Journal of Engineering Physics and Thermophysics, Vol. 75, No. 5, 2002

INFLUENCE OF THE CONFIGURATION OF A NONUNIFORMLY COOLED PLATE ON THE INTENSITY OF THE PROCESS OF FILM CONDENSATION FROM A MOIST-AIR FLOW

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UDC 536.423.4:532.62

The intensity of the process of film condensation from a steam-gas flow as a function of the configuration of a nonuniformly cooled plate has been investigated. The range of the parameters of the process where the plate can be considered to be isothermal has been found.

Consideration is given to a flat plate of length L_X, width L_Z, and thickness $2\delta_w$ along which there is a flow of moist air. The state of the flow is determined by the quantities u_0, t_0, and ρ_0. The temperature of one (streamwise) edge of the plate is kept constant, while the remaining edges are heat-insulated (Fig. 1).

We have made the following assumptions on the character of the occurring processes:

(a) the plate is coated with a thin continuous liquid film;

(b) the thermal resistance of the liquid film is negligibly small;

(c) the temperature field in the plate is described by the approximation of a thermally thin wall;

(d) the thermophysical properties of the moist air and the condensate are independent of temperature.

With allowance for the assumptions made a change in the wall temperature can be described by the two-dimensional heat-conduction equation

$$\lambda_w \delta_w \left(\frac{\partial^2 t_w}{\partial x^2} + \frac{\partial^2 t_w}{\partial z^2} \right) + q = 0 . \tag{1}$$

The boundary conditions for Eq. (1) are

$$t_w (x, 0) = t_{w0} , \quad \left. \frac{\partial t_w}{\partial z} \right|_{z=L_Z} = \left. \frac{\partial t_w}{\partial x} \right|_{x=0} = \left. \frac{\partial t_w}{\partial x} \right|_{x=L_X} = 0 .$$

The heat flux q to the plate surface is made up of the heat of phase transition q_1 and the heat transferred by convection q_2:

$$q_1 = jr = h_D (\rho_0 - \rho_w) r , \tag{2}$$

$$q_2 = \alpha (t_0 - t_w) . \tag{3}$$

The coefficients of heat and mass exchange are calculated on the basis of the boundary-layer approximation from the formulas [1]

$$\alpha = 0.332\lambda \sqrt[3]{\mathrm{Pr}} \sqrt{\frac{u_0}{x\nu}} , \quad h_D = 0.332 D \sqrt[3]{\mathrm{Sc}} \sqrt{\frac{u_0}{x\nu}} . \tag{4}$$

Federal State Unitary Enterprise "M. V. Keldysh Research Center," Moscow, Russia. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 75, No. 5, pp. 17–20, September–October, 2002. Original article submitted February 20, 2002.

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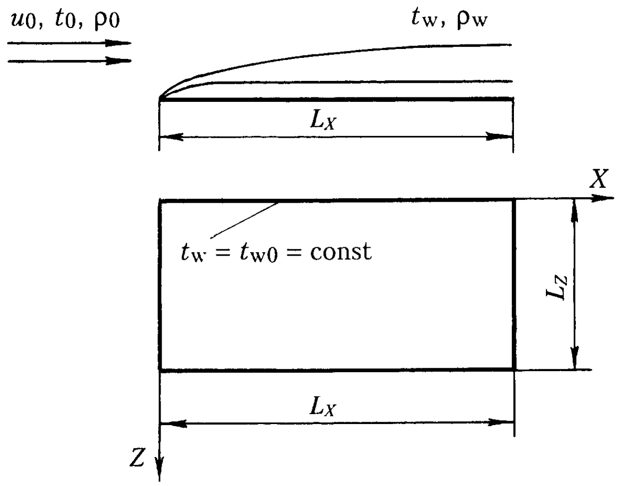

Fig. 1. Scheme of the computational region.

