficient is as follows

$$\xi_r = e^{\left(-117\cdot 10^{-9}-\mathrm{Re}^2-\frac{427}{\mathrm{Re}}\right)+\left(0.5-\frac{F_0}{F_1}\right)\cdot 1.6}.$$

Pressure loss is associated with the velocity in the narrow part, i. e. is an average velocity in the nozzles.

The mathematical model of aerodynamic resistance is a transcendent equation. Its right-hand side is the sum of resistance encountered in the way of the airflow and the left-hand side is the pressure and flow characteristics of the ventilator being used:

$$P_1(G) = P_2(G, h, l).$$

The solution of this equation under specific geometrical parameters of the cooler allows the determination of the air flow rate. The joint solution of the mathematical model of heat and mass transfer set forth in [5] and the above model of aerodynamic resistance allows one to determine how temperature and humidity parameters of the cooled air depend on the geometrical sizes of the cooler.

2. Implementation of the joint model

Let us look at how the length of the nozzle and width of the nozzles affect the cooling system performance. A cooler consisting of the ventilator unit BO-14-320-6.3 fitted with the engine АИР71А6 was chosen as a sample. Its pressure and flow characteristics are given by the ratio

$$P_1(G) = e^{-0.029645\cdot e^G + 4.64}.$$

The nozzle is 0.95×0.95 m vertically and horizontally. The parameters of the outside air are: incoming temperature — $t_н = 30$ °C; relative humidity — $\varphi_н = 40$ %.

In the initial area cooling intensity is strongly on the rise as the operating area of the baffle plates increases. Then, after reaching a certain value, its growth flatlines (Fig. 2). The air flow rate is inversely proportional to the growth of the nozzle length. This is due to increasing transport resistance of the cooler (Fig. 3).

The mutual influence of cooling intensity and air flow on the cooling performance is the reason behind its curve behaving in a certain way. The cooling performance rises as the length of the nozzle rises. The cooling performance goes down as the length increases further after reaching a certain maximum (Fig. 4).

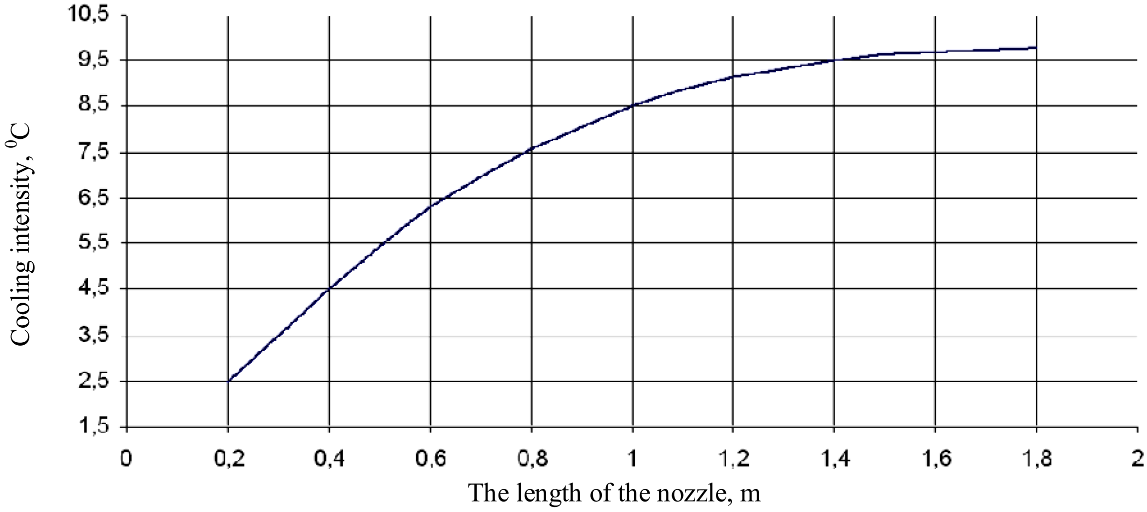

Fig. 2. Dependence of cooling intensity on the length of the nozzle while using the axial fan BO–14–320–6.3

