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DEVELOPMENT OF AN EVAPORATING TYPE AIR COOLER FOR VENTILATION SYSTEMS

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Statement of the problem. The air supplied to the premises during the hot season must be cooled to comfortable temperatures. Due to the fact that additional energy consumption for this cooling is not provided, it is possible to use an evaporative-type air cooler. However, the currently known results of experimental and theoretical studies of such devices do not allow their design, which prevents their spread. The structure of such an apparatus is considered and its theoretical and experimental studies are carried out, the results of which can be used for engineering calculation and design of such apparatus.
Results. An evaporative-type air cooler designed by the authors for ventilation systems is described. A theoretical and experimental study of the air cooler has been carried out. Analytical relationships were obtained for determining the time of movement of the material checker in the "wet" chamber of the apparatus, the temperature of the cooled air and the temperature of the checker in any section of the circulation loop. Empirical relationships have been obtained for the efficiency coefficient of the cooler and its hydraulic resistance.
Conclusions. The obtained dependencies will serve as the basis for the development of a methodology for the design calculation of indirect-evaporative air coolers with a moving fluidized bed in the field of centrifugal forces.

Keywords: ventilation systems, air cooler, centrifugal fluidized bed, time, temperature, efficiency coefficient, hydraulic resistance.

Introduction. Supply and exhaust ventilation systems for industrial and public premises are known to be designed for comfortable and safe conditions in areas. The major issues associated with these systems emerge during the cold season when the outside air temperature exceeds the design temperature for ventilation, which is typically due to the lack of heat for the supply air. However, there can be some problems during the hot season as well when the outside air temperature is higher than the comfortable one, as the energy required to lower the temperature of the air entering the room is not provided in the system of ventilation and conditioning (SVK) data [18, 19]. It is possible to use renewable energy sources to reduce energy consumption for producing cold. One of these can be the method of water-evaporative cooling [14, 15, 17] relying on the thermodynamic non-uniformity of the atmospheric air. Its use in the air cooler for ventilation and air conditioning systems of industrial and public premises will allow energy saving reserves to be more completely utilized [8—10, 16].

1. Schematic diagram of the air cooler. One of the types of intermediate cooler is a nozzle which circulates in the annular channel layer of fluidized dispersed material. It has a high specific contact surface of the interacting phases and thereby a high intensity of heat and mass transfer between them. This nozzle is made of corrosion-resistant particles of various shapes with an equivalent diameter of 1 to 6 mm, which is central to their high wettability. Such a nozzle has a low cost and allows heat to continuously transfer from the main (cooled) air flow to the auxiliary (cooling) [1,2]. The schematic diagram of the indirect evaporative cooling apparatus is shown in Fig. 1 [11, 12].

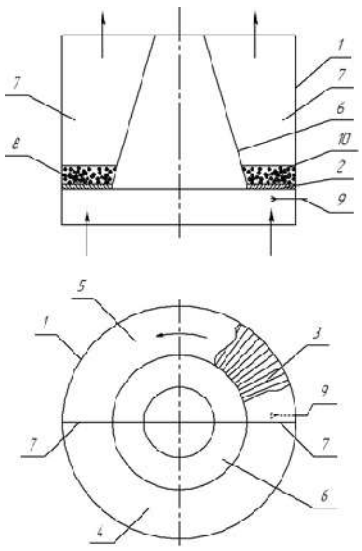

Fig. 1. Schematic diagram of the air cooler

Its major element is an annular working chamber 1 with a central part of the conical shape 6 separated by two partitions 7 into a «dry» section 4 and a «wet» section 5. The partitions are made of overflow windows 8 for free circulation of dispersed material (nozzles) 2 with inclined blades 3 serves to form a circulating fluidized bed. The nozzle 9 is designed to moisten the material in the «wet» section of the operating chamber.

The design of such an air cooler calls for a scientific basis for the development of methods for its design thermal and hydraulic calculations. This device was investigated in order to obtain ratios for identifying its design and operating parameters.

