Heat and mass transfer during water film evaporation in a horizontal channel with a cocurrent flow of moist air

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Abstract. In this paper, we consider the issues of modeling of joint processes of heat and mass transfer during evaporative cooling of a laminar flow of moist air in a flat horizontal channel, taking into account the finite thickness of the water film. Numerical simulation of processes is based on solving the system of differential equations in boundary layer approximation and in two-dimensional formulation. The research is conducted in the range of input parameters: temperature $T_0 = 10 \div 50 \, ^\circ C$, relative humidity $\varphi_0 = 0 \div 100 \, \%$, and Reynolds number $Re = 50 \div 1000$ at atmospheric pressure. The influence of the Reynolds number and relative humidity on the change of local and average thermodynamic and thermohydraulic parameters of flows along the channel length is shown.

1. Introduction
Reducing the temperature using water evaporation is one of the simplest and most effective ways to cool the air flow. Evaporation of liquids in air exists in nature and is used in various industrial applications, such as cooling of electronic devices, in two-phase heat exchangers, solar collectors, when cooling nuclear reactors, in air conditioning and water desalination systems. A large number of works have been devoted to the study of convective heat and mass transfer processes in a water film flow with a cocurrent flow of moist air [1-6]. Research in this direction is currently developing quite intensively; however, this problem is far from full understanding due to its complexity and multiparametricity.

In most works on evaporative cooling, the impact of thermal resistance of the water film on the evaporation rate is neglected. This approximation cannot always be performed, so assessing the influence of the presence of a water film on heat and mass transfer and evaporation rate is of great interest. In this paper, the study of conjugated heat and mass transfer at cocurrent laminar flow of water film and air flow in a horizontal channel is carried out.

2. Computational scheme, modeling and analysis of results
The computational scheme of the channel is demonstrated in Figure 1. The geometric dimensions of the channel are taken as follows: channel height $H = 6 \, mm$ and length $L = 50H$. A film of water is fed into the channel through a slit with height $\delta_0$ at the same temperature as the moist air $T_{0,L} = T_{0,A}$. The Reynolds number for the water film is constant and equal to $Re_L = 0.5$. 

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[1] Gorbachev, M. and Terekhov, V. (2020). Heat and mass transfer during water film evaporation in a horizontal channel with a cocurrent flow of moist air. Journal of Physics: Conference Series, 1677(1), 012086. doi:10.1088/1742-6596/1677/1/012086
The assumptions made when developing the mathematical model are as follows:

- the flow regimes of moist air and water film are laminar. Wave effects on the surface of the liquid film are not taken into account;
- the surface tension of the water film is not taken into account. The interface is in thermodynamic equilibrium and the conditions of equality of temperatures, velocities, heat fluxes and tangential stresses are met;
- the channel walls were adiabatic \( (q_v = q_w = 0) \);
- radiation heat transfer, temperature dissipation due to viscosity, as well as Dufour and Soret effects are not taken into account;
- longitudinal thermal conductivity and diffusion are neglected.

The problem solution is realized in two-dimensional formulation in the boundary layer approximation for the gas and liquid phases. The system of differential equations for water film and moist air flow include the equations of continuity, motion, energy, and diffusion.

Constancy of all parameters for the water film and the moist air flow along the height is set as the boundary condition at the entrance to the channel \( (x = 0) \):

\[
\begin{align*}
    u_L &= u_{0,L}, & u_A &= u_{0,A}, & T_0 = T_{0,A}, & K_A &= K_{0,A}.
\end{align*}
\]

At the interface of the liquid and gas phases \( (y = \delta) \) there is an equality of velocities \( u_L = u_A \) and tangential stresses

\[
\mu_L \left( \frac{\partial u_L}{\partial y} \right)_{y=\delta} = \mu_A \left( \frac{\partial u_A}{\partial y} \right)_{y=\delta}.
\]

The transverse component of the steam flow velocity on the surface of the water film is written as

\[
v_y = -\frac{D}{1 - K_y} \left( \frac{\partial K}{\partial y} \right)_{y=\delta},
\]

and the surface temperature of the evaporating film is found from the heat balance equation:

\[
-\lambda_L \left( \frac{\partial T_L}{\partial y} \right)_{y=\delta} = -\lambda_A \left( \frac{\partial T_A}{\partial y} \right)_{y=\delta} - \frac{\rho_{dr} D}{1 - K_y} \left( \frac{\partial K_A}{\partial y} \right)_{y=\delta},
\]

where \( r \) is the latent heat of water vaporization and \( D \) is the diffusion coefficient. In this case, the evaporation process is considered to be in equilibrium and the vapor concentration is associated with the film surface temperature of the saturation curve \( (K_y)_{y=\delta} = f(T)_{y=\delta} \).

The system of differential equations together with boundary conditions is solved numerically using the finite difference method. The non-linearity of the differential equations is eliminated by simple iterations at each integration step with an accuracy of \( 10^{-5} \) %. The step along the axis is assumed to be
uniform. The grid is compressed evenly along the $y$-axis with a compression ratio of 1.05. Test computational experiments have shown that the optimal grid size is 400x120x70 cells in the longitudinal and transverse (for the water film and air) directions, respectively.

Numerical simulation of joint processes of heat and mass transfer in a horizontal channel is carried out in a fairly wide range of input parameters: temperature $T_0 = 10 + 50 \, ^\circ C$, Reynolds number for air flow $Re_A = 50 + 1000$, and relative humidity $\varphi_0 = 0 + 100 \%$.

The results of numerical studies are the main thermodynamic and thermohydraulic parameters of the water film and the moist air flow, such as: velocity fields, temperatures and concentrations, as well as local characteristics (heat and mass flows, as well as Nusselt and Sherwood numbers).

