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Voronezh State Agrarian University

D. Sc. in Engineering, Prof. of Department

of Higher Mathematics and Theoretical Mechanics V. P. Shatsky

Assistant lecturer of Department of Higher Mathematics

and Theoretical Mechanics Zh. V. Vysotskaya

Assistant lecturer of Department of Higher Mathematics

and Theoretical Mechanics V. A. Gulevsky

Russia, Voronezh, tel.: +7(4732)71-53-21;

e-mail: vgasupb@mail.ru

V. P. Shatsky, Zh. V. Vysotskaya, V. A. Gulevsky

TO THE QUESTION ABOUT WORK OF WATER- EVAPORATING COOLERS

The results of inclusion of limited number of water vaporizing attachments in ventilation system for normalization of temperature/moisture air parameters are considered. The problem of necessity of achievement of maximally possible depth of cooling with the use of mathematical model is investigated.

Keywords: ventilation system, water vaporizing attachment, temperature/moisture parameters, maximum depth of cooling.

Introduction

It is well-known that certain temperature/moisture parameters should be kept in premise depending on technological process progressed in it, for this purpose, system of air ventilation should be installed in premises. But in the case of considerable heat gain outdoor air ventilation can't provide temperature/moisture parameters inside the premises. Because of this it is necessary to apply the systems of air-conditioning in the premises. In last decades in Russian Federation and abroad vaporization-type air coolers have been used.

Water is the working substance of such coolers. The advantages of such coolers are low energy intensity, ecological safety, the ease of construction, cooling efficiency self-stabilization. Repair and maintenance of such devices require little operator skill.

The basic structural component of the vaporization-type air cooler is vaporizing attachment in channels of which air cooling occurs in the process of heat and mass exchange. The operation of the coolers is evaluated with such indicators as their capacity and temperature coefficient of efficiency.

Air-cooler capacity depends on air volume consumption in channels of the attachment G, m^3/sec, and depth of cooling Δt, ^oC, that is the air temperature difference at the cooler inlet and outlet:

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

where C is the specific heat capacity of air, J/(kg·K), ρ is the density of air, kg/m^3.

Temperature coefficient of efficiency

$$E = \frac{t_{вх} - t_{вых}}{t_{вх} - t_{нм}}\,,$$

where $t_{вх}$ is the outdoor air temperature, ^0C; $t_{вых}$ is the temperature at the cooler outlet, ^0C; $t_{нм}$ is wet-bulb temperature of the outdoor air, ^0C.

The depth of cooling and air consumption depend on each other under air cooler operation. Theoretical researches of vaporization-type air cooler operation have been performed on the basis of heat balance equation and ***i-d*** diagram calculation of humid air state [2]. This approach allows to evaluate efficiency of the coolers operation by air-cooler capacity and depth of cooling, but it doesn’t reflect evolution of temperature and moisture by water vaporizing attachment length, therefore, this approach doesn’t provide selection of the most efficient geometric parameters of air coolers. Mathematical modeling is one of the most efficient methods for the study of heat and mass transfer in the channels of vaporizing attachment.

The cooler structure with temperature of air at vaporizing attachment outlet t_k equal to wet bulb temperature $t_{нм}$ (E = 1) is considered to be the optimal cooler structure. But it should be taken into account that relative humidity of air at the cooler outlet will be equal to 100 %, and this leads to excess of relative humidity of air inside the premises.

This raises the question of obtaining high air-cooler capacity of straight vaporization-type air cooler under reduction in relative humidity of air at their outlet. This problem requires description of the processes in vaporizing attachments channels. For this purpose mathematical model has been developed which describes heat and mass transfer processes inside the cooler (Fig. 1).