2. Modeling of the parameters of the air cooler. The time when the particles in the "wet" chamber dry out is given by the heat balance equation [3, 4]. Let us assume that as time passes, the volume of water $d\tau$ on the surface of the particle will decrease by dv.

$$qf_ч d\tau = -\rho_{ж}\left[c_{ж}\left(t_{нас} - t_{ж}\right) + r_п\right]dv, \tag{1}$$

q is the density of the heat flow; $f_ч$ is the area of the surface of the wetted particle; $t_{нас}$ is the saturation temperature; $t_{ж}$ is the water temperature; $r_п$ is the hidden vaporization temperature; $\rho_{ж}$ is the water density; $c_{ж}$ is the water heat capacity; v is the volume; τ is the time.

Given that $dv = f_ч dr$, using (1) we get

$$\tau = -\int_{r+\delta}^{r} \frac{\rho_{ж}\left[c_{ж}\left(t_{нас} - t_{ж}\right) + r_п\right]}{q} dr, \tag{2}$$

r is the diameter of the particle; δ is the thickness of the water film on the particle surface.

Convection is the major method of heat supply perceived by the particle surface. Therefore using the Newton-Richman law, the heat flux density is identified:

$$q = \alpha\left(t_в - t_{нас}\right), \tag{3}$$

α is the interphase thermal coefficient in the «wet» chamber; $t_в$ is the temperature of teh atmospheric air.

Assuming the insignificance of interfacial heat exchange through a thin surface water film of the particle, the heat transfer coefficient is given by the following criterion equation [6, 7, 13]:

$$Nu_м = 0.51\,\mathrm{Re}_н^{\,0.65}, \tag{4}$$

$Nu_м = \dfrac{\alpha d_э}{\lambda_в}$ is the Nusselt's criterion; $\mathrm{Re}_н = \dfrac{w_в d_{экв}}{\nu_в}$ is the Reynolds criterion; $\lambda_в$ is the coefficient of the air heat conductivity; $\nu_в$ is the kinematic viscosity coefficient; $d_{экв}$ is the equivalent diameter of the nozzle particle; $w_в$ is the velocity of the air.

Solving and integrating the system of equations (2), (3) and (4) we get a formula for calculating the drying time of the particle

$$\tau = \frac{\rho_{жс}\left[c_{жс}\left(t_{нас} - t_{жс}\right) + r_{п}\right]}{0,42\lambda_{в} w_{в}^{0.65} \nu_{в}^{-0.65}\left(t_{в} - t_{нас}\right)} \cdot \left[\left(r+\delta\right)^{0.65} - r^{0.65}\right]. \tag{5}$$

The solution of the system of differential equations of heat balance and Newton-Richman (in regards to the elementary volume of the fluidized layer) [5] made it possible to obtain an expression for the distribution of air temperature along the height of the layer of the dispersed nozzle

$$t_{в} = t_{н} + (t_{в}^{'} + t_{н}^{'})\exp\left(-\frac{\alpha\left(1-\varepsilon\right) f_{vн}}{c_{в} w_{в} \rho_{в}} y\right), \tag{6}$$

y is the coordinate; $t_{в}^{'}, t_{н}^{'}$ is the water and nozzle temperature at the inlet into the «dry» chamber; $t_{н}$ is the nozzle temperature; $f_{vн}$ is the specific surface of the nozzle layer; ε is the porosity of the nozzle layer; $\rho_{в}$ is the air density; $c_{в}$ is the air heat conductivity.

Significant movement of particles along the height of the fluidized layer during its motion enables us to conclude that there is a dependence of their temperature only in the longitudinal direction of the chamber (from the x coordinate) as shown in the following relation [5]:

$$t_{н} = t_{н}^{'} + \left(t_{в}^{'} - t_{н}^{'}\right)\exp\left\{-\frac{c_{в} w_{в} \rho_{в} x}{c_{н} w_{н}\left(1-\varepsilon\right)\rho_{н} h}\left[1 - \exp\left(-\frac{\alpha f_{vн} h\left(1-\varepsilon\right)}{c_{в}\rho_{в}}\right)\right]\right\}, \tag{8}$$

$\rho_{н}$ is the nozzle density; $c_{н}$ is the heat capacity of the nozzle; $w_{н}$ is the velocity of the intermediate cooler; h is the height of the layer; x is the transverse coordinate.