Figure 2 shows local profiles of velocities, temperatures, and vapor concentrations for different distances from the channel entrance. It can be seen that the velocity profile for the gas phase already at $x / H = 10$ corresponds to the developed flow. At the same time, further away from the input section, the concentration profiles only tend to be evenly distributed. Temperature profiles are not uniform even at a sufficient distance from the channel entrance ($x / H = 50$). This can be explained by the fact that when dry air is supplied to the channel ($\varphi_0 = 0$), in the initial section there are more intense evaporation processes, which lead to a fairly sharp decrease in the temperature of the water film, while the air parameters at the interface almost correspond to the saturation parameters. Further, the process of air cooling due to thermal conductivity proceeds less intensively.

![Figure 2. Profiles of velocities, temperatures and concentrations](image)

The dependences of changes in the average mass parameters of air and water film along the length of the channel with variations in the Reynolds number for air are shown in Figure 3. The Reynolds number of the liquid phase ($Re_L = 0.5$), and the temperature of the vapor-gas mixture at the entrance to the channel ($T_{0,A} = T_{0,L} = 30 \, ^\circ C$) remain unchanged in these calculations. Due to the evaporation process, the bulk temperature of moist air $T_{ma}$ decreases along the length of the channel. In this case, the average temperature of the liquid film decreases more intensively and at a certain distance from the entrance it reaches a saturation state. The weakly expressed minimum temperature values for the water film are due to the influence of increasing moisture content of the air flow on the evaporation intensity. Such a process of reverse influence of evaporated water vapor can strongly affect the conjugated heat and mass transfer and under certain conditions it can completely suspend the evaporation of the liquid film. An increase in the Reynolds number leads to an increase in the bulk temperature of the moist air flow, which
indicates deterioration in the gas cooling process. The average mass concentration of water vapor \((K_{m1})\) due to evaporation processes continuously increases along the length of the channel, but, similar to the temperature, the moisture content decreases significantly at high Reynolds numbers.

![Figure 3](image)

**Figure 3.** Changes of local parameters over the channel length \((T_0 = T_{0,A} = T_{0,L} = 30^\circ C, \varphi_0 = 0, \text{Re}_L = 0.5\).

The rate of \(K_{m1}\) change is proportional to the intensity of liquid evaporation. Expectedly, the most intensive evaporation processes occur when dry air is supplied to the inlet.

Figure 3 shows changes in the heat flux components at the interface. The following notations are used here:

- convective components of the heat flux at the interface for air and water film, respectively:

\[
q_{c,A} = -\lambda_A \left( \frac{\partial T_A}{\partial y} \right)_{y=\delta}, \quad q_{c,L} = -\lambda_L \left( \frac{\partial T_L}{\partial y} \right)_{y=\delta};
\]

- heat flux due to the latent heat of vaporization:

\[
q_j = \rho_A r D \left( \frac{\partial K_A}{\partial y} \right)_{y=\delta} = j_r r .
\]

The nature of the influence of the relative humidity of the input air on the change in the cross-flow of steam due to the evaporation process is also shown in Figure 3. The stage of cooling the liquid film to the adiabatic saturation temperature (the temperature of the "wet" thermometer) is completed at approximately \(x / H \approx 10\), after which the heat supplied to the liquid film by convection from the gas is completely spent on the phase transition. The figure also shows that an increase in relative humidity leads to a significant decrease in the intensity of evaporation processes. In the limit at \(\varphi_0 = 100\%\) the heat and mass exchange between the saturated vapor-gas mixture and the film stops. This is also evidenced by the calculated data on the value of the cross-flow of steam on the evaporating surface.

The results of numerical calculations shown in Figure 4 allow quantifying the effect of the Reynolds number \(\text{Re}_A\) and relative humidity on the average mass output in the channel from the water film surface. From the analysis of these data, it follows that the increase of \(\varphi_0\) leads to a decrease in the average values of the cross-flow of steam from the surface of the water film. The results also show that the dependences are similar for different Reynolds numbers \(\text{Re}_A\).
Figure 4. Influence of relative humidity of the input air and Reynolds number 

\( T_0 = T_{0,A} = T_{0,L} = 30^\circ C, \ Re_L = 0.5 \).

The change in the bulk temperature of the vapor-gas medium and the vapor concentration at the outlet of the channel (\( x/H = 50 \)) with variations in the input parameters is also shown in Figure 4. The values vary within \( \phi_0 = 0 \pm 100\% \) and \( Re_A = 50 \pm 1000 \). From these data it follows that increasing the Reynolds number leads to an increase in bulk temperature at the exit of the channel for the air medium due to the decrease in the intensity of the evaporation process, and as a consequence a decrease in bulk concentration \( k_{m1} \).

**Conclusion**

In this paper, heat and mass transfer in the horizontal channel has been numerically simulated during evaporation of the water film in the mode of forced laminar convection. On the basis of the performed computational studies, the following conclusions may be drawn:

- the physical and mathematical models of conjugated heat and mass transfer in a horizontal channel in a wide range of input parameters have been developed;
- on the basis of numerical calculations, the dependence of the main parameters of coolants, which include temperature-humidity and thermal-hydraulic characteristics, on the initial parameters has been revealed;
- the influence of high humidity of the input air and the Reynolds number on the overall level of vapor concentration and its distribution along the length of the channel has been quantitatively established. Dependences have been obtained at \( T_0 = T_{0,A} = T_{0,L} = 30^\circ C, \ Re_L = 0.5 \).

Assumingly, these trends will be observed at other air temperatures, but this finding requires additional research.

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