A comparative analysis of schemes of indirect evaporation type apparatuses

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Abstract. In this paper, a comparative analysis of schemes of heat and mass transfer devices of indirect evaporation type is carried out. These devices are a family of plane-parallel channels, in one of which the evaporation of water film takes place. Mathematical modeling of heat and mass transfer devices is based on solving a system of differential equations that represent the heat balance equations for each of the channels. Numerical studies are carried out in a wide range of input parameters: temperature $t_0 = 15\pm 50^\circ$C, relative air humidity $\phi_0 = 20\pm 100\%$, and Reynolds number $Re = 50\div 1500$. The assumed geometric dimensions of the channels are height $H = 6$ mm and length $L = 50H$. The calculations are carried out at atmospheric pressure.

1. Introduction

Heat exchangers, in the channels of which evaporative cooling is implemented, are among the simplest and most effective ways to reduce the temperature of air or liquid flows. One of the inexhaustible energy sources, which is available almost anywhere in the world, is the energy of moist atmospheric air, being a mixture of dry gases and water vapor. In the adiabatic evaporation of water, the temperature of the gas decreases. Temperature disequilibrium in the form of a psychrometric temperature difference between dry and wet thermometers can be used as a renewable energy source [1, 2].

The known literature distinguishes between two methods of evaporative cooling in channels: direct and indirect-evaporation (Figure 1) [3, 4]. In direct evaporative cooling, the air flow is in direct contact with the surface of the water film. In this case, the flow temperature decreases and its moisture content increases. The minimum achievable temperature is the temperature of the wet thermometer ($t_{wb}$). With indirect-evaporative cooling (Figure 1), the working air stream flows in an auxiliary channel that contacts the water film through the separating plate. At that, the moisture content in the working channel remains unchanged, which is the main positive factor. In a countercurrent flow scheme of heat carriers (Figure 1, $a$), the minimum achievable temperature is that of the wet thermometer, similar to the case of direct evaporative cooling.

The schemes of heat and mass transfer devices (HMTD) of the indirect-evaporative type, shown in Figure 1, $b-c$, are known in the literature as By-pass and M-cycle (Maisotsenko cycle) [5], respectively. In this type of apparatus, the maximum cooling value is the “dew point” temperature ($t_{dp}$).
Currently, the thermal characteristics of devices that implement indirect evaporation are actively investigated theoretically [4, 6, 7] and experimentally [8, 9]. The study of the regularities of heat and mass transfer processes in HMTD channels is an important step in the development of more complex indirect evaporation devices. In this paper, we consider the issues of modeling the processes of heat and mass transfer in heat and mass transfer devices of the indirect-evaporative type with countercurrent and regenerative flow schemes of heat carriers as well as their comparison.

2. Modelling and analysis

To simulate the characteristics of heat and mass transfer devices, a small calculation element \( dx \) is separated along the length. The calculated schemes of the elements are shown in Figure 2 (Figure 2, \( a \) corresponds to two-channel schemes, and Figure 2, \( b \) corresponds to a three-channel scheme).

When modelling heat and mass transfer devices, the following assumptions are made:

- the mode of air flow in the channels is laminar and stationary, and the flow is stabilized;
- on the outer sides of the channels, the plates are heat-insulated, and the walls of the wet channel are wetted with a thin film of water evenly along the entire length;
- the parameters of the air flow on the surface of the plates in the wet channel correspond to the saturation parameters;
- the thermal resistance of the plates and the water film is not taken into account.

![Figure 1. Schematic diagrams and i-d diagrams of evaporative apparatuses: a – counter-current diagram; b – By-pass; c – M-cycle.](image)

![Figure 2. Calculation schemes of HMTD.](image)
The system of differential equations that allows determining the main parameters is as follows:

1) for two-channel flow schemes of heat carriers (Figure 1, a-b):
- equation of heat balance of the differential element in question in the dry channel is as follows:

\[ G_c c_p \frac{dT}{dx} = \alpha_d \left( T_d - T_{wf} \right), \tag{1} \]

where \( \alpha \) is the heat transfer coefficient, \( G \) is the mass flow rate of air, and \( c_p \) is the isobaric heat capacity. Equation (1) shows that the total change in the enthalpy of the air flow in this channel is equal to the total heat transfer between the dry and wet channels;
- the law of conservation of energy for air in a wet channel is the same as in a dry one:

\[ G_c c_p \frac{dT}{dx} = \alpha_w \left( T_u - T_{wf} \right). \tag{2} \]

The physical meaning of equation (2) reflects the fact that the change in the enthalpy of air in a wet channel is equal to the total energy transfer by convective heat transfer between the air flow and the water film.
- The law of conservation of mass of steam in a wet channel:

\[ G_u \frac{d(d_u)}{dx} = \beta \left( d_{wf} - d_u \right), \tag{3} \]

where \( d \) is the moisture content of the air flow, and \( \beta \) is the mass transfer coefficient. The left part of equation (3) represents the change in the moisture content of the air flow in the wet channel, and the right part characterizes the convective mass transfer between the air flow and the water film.
- the law of energy conservation on a separation wall may be written as follows:

\[ \alpha_d \left( T_u - T_{wf} \right) dx + \alpha_u \left( T_u - T_{wf} \right) dx = r \beta \left( d_{wf} - d_u \right) dx, \tag{4} \]

where \( r \) is the specific heat of vaporization. Expression (4) means that the change in the enthalpy of the air flow in the dry channel is equal to that in the wet channel.

