

On Problem of Mathematical Modelling of Thermo-Physical Processes in Regenerative Water-Evaporating Coolers

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Abstract. For cooling the air environment of industrial premises water-evaporating air, conditioners are being increasingly applied. The simplicity of their construction, ecological safety and low power consumption distinguish them from the coolers of other types. Cooling the processed air is due to the loss of energy for the evaporation of moisture from the surface of the water-wetted plates that form air channels. As a result of this process, cooled air is often saturated with moisture, which limits the possibilities for the operation of the coolers of this type. In these cases, more complex coolers of indirect principle without such drawback should be applied. The most effective modification of indirect cooling is the installation of recuperative principle units. The paper presents a mathematical model of heat-mass transfer in such water-evaporating coolers. The scheme of realization of this model based on an iterative algorithm of solution of the system of finite-difference linear equations that takes into account longitudinal and transverse thermal conductivity of the heat transfer plates is suggested. The possibility of obtaining the optimal values of the redistribution of the main and auxiliary air flows through the substantiation of the aerodynamic resistance of the output grid is proved. This allows refusing the inclusion in the additional system cooling fan unit for discharging an auxiliary stream of air.

1. Introduction

Water-evaporating air conditioners are simple in design and operation, environmentally friendly, obtain low power consumption, self-regulated by the cooling efficiency depending on the temperature and humidity components of the cooling air.

Experimental studies conducted to justify the choice of design alternative characteristics of such coolers do not often provide the opportunity not only to study the impact of the full range of geometric dimensions and flow characteristics of the cooling blocks on their performance, but also do not allow one to forecast the intensity of the processes of heat and mass transfer under conditions different from the control in the experiment.

Well known methods of modelling based on the balance equations do not allow one to track the flow of physical processes in the channels of the cooler and offer no opportunities to analyse the influence of design parameters on the efficiency of cooling.

A modern approach is to build a numerical experiment based on mathematical modelling of thermo-physical processes.

The main element of the cooler is the evaporation unit, in which channels of the processes of heat and mass transfer occur, causing the air cooling. The unit consists of capillary-porous plates between



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which are the air channels. It is known [1], [2] that the blocks with direct cooling principle are constructively rather simple in manufacturing and operation. Thus the treated air passing through the channels is in direct contact with moistened plates and is cooled due to the energy costs of water evaporation from their surface. This process is carried out according to the adiabatic law, which does not lead to the change of enthalpy of air since the cooling of the air occurs along with the saturated vapours of the evaporated water.

An obvious disadvantage of the direct cooler of the principle action is the wetting of the air and, as a consequence, the limited areas of their operation.

2. Results and Discussion

A similar device can serve as a technical solution to reduce the humidity of the cooled air, based on indirect cooling of the primary air flow [3]. They are more difficult to manufacture and operate, require additional energy costs; however, they are much cheaper than the freon and air coolers and provide cooled air of a sufficiently low relative humidity.

The indirect blocks of cooling air channels are divided into major and minor ones (Fig. 1). The first one is called "dry" as it is the main stream of dry air with temperature T . The second group is called "wet" through which the auxiliary air flow passes with temperature t , in contact with the wet surface of the capillary-porous plates.

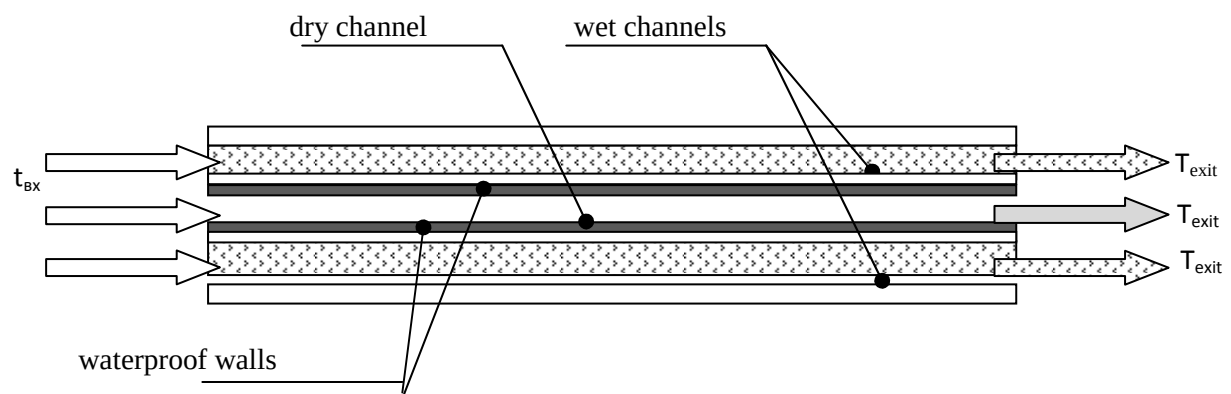


Figure 1. Fragment of block of indirect cooler

The air in the auxiliary channels in contact with the wet surface of the plates, saturated with moisture, evaporates from their surface. As a result of this process, it loses energy through evaporation, its temperature being reduced to t_{exit} after which the air is expelled outside the room to be cooled. The main flow of air coming in dry channels is cooled by heat transfer through waterproof walls to temperature T_{exit} and served in a cooled volume. Its moisture content does not change.

