HEAT AND GAS SUPPLY, VENTILATION, AIR CONDITIONING, GAS SUPPLY AND ILLUMINATION

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SIMULATION OF HEAT AND MASS TRANSFER IN INDIRECT EVAPORATIVE AIR-COOLER

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Statement of the problem. A way of solving the problem of energy conservation in the design of new and modernization of existing facilities construction is the use of natural temperature gradients in the main and auxiliary equipment of heating, ventilation and air conditioning systems. The concept of indirect evaporative air-cooler with centrifugal fluidized bed of particulate material is developed at the Voronezh State Technical University. A mandatory condition for the effective operation of such devices is the presence of good engineering calculation methods. This necessitated the development and implementation of a mathematical model of heat and mass transfer in the air-cooler.
Results. A mathematical model allowing one to determine the height of a bulk layer, an optimum particle size and the length of the “wet” and “dry” air cooler chambers.
Conclusion. The results of mathematical modeling made it possible to obtain analytical relations for the development of engineering methods of calculating indirect-evaporative air cooler.

Keywords: air cooling, temperature gradient, fluidized bed.

Introduction

One of energy-saving resources is known to be the use of non-traditional energy sources as well as available temperature gradients.

Water evaporative cooling is based on thermodynamic heterogeneity of the atmospheric air which is considered to be the energy for obtaining cold in air conditioning systems of industrial and residential premises.

As theoretical and experimental research suggests [1, 2], passive evaporative air coolers are promising to use with a nozzle with a centrifugal pseudo liquefied layer of a disperse material that circulates due to a dynamic interaction of the major and extra layers of the air [3]. Gas dynamic laws of the formation and movement of this layer have currently been thoroughly studied. Reliable analytical and empirical dependencies have been obtained, which allow one to identify the speed of a liquefied gas [4], speed of the movement of a solid phase [5] and hydraulic resistance of a layer [6].

There have been experimental studies of intensity of an interphase thermal exchange in a pseudo liquefied layer moving through different ducts. The results of these experiments are summed up as empirical criterial ratios [6—8]. In order to introduce this device into wide applications, an engineering calculation method has to be developed. Therefore a mathematical model of thermal mass exchange in a passive evaporative cooler has been designed.

1. Structure of a passive evaporative cooler. The general view of the device is in Fig. 1. It consists of an annular space divided by vertical partitions 1 into “wet” and “dry” areas. In the lower part of the partitions (near a gas distribution grid) there are two windows for unobstructed movement of a nozzle. At the beginning of the “wet” chamber there is a nozzle 2 for wetting the material. Water supply should be controlled so that at the end of the “wet” chamber the nozzle dries completely. The temperature of the nozzle decreases. Its specific value equals the temperature of a wet thermometer. In the “dry” chamber the nozzle heats up providing corresponding cooling of the air.

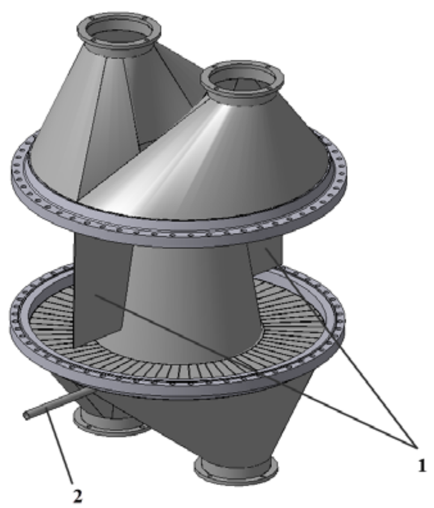

Fig. 1. Passive evaporative cooler

2. Mathematical modelling of thermal mass exchange. Modelling involves determining major structural and operational parameters of an air cooler: height of a nozzle layer, length of the “wet” and “dry” chambers, initial and final air temperatures in the “dry” and “wet” chambers.

In solving these problems the following assumptions were made:

– a temperature field in wetted and dry particles is homogenous;

– particles of a layer intensely mix along a vertical keeping their temperatures constant along its height.