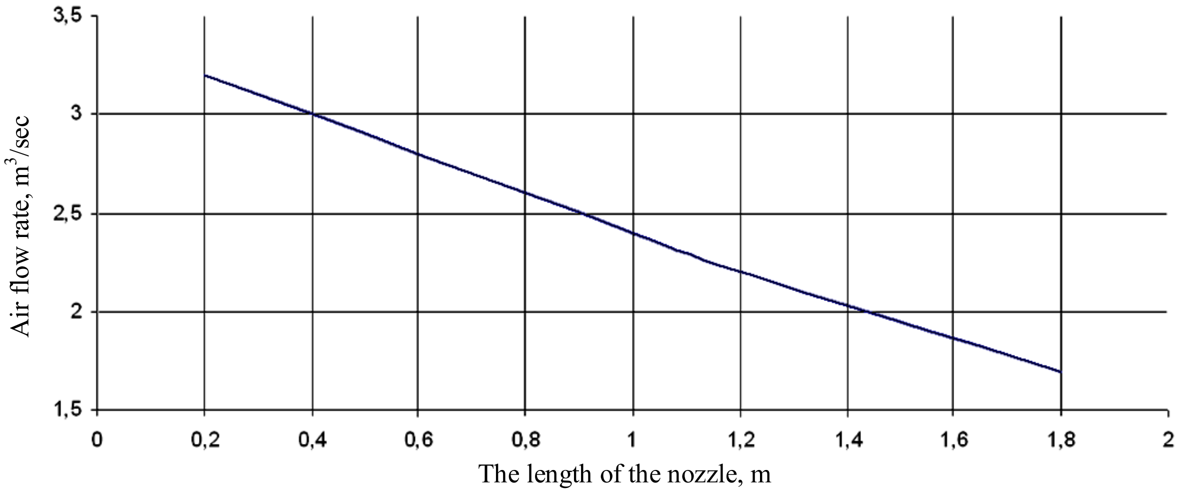

Fig. 3. Dependence of air flow rate on the length of the nozzle while using the axial fan BO–14–320–6.3

It is noteworthy that the cooling performance takes much longer to drop than to grow. This behaviour of the curve is due to a less intense increase in aerodynamic resistance owing to a drop in the air velocity.

The above means that an increase in the length of the nozzle of the cooler over a certain value results in decreasing cooling performance and an unreasonable rise in the relative humidity of the cooled air.

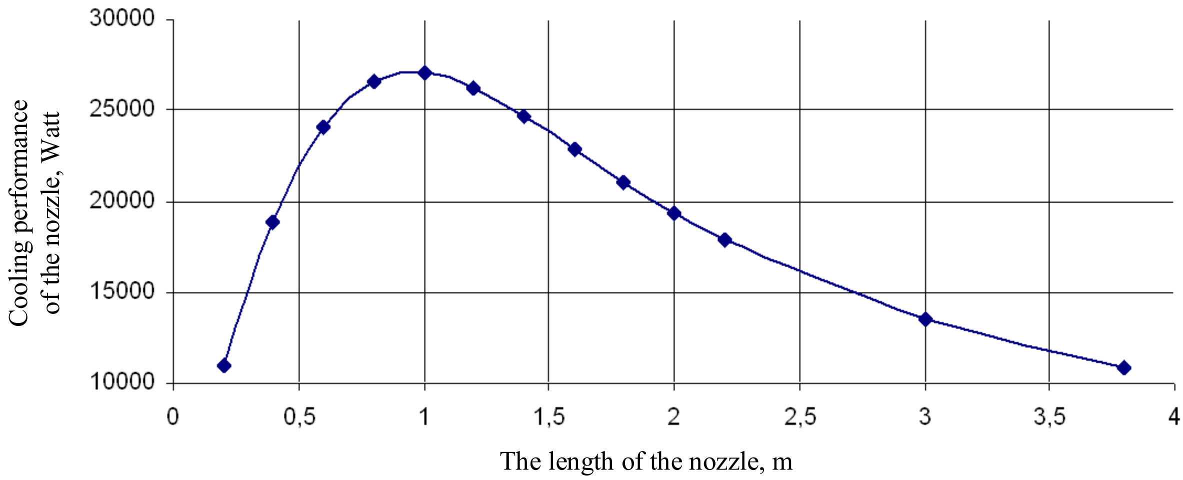

Fig. 4. Dependence of the cooling performance of the nozzle on its length

Conclusions

We have thus designed a mathematical model that allows the determination of the air flow rate in the ventilation system fitted with evaporative water coolers under the known pressure and flow characteristics of ventilator units.

This model in conjunction with the heat and mass transfer model in the nozzles of the nozzle assists in finding most optimal geometrical parameters of coolers depending on the construction solution of the ventilation systems of a variety of industrial facilities.

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