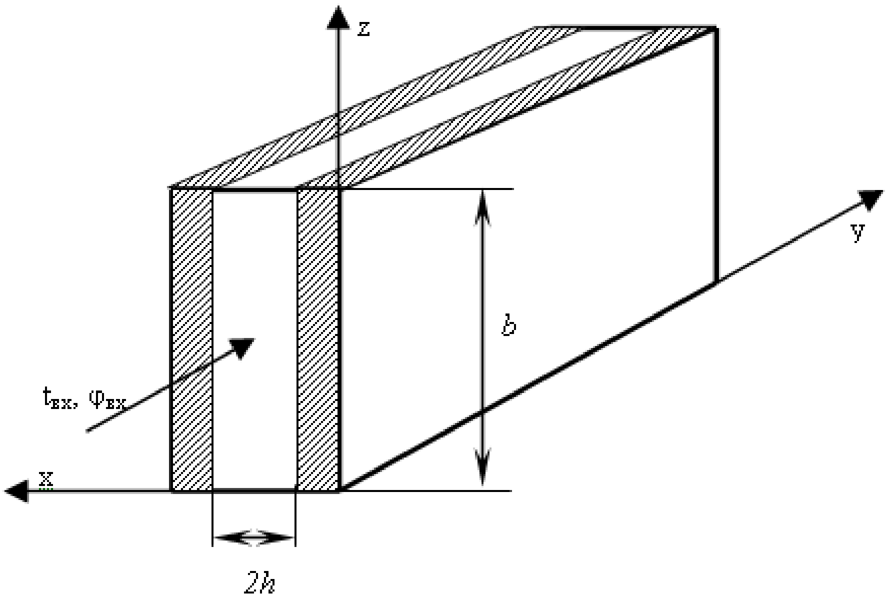

Fig. 1

This model amounts to the system of quasilinear parabolic partial equations:

$$\rho \cdot V(y,z) \cdot C \cdot \frac{\partial t}{\partial x} = \frac{\partial}{\partial y}\left(\lambda \frac{\partial t}{\partial y}\right) + \frac{\partial}{\partial z}\left(\lambda \frac{\partial t}{\partial z}\right) +$$

$$+(C_B - C_\Pi) \cdot \left(\frac{\partial}{\partial y} \cdot \left(\frac{\partial}{\partial y}\left(D \frac{\partial \rho_n}{\partial y} \right) \cdot t \right) - \frac{\partial}{\partial z} \cdot \left(\frac{\partial}{\partial z}\left(D \frac{\partial \rho_n}{\partial z} \right) \cdot t \right) \right), \tag{1}$$

$$V(x,y) \cdot \frac{\partial \rho_n}{\partial x} = \frac{\partial}{\partial y}\left(D \frac{\partial \rho_n}{\partial y} \right) + \frac{\partial}{\partial z}\left(D \frac{\partial \rho_n}{\partial z} \right). \tag{2}$$

Entry conditions are air parameters at the inlet of the cooler $t_{вх}$ and density of vapor in it, determined by relative air humidity $\varphi_{вх}$ expressed as a decimal fraction.

$$t\big|_{x=0} = t_{BX}, \ \rho_\Pi \big|_{x=0} = \varphi_{BX} \cdot \rho_{\Pi H}(t_{BX}),$$

where $\rho_{пн}(t_{вх})$ is the density of saturated vapor at the temperature of inlet air.

The boundary conditions at the surface of the plates are formulated on the basis of energy for liquid vaporization ($R \cdot J$), which is derived from air flow ($\lambda_в \dfrac{\partial t^c}{\partial y}$). It should be taken into account that a portion of energy is extended for heat of cooling liquid entered from the pan $\left(C_{ж} \cdot J \cdot (t_{ж} - t)\right)$. Owing to stationarity of the process the flow of liquid vaporizing from the surface J is equal to the flow of cooling liquid.