2.1. Determining the height of a nozzle layer. Let us consider thermal mass exchange in the "wet" chamber of the device (Fig. 2). Let us single out an element *dzdy* and design its thermal equilibrium equation:

$$c_в \rho_в w_в dy dt_в = \alpha \left(t_н - t_в \right) f_{уд} dz dy, \tag{1}$$

where c is a heat capacity; ρ is a density; w is a speed; t is a temperature; $f_{уд}$ is a specific surface of disperse material (nozzle); α is a heat emission coefficient; the indices *в, н* are the air and nozzle respectively.

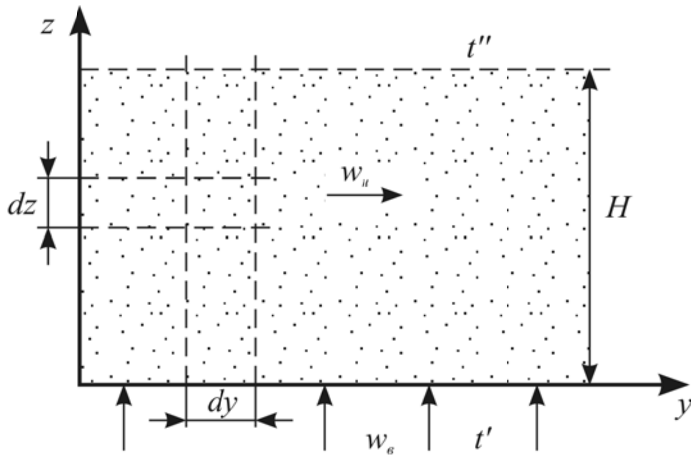

Fig. 2. Calculation scheme

Dividing the variables and integrating the equations (1) under the initial condition $z = 0$, $t_в = t'_в$ we get:

$$t_в = t_н + \left(t'_в - t_н \right) exp\left(-\frac{\alpha f_{уд}}{c_в \rho_в w_в} z \right). \tag{2}$$

The air temperature as it leaves the layer is obtained from (2) at $z = H$ and a necessary height of the nozzle layer is identified based on thermal exchange having to be complete:

$$H = 2{,}3 \frac{c_в \rho_в w_в}{\alpha f_{уд}}. \tag{3}$$

Considering that a specific surface of a disperse material is given by the formula [3]

$$f_{уд} = \frac{6 \cdot (1 - \varepsilon)}{d_э},$$

and a protruding height of the layer is calculated again using the following ratio

$$H_0 = H\frac{1-\varepsilon}{1-\varepsilon_0},$$

finally we get:

$$H_0 = 13{,}8\frac{c_в \rho_в w_в d_э}{\alpha(1-\varepsilon_0)}, \tag{4}$$

where $d_э$ is an equivalent diameter of the nozzle particles; ε is porosity of a centifugal pseudo liquefied layer; ε_0 is protruding porosity of a disperse material.

For a coefficient of heat emission which are typical of a centrifugal pseudo liquefied layer ($\alpha = 400\ldots800$ Watt/(m^2·K)) and for optimal sizes of a disperse material particle ($d_э = 1\ldots4$ mm) a protruding height of a nozzle layer in an air cooler will be $H_0 = 20\ldots60$ mm. It should be noted that the height of a nozzle layer determined based on mass exchange having to be complete in the "wet" chamber of the device is almost identical with the result obtained by means of the equation (4).

2.2. Length of the "wet" chamber of the device. In order to determine the temperatures of the nozzle $t_н$ using the coordinate y let us write the equation of a heat balance for the element Hdy (Fig. 2):

$$c_н G_н dt_н = c_в r_в w_в (t'_в - t''_в) b dy + r_в w_в (x' - x'') b r_п dy, \tag{5}$$

where $G_н$ is mass consumption of the nozzle; b is the width of the chamber of the device; x' and x'' is the moisture content of the air at the inlet and outlet of the device; $r_п$ is latent heat of vapour.

The moisture content at the inlet and outlet of the "wet" chamber can be expressed using corresponding partial pressures of water vapour using the formula [1]

$$x = 0{,}622\frac{P_п}{P}, \tag{6}$$

where $P_п$ is the partial pressure of water vapor in the air; P is the total air pressure.