3. Experimental study of the air cooler. Experimental studies have made it possible to assess the operability of the device and to evaluate one of its most important parameters, i.e., the coefficient of thermal efficiency. Two coaxial cylinders 0.3 m and 0.2 m with the height of 0.5 m form the body of the experimental air cooler. The outer case is made of polymethyl methacrylate for convenient visual observation of processes. The «dry» and «wet» chambers are formed by 2 vertical partitions. The height of the partitions is less than the height of the chambers, which allows the bottom of the device flow windows to be arranged. There is also a gas distribution grid made in the type of blinds. It is possible to change the angle of inclination of the blades in the range from 20 to 40 °. The air supply to each chamber of the device was carried by high-pressure fans, and its flow was measured by averaging tubes with multi-limit micromanometers with an inclined tube MMN-240. The universal temperature regulator TPM138 in conjunction with the computer and thermocouples of the HC type served to measure and store in memory of the air temperatures in various points of the device. The nozzle was a particle of aluminozinc alloy ($\rho_н$ = 2850 kg/m^3, $d_{ЭКВ}$ = 2.6; 2.9; 4.6; and 5 mm) and quartz sand ($\rho_н$ = 2650 kg/m^3, $d_{ЭКВ}$ = 2.7 and 3.2 mm). For various experiments in the range of 0.5—3.5 kg, the weight of the nozzle was varied. A mechanical nozzle providing a water flow rate of 0.0004—0.0024 kg/sec moistened the dispersed material. The hydraulic resistance of the apparatus chambers was measured by differential pressure sensors connected to the TPM-138. The general view of the experimental air cooler installed in the scientific laboratory of the TPTE Department of the VSTU is shown in Fig. 2.

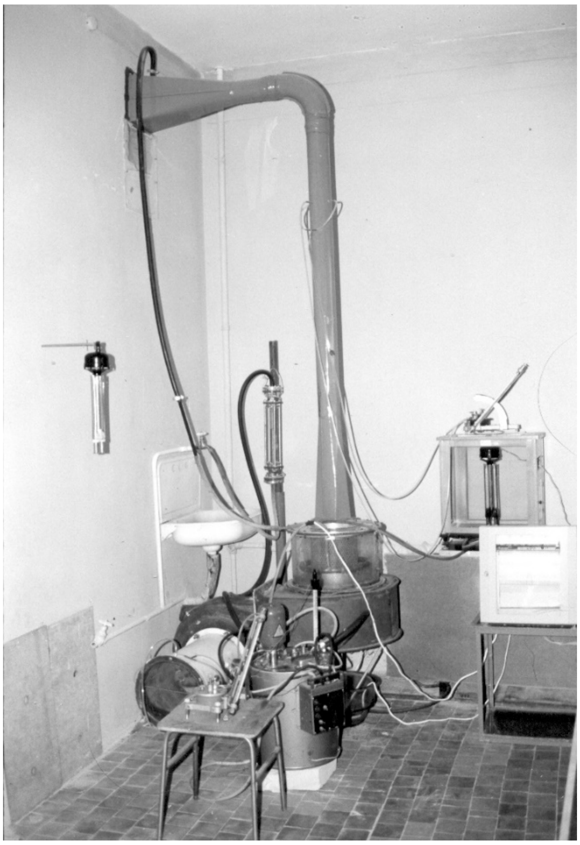

Fig. 2. General view of the experimental air cooler

The following experimental studies were conducted. A certain amount of nozzle was put into the device. Pressure fans were turned on. To ensure reliable movement (circulation) of the dispersed material in the chambers of the device, the flow rates of both air streams were regulated. Then, using the nozzle at the inlet to the «wet» chamber, the nozzle was wetted with water. Measurement of temperatures and air flow rates (main and auxiliary flows, 12 points) as well as the hydraulic resistance of the setup chambers was performed only after the quasi-static mode had been set. Overall, more than 50 mode parameters of the air cooler were examined.

According to the measurement results, the thermal efficiency coefficient was calculated

$$\eta = \frac{t_в^{'} - \overline{t_в}^{"}}{t_в^{'} - t_м} \cdot 100\ \%, \tag{9}$$

$t_м$ is the temperature of the «wet» thermometer; $\overline{t_в}^{"}$ is the average integral air temperature at the inlet to the «dry» chamber.