Thus, $$R \cdot J = \lambda \cdot \frac{\partial t}{\partial y}\Big|_{y=0} + C_{ж} \cdot J \cdot (t - t_{ж})\Big|_{y=0}.$$

Vapor density at the surface of the plate is taken equal to the density of saturation of at the temperature of the surface of the plate:

$$\rho_n\Big|_{y=0} = \rho_{пн}\left(t\Big|_{y=0}\right).$$

The upper bound is the cooler cover which has good enough heat-insulating properties and naturally is vapor proof. Thus, the upper bound conditions are inpermeability conditions:

$$\frac{\partial t}{\partial z}\Big|_{z=b} = 0,\ \frac{\partial \rho_n}{\partial z}\Big|_{z=b} = 0.$$

As the plates are immersed in the pan with water, the low horizontal bound is water surface. Therefore, temperature with $z = 0$ is taken equal to the temperature of liquid in the pan $t_{ж}$, and vapor density is taken equal to saturation density at this temperature.

$$t\Big|_{z=0} = t_{ж},\ \ \rho_n\Big|_{z=0} = \rho_{пн}(t_{ж}),$$

Evenness conditions are set at the symmetry axis of the channels:

$$\frac{\partial t}{\partial y}_{|y=h} = 0,\ \frac{\partial \rho_n}{\partial y}_{|y=h} = 0.$$

The kinematics in the channels are described by the equation [1]:

$$V(y,z) = \frac{8 \cdot \xi \cdot \rho \cdot w^2}{\pi^4 \cdot \mu \cdot d_э} \cdot \sum_{m=1,2...} \sum_{n=1,2...} \frac{\sin\dfrac{m \cdot \pi \cdot y}{2 \cdot \delta} \cdot \sin\dfrac{n \cdot \pi \cdot z}{b}}{m \cdot n \cdot \left(\dfrac{m^2}{4 \cdot \delta^2} + \dfrac{n^2}{b^2}\right)},$$

where ξ is the coefficient of friction resistance for right-angled pipe, μ is the dynamic viscosity, Pa·sec, ρ is the density of air-vapor mixture, kg/m^3, $d_э$ is the equivalent diameter, m, w is the section-average velocity, m/sec.

$$D = 10^{-5} \cdot exp\ (0.00616 \cdot t + 0.719),$$

$$\rho_{пн} = exp(0.0553 \cdot t - 5.165),$$

$$\lambda = 0.01 \cdot (2.44 + 0.007 \cdot t),$$

$$R = (2500.6 - 2.372 \cdot t) \cdot 10^3.$$

The calculations using this model showed (Fig. 2) that excessive increase in cooler length causes increase in depth of cooling, for temperature at the outlet of cooler cannot be lower than wet bulb temperature.

Therewith it is known that increase in attachment length causes reduction in air consumption and reduction in air-cooler capacity.

In parallel with theoretical researches a series of experiments has been performed, to study influence of geometrics parameters of the cooler on its mode of operation.