Thus, cooling of the main flow is affected not only by the physical properties of the plates, but their thermal resistance is directly dependent on the cross sectional area of the plate and thermal conductivity of the material, from which it is prepared.

The most effective modification of indirect cooling is the installation of the regenerative principle [4,5]. The scheme of their operation is that a part of the cooled primary air stream at the outlet of the evaporator takes place and is directed into the auxiliary channels of the evaporative pads. Moving in the opposite direction, the flow of liquid vapours, and, due to heat transfer through the fins, absorbs the heat from the primary air flow. Let us note that the temperature at the inlet of the auxiliary channel is unknown and must be determined in the solution process.

Let us note that one of the main characteristics of the efficiency of air coolers is the cooling capacity characterizing the ability to neutralize the heat gain that is equal to $Q = C\rho G_s(T_{in} - T_{out})$,

where G_s is the flow rate of the main flow of air, ρ , C is respectively the air density, kg/m^3 and the specific heat, joule /kg/deg .

The authors presented a model of heat and mass transfer in the channels of the nozzle regenerative evaporative coolers, which calculates all parameters of the treated air, both in length and cross section of the channels. It allows assessing the impact of many parameters (as specified and modified) on the intensity of the processes of thermomass exchange.

Numerical solution of the equations in each separate channel is impossible. This is due to several reasons, the main of which is that the surface of the plate on its border is experiencing a counter heat flow. It does not allow translational movement by steps in the direction of air flow in the channels of the cooler. Therefore, for the numerical solution of the constructed difference, the analogue of the proposed mathematical model was developed [5-7].

Implementation of mathematical models allowed conducting numerical experiments, reflecting the impact of various factors on the efficiency of the coolers.

One of the most important results of the calculations was to determine ratio k of the flow of the main flow to the auxiliary one to obtain the maximum cooling capacity and operation regimes of water evaporating coolers on the performance indicators. Given that the total air flow is the sum of the costs

of core G_s and auxiliary G_m air flows, it was found that $G_s = G \frac{k}{k+1}$ while the cooling capacity is

defined as $Q = C\rho G \frac{k}{k+1} (T_{\text{in}} - T_{\text{out}})$. At a constant total air flow, taking $C\rho G$ as the unit, let the

$Q_{\text{sp}} = \frac{k}{k+1} (T_{\text{in}} - T_{\text{out}})$ be the unit of cooling capacity. The results of theoretical calculations are confirmed by the experiment and show that the cooling capacity is high, which is achieved when the ratio of the main flow to the auxiliary is 3-to-1.

As an example, Fig. 2 presents the graphs of mean consumption temperatures in indirect-regenerative cooler with the length of 0.3 m, the cross-sections of channels - 2 mm, the plate cross section of 2 mm, plate length of 0.3 m, temperature at the inlet is 40°C with relative humidity being 40%. The speed of the overall flow was taken equal to 5 m/s, $k=3$. Lighter tones correspond to higher temperature. The solid line in the graphs corresponds to the temperature of «dry» air; the bar one corresponds to «wet» air.