As a result of the analysis of the results, the major parameters central to the thermal efficiency of the air cooler were obtained. These include the air speed in the «dry» chamber of the device and the mass of the nozzle. The least squares approximation applied to the experimental data made it possible to obtain an empirical equation for calculating the thermal efficiency coefficient

$$\eta = 4.55 w_в^{1.01} M^{0.94}, \tag{10}$$

M is the nozzle mass in the device, kg.

Fig. 3 shows some of the results of experimental and calculated data from (10). The standard deviation does not exceed 3 %.

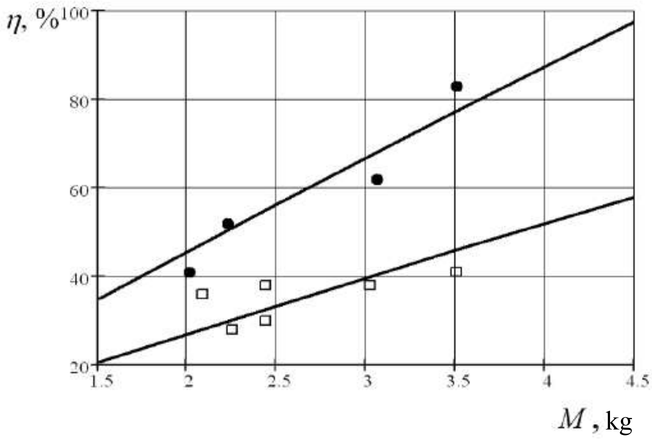

Fig. 3. Graph of the dependence of the efficiency coefficient of the device on the nozzle mass and velocity of the major air flow:

● - $w_в$ = 5,13 m/sec; □ - $w_в$ = 3,06 m/sec; — is the calculation based on (10)

The outcome of the subsequent statistical processing of the results of the experiment on the hydraulic resistance of the apparatus was the following empirical equation

$$\Delta P = 2.58 w_в^{2.15} M^{0.73}, \tag{11}$$

Fig. 4 shows the results of experimental and calculated data based on (11). The standard deviation does not exceed 4 %.

The hydraulic resistance of the «wet» chamber of the air cooler turned out to be 20 % higher than that calculated based on (11).

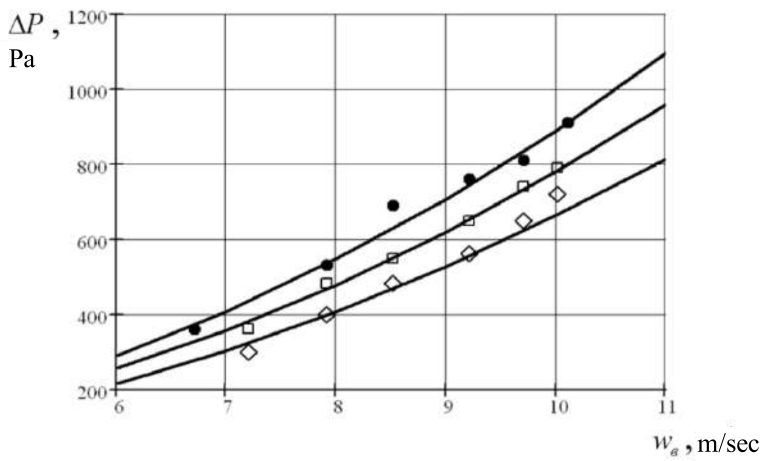

Fig. 4. Graph of the dependence of the hydraulic resistance of the air cooler chambers on the velocity of the major air flow and the mass of the nozzle:

● - M = 3 kg; □ - M = 2,5 kg; ◇ - M = 2 kg; — is the calculation based on (11)

4. Conclusion. The efficiency of supply ventilation in hot weather can be increased by using an evaporative air cooler. The article gives a detailed description of the design of the cooler developed by the authors where a fluidized layer in the field of centrifugal forces serves as a nozzle.

Theoretical and experimental studies of such an air cooler have been conducted. Analytical relations were obtained in order to identify the time of movement of the nozzle in the «wet» chamber of the device, the temperature of the cooled air and the temperature of the nozzle in any section of the circulation circuit. Empirical relations are obtained for the efficiency coefficient of the cooler and its hydraulic resistance.

The above dependences serve as a foundation for developing a technique of design calculation of similar air coolers.

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