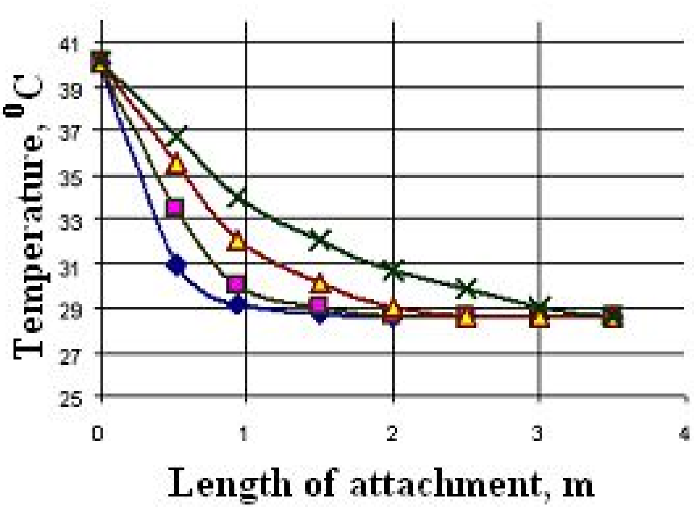

Fig. 2

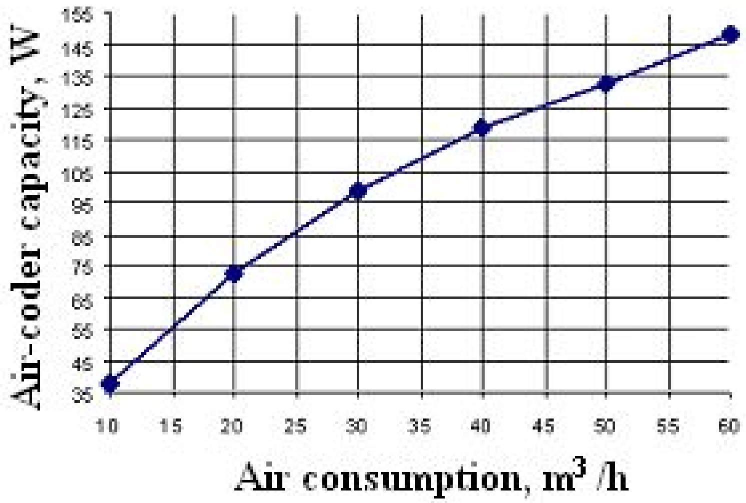

Fig. 3

The results obtained make it possible to draw following conclusion: 5—6 times increase in air consumption brings about 3—4 times increase in air-cooler capacity, in so doing depth of cooling decrease by 3—4 ^{o}C (Fig. 4).

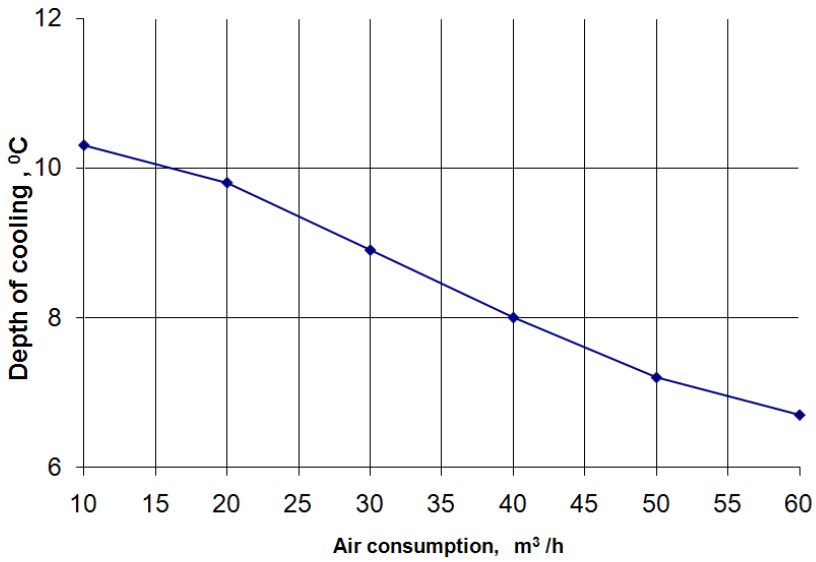

Fig. 4

On this basis it has been suggested to decrease relative humidity of air at the outlet of the cooler by decrease in its length to increase air-cooler capacity.

Using straight cooler (1 m×1 m with channel width 4 mm, outdoor air temperature $t_н = 30$ °C, relative humidity of air $\varphi_н$=40 %) as an example calculations have been performed. It has been found (see Table) that decrease in cooler length makes it possible to double air-cooler capacity, in doing so relative humidity of air at the outlet of the cooler decrease by 8—10 %.

Table

L, m	*G*, m^3/sec	$t_к$, °C	$\varphi_к$, %	*Q*, W	*E*
1.2	0.98	23.8	99.9	14475	0.99
0.8	1.60	24.3	95.1	21587	0.92
0.6	2.20	25.4	88.1	27854	0.87

Conclusion

Thus, suggested model allow to increase refrigerating capacity of the cooler when reducing its length and meeting certain conditions.

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