2) for the three-channel flow diagram of heat carriers (Figure 1, c), in addition to equations (1–2), the following relations are added:

\[ G_{work} c_p \frac{d(T_{work})}{dx} = \alpha_{work} \left( T_{work} - T_{wf1} \right) + \alpha_{work} \left( T_{work} - T_{wf2} \right); \tag{5} \]

\[ G_u \frac{d(d_u)}{dx} = \beta \left( d_{wf1} - d_u \right) + \beta \left( d_{wf2} - d_u \right); \tag{6} \]

\[ \alpha_d \left( T_d - T_{wf1} \right) dx + \alpha_u \left( T_u - T_{wf1} \right) dx = r \beta \left( d_{wf1} - d_u \right) dx; \tag{7} \]
\[
\alpha_d \left( T_d - T_{sf_2} \right) dx + \alpha_w \left( T_w - T_{sf_2} \right) dx = \rho \beta \left( d_{sf_2} - d_w \right) dx,
\]

they also represent the law of conservation of energy in the working and wet channels, respectively.

The indices “d”, “w”, and “work” characterize the parameters of air flows in the dry, wet, and working channels, respectively. The values of the heat transfer and mass transfer coefficients included in the system of differential equations (1–8) are calculated from the dependences given in [2, 10]. To close the system of differential equations (1–8), the boundary conditions given in Table 1 are set.

Table 1. Boundary condition

| Flow diagrams of heat carriers: | counter-current | By-pass | M-cycle |
|---------------------------------|-----------------|---------|---------|
| \( x = 0 \)                     | \( T_d = T_0 \) | \( T_w = T_0 \) | \( T = T_0 \) |
| \( x = L \)                     | \( T_w = T_0 \) | \( T_{d,0} \)  | \( T_{work} = T_0 \) |
|                                 | \( d_w = d_0 \) | \( d_{w,0} \)  | \( d_{work} \) |

The reliability of the data obtained using mathematical modeling is evaluated by comparing it with the results of the PhD work of Khafaji H.Q.A (2016). The change in temperature and moisture content in the HMTD channels is shown in Figure 3.

![Figure 3](image_url)

**Figure 3.** Changes in the parameters of air flows along the length:
\( (t_0 = 30^\circ\text{C}, \varphi_0 = 30\%, \text{Re} = 100, m = 0.5) \).

It follows from the above graphical dependences that the main deviations of temperature and moisture content are observed in the initial sections of the corresponding channels. This is due to the assumption of the present mathematical model, namely, on the constancy of the heat and mass transfer coefficients along the length of the channels. Nevertheless, the difference in the values of the parameters is no more than 1°C for temperature and 1 g/kg for moisture content. This fact indicates the adequacy of the developed mathematical model.

It also follows from Figure 3 that there is a minimum in the temperature distribution along the length in the wet channels, which is due to heat removal to the phase transition, and on the other hand, to the supply of heat from the dry channel. The specified character of temperature changes is observed for all considered flow schemes of heat carriers. It should be borne in mind that the moisture content of the air in the dry and working channels remains unchanged. The decrease in the temperature in the dry channel to its outlet is the main positive factor of the considered schemes of indirect-evaporative HMTD.
Changes in the thermodynamic parameters of the flows with the variation of the input parameters are shown in Figure 4. It can be seen that for all schemes of heat and mass transfer devices, the changes in the parameters are qualitatively similar. At the same time, in regenerative schemes, under the same input conditions, it is possible to obtain a lower temperature level at the outlet of the dry channel in comparison with the countercurrent supply of heat carriers.

One of the important characteristics of heat and mass transfer devices is the value of their cooling capacity, which can be estimated by the expression:

\[ Q = G_s c_p (t_0 - t_{out}) \]

Figure 4. The influence of initial parameters on the output temperature and cooling capacity of the HMTD:

- a \[- \varphi_0 = 30\% , \ Re = 100 , \ m = 0.5 \] ;
- b \[- t_0 = 30^\circ \mathrm{C} , \ Re = 100 , \ m = 0.5 \] ;
- c \[- t_0 = 30^\circ \mathrm{C} , \ \varphi_0 = 30\% , \ m = 0.5 \] .

The data in Figure 4 also show that the lowest values of the temperature at the HMTD outlet, and, consequently, the highest cooling capacity, are observed for the M-cycle flow scheme of heat carriers. An increase in the relative humidity at the inlet leads to an increase in the temperature at the outlet, and, as a result, to a decrease in the cooling capacity of all HMTD schemes. This is explained by the fact that the increase in \( \varphi_0 \) leads to a decrease in the intensity of the water film evaporation.

Conclusions
The developed mathematical model describing the processes of heat and mass transfer during evaporative cooling of air in devices of indirect-evaporative type allows analyzing a complex multiparametric problem.

The obtained cooling effect in HMTD can be quite high and comparable to traditional air conditioning schemes, including steam compression refrigerating machines. Heat and mass transfer devices of the considered flow schemes of heat carriers have high efficiency, low unit cost, low operating costs, and structural simplicity. Despite the advantages, HMTD has a number of drawbacks, primarily due to a decrease in cooling capacity when using air with high humidity. The choice of a specific design scheme for the air flows depends on the purpose and the required thermal and humidity output parameters of the HMTD. The obtained data may be used for optimization analysis of HMTD.
schemes with variations in the Reynolds number, air humidity, channel length, and geometric dimensions.

Acknowledgments
The work was financially supported by the Russian Foundation for Basic Research (grant No. 20-08-00717 A).

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