The partial pressure of water vapor in any transverse section of the layer is [1]

$$P_п = P_{пw} + (P'_п - P_{пw}) exp\left(-\frac{\beta f_{уд} P}{0{,}622\rho_в w_в R_п \overline{T}} z\right), \tag{7}$$

where $P_{пw}$ is partial pressure of vapor at the surface of the particles; β is a coefficient of mass transfer from the air to the surface of the particles; $R_п$ is a gas constant of vapor; $\overline{T}$ is an average absolute of the air temperature.

The dependence of a partial pressure of vapor at the surface of the particles P_{nw} on the absolute temperature of the particle $T_н$ is given by the ratio [9, 10]

$$P_{nw} = P'_n \left[1 + \frac{r_n}{R_n} \cdot \frac{T_н - T'_{нас}}{\left(T'_{нас}\right)^2} \right], \tag{8}$$

where $T'_{нас}$ is the temperature of saturation of vapor for a partial pressure of vapor P'_n. After inserting (6), (7) and (8) into (5) and integrating under the initial condition $y = 0$, $t_н = T'_н$ we get:

$$t_н = \frac{A_1 t'_в}{A_1 + A_2} + \frac{A_2 t'_{нас}}{A_1 + A_2} + \left[t'_н - \frac{A_1 t'_в}{A_1 + A_2} + \frac{A_2 t'_{нас}}{A_1 + A_2} \right] \cdot exp\left[-\left(A_1 + A_2\right) y \right], \tag{9}$$

$$A_1 = \frac{c_в \rho_в w_в b}{c_н G_н P R_n \left(T'_{нас}\right)^2} \left[1 - exp\left(-\frac{\alpha f_{уд} H}{c_в \rho_в w_в} \right) \right],$$

$$A_2 = 0,622 \frac{\rho_в w_в b r_n^2 P'_n}{c_н G_н P R_n \left(T'_{нас}\right)^2} \left[1 - exp\left(-\frac{\beta f_{уд} H}{0,622 \rho_в w_в R_n T} \right) \right].$$

From the physical ratios we conclude that at $y \to \infty$ the temperature of the nozzle equals that of a wet thermometer and thus based on (9) we get:

$$t_м = \frac{A_1 t'_в}{A_1 + A_2} + \frac{A_2 t'_м}{A_1 + A_2}, \tag{10}$$

where $t_м$ is the temperature of a wet thermometer.

Specifying the ratio

$$\frac{t''_н - t_м}{t'_н - t_м} = 0,1$$

using a combination of solutions for (9) and (10), we find a necessary length of the "wet" chamber:

$$L_м = \frac{2,3}{A_1 + A_2}. \tag{11}$$

2.3. Length of the "wet" chamber of the device. Calculation ratios for the "wet" chamber it can be obtained using corresponding formulas for the "wet" chamber assuming that a coefficient of mass transfer $\beta = 0$.

Then based on (9) we get:

$$t_м = t'_в + \left(t'_м - t'_в\right) exp\left(-A_1 y\right), \tag{12}$$

and based on (11):

$$L_м = \frac{2,3}{A_1}. \tag{12'}$$

The temperature of a nozzle at the outlet of the "wet" chamber is determined according to (12) by inserting $y = L_c$ and a heat flow diverted from the air to the nozzle in the "wet" chamber:

$$Q = c_н G_н \left(t'_в - t'_н\right)\left[1 - exp\left(-A_1 L_c\right)\right]. \quad (13)$$

Then the air temperature at the outlet of the "wet" chamber equals

$$t''_в = t'_в + \frac{Q}{c_в \rho_в w_в b}. \quad (14)$$

Conclusions

1. A mathematical model of thermal and mass transfer in a moving pseudo liquefied layer of a complex disperse material.

2. As a result of implementing the model the analytical dependencies were first obtained allowing one to determine "protruding" height of a dispersed material, a necessary length of the "wet" and "dry" chambers of air coolers and air temperature at the outlet of the "wet" chamber.

3. The obtained analytical ratios are a scientific base for developing engineering methods of calculating a passive evaporative cooler with a dispersed nozzle.

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