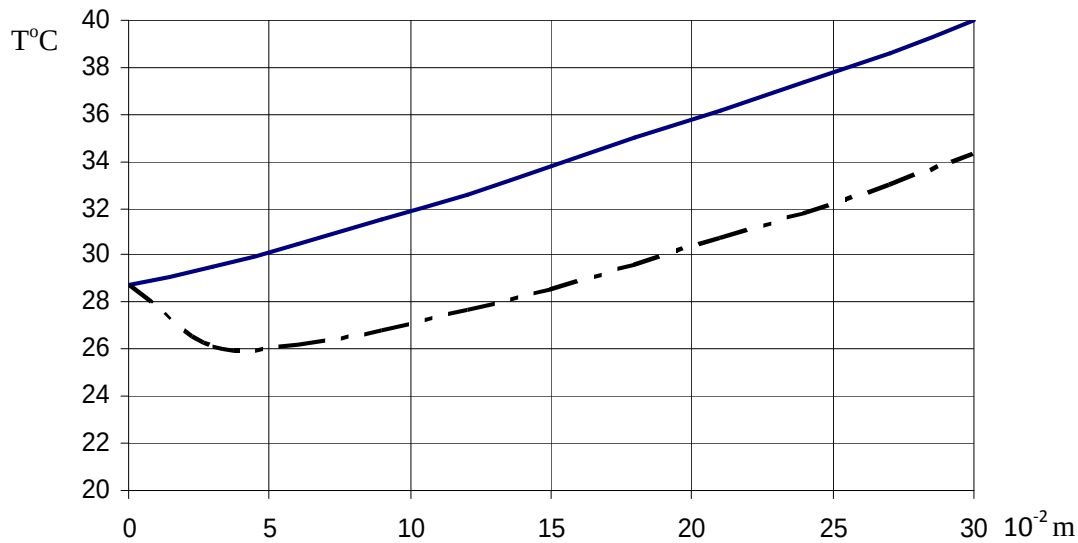


Figure 2. Temperature graph

— main thread

- - - helper thread

3. Modelling

It becomes possible to achieve such flow distribution without introducing additional auxiliary fans for air flow, using the installation at the outlet of the main air flow grille, which in addition to decorative functions will create a desired resistance to the flow of «dry» air to provide the necessary support flow.

The drag coefficient of a flat lattice depends on the ratio of the discharging grid, $f = (\Sigma f_{\text{port}}) / F_p$, (F_p is the total area of the grid, Σf_{port} is the sum of the areas of the holes), on the form of the edges of the holes, as well as on the Reynolds number:

$$Re = \frac{V_{\text{port}} \cdot d_{\text{port}}}{\nu},$$

where V_2 is the speed in the hole, m d_{ot} is the hole diameter, m ; ν is kinematic viscosity coefficient, m^2/s . Pressure loss during the passage of the main stream grid is defined by the formula:

$$\Delta P_{\text{port}} = \xi_{\text{port}} \frac{\rho V_2^2}{2},$$

where V_2 is the average speed before the obstacle, and the coefficient of resistance is determined by formula [10]: $\xi_{\text{port}} = \xi / f^2 + E \cdot \xi_1$

Values for coefficient E is determined by approximating the corresponding formula in the range of $100 \leq Re \leq 4 \cdot 10^3$

$$E = (0.0929 \cdot \ln Re - 0.221)^{0.5}.$$

Coefficient ξ_1 can be computed as follows [9]:

$$\xi_1 = 1/f * [0.707 * (1-f)^{0.375} + 1-f]^2.$$

There are also tabular values ξ depending on the changes in values of Re and «live» cross section of lattice f .

For the software implementation, it is more convenient to use a continuous analogue of this relationship, expressed as a function:

$$\xi = 5,24 \cdot 10^{-9} Re^2 + 0,74 \cdot f^2 + 8,8 \cdot 10^{-5} \cdot Re \cdot f - 1,1 \cdot 10^{-4} \cdot Re - 1,65 f + 0,89.$$

The lattice parameters are determined by numerical solution of the transcendental equation derived by equating the aerodynamic resistance of the main and auxiliary flows.

For example, at the nozzle length of 0.4 m, the cross-sections of the channels equal to 0.002 m and the velocity of the subsidiary stream of 2 m/s, the equation to determine f is as follows:

$$2,26(0,002 / f^2 + 0,741 f^2 + 0,87 - 0,0687 / f - 1,65 f) + \\ 2,26\sqrt{0,1\ln(625 / f) - 0,221(0,707(1 - f)^{0,375} + 1 - f)^2} = 57,5 f^2$$

The solution of this equation for flow distribution in the ratio of 3:1 in favour of the main thread is value $f = 0.25$.

4. Conclusion

In the study of indirect evaporative cooling modelling of physical processes involving heat transfer coefficients, there are a number of difficulties because their expression includes the heat-transfer coefficients, the value of which is difficult to determine. The calculations showed that they not only depend on the cross-section channels, but also on the longitudinal coordinate and the velocity of the air flow. In this regard, modelling should be carried out using differential equations in partial derivatives of elliptic and parabolic types with the corresponding initial and boundary conditions. To avoid a number of simplifications and assumptions, used as the method of implementing the presented model, the solution of the system of finite difference equations is used.

The presented model and the method of its implementation allow determining the temperature of the air flow along the length of the coolers that enables the choice of their geometrical parameters.

In the presented work rational modes of operation of these coolers and the identified geometric parameters of the installations are revealed. The results can be used in the design of the coolers of specified principle of operation.

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