We represent the heat-conduction equation (1) in dimensionless form as

$$\left(\frac{\partial^2 T_w}{\partial X^2} + \frac{\partial^2 T_w}{\partial Z^2}\right) + A_2 \frac{1}{\sqrt{X}} (A_1 (1 - \bar{\rho}_w) + (1 - T_w)) = 0 , \tag{5}$$

where

$$A_1 = K_1 \,\mathrm{Le}^{2/3} ; \quad A_2 = 0.331 \frac{L_Z}{\delta_w} \frac{\lambda}{\lambda_w} \sqrt{\mathrm{Re}} \sqrt[3]{\mathrm{Pr}} ;$$

$K_1 = r/[(t_0 - t_{w0})\, c_p]$ is the phase-transition number and Le = Pr/Sc.

The partial density of the steam in the incoming flow is determined by the temperature and the moisture content of the air, and the density of the steam above the plate surface is equal to the density of a saturated steam at the plate temperature at a given point.

The integral flux of a mass condensing on the entire surface of the plate can be obtained by integration of the local flux of the condensate mass:

$$J = \int_0^{L_Z} \int_0^{L_X} j(x, z)\, dxdz = 0.262 D \sqrt{\frac{u_0}{\nu}} \int_0^{L_Z} \int_0^{L_X} \frac{\rho_0 - \rho_w}{\sqrt{x}}\, dxdz . \tag{6}$$

Equation (6) involves the difference of the densities of the steam in the flow and on the plate surface whose value determines the intensity of the process of mass exchange and whose sign determines the direction of the process. Thus, if the local temperature of the plate surface is below the dew point the process of condensation occurs; otherwise, the moisture on the surface is evaporated. We believe that if the plate temperature at any point is higher than the dew point, there is no moisture on the plate surface and no evaporation occurs.

The problem is solved numerically by the grid method, and the unsteady equation (5) is solved by the establishment method. At each instant of time, from the temperature field found, we calculate the values of the densities of a saturated steam above the plate surface from the saturation curve and the values of the mass flux at each point.

Using the procedure presented above, we have performed calculations to determine the dependence of the intensity of the condensate-mass flux on the geometric parameters of the plate. We consider as an example results of a calculation of the particular case of moist-air flow with the parameters u_0 = 1 m/sec and t_0 = 293 K about a copper plate of thickness δ_w = 0.001 m; the steam density in the flow is determined from the moisture content, which was prescribed to be equal to 100, 80, and 60%; the temperature on the cold edge of the plate was constant and equal to t_{w0} = 273 K.

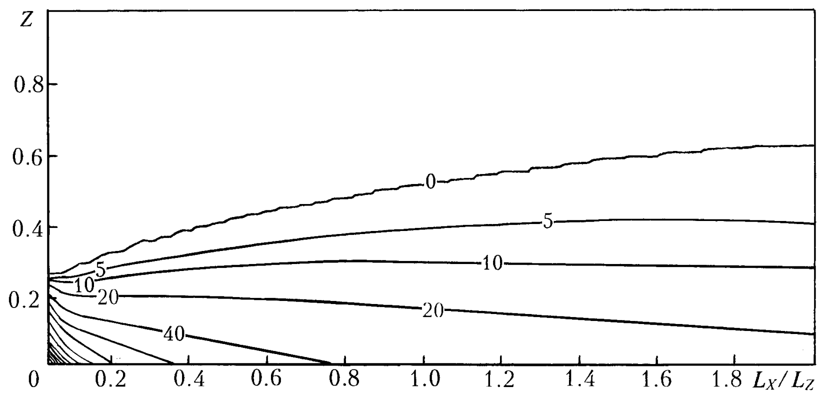

Fig. 2. Isolines of the condensate-mass flux (g/(m^2·h)). The calculation has been carried out for L_X = 3 m, L_Z = 1.5 m, T_0 = 293 K, T_{w0} = 273 K, and φ = 80%. The coordinates are dimensionless.

With the moisture content of the incoming flow lower than 100% on the plate surface there can exist portions where the local temperature is above the dew point. Here no condensation will occur and a part of the plate will remain dry. Figure 2 shows isolines of the flux of the condensate mass for the case of flow with a moisture content of 80% about the plate. The isoline of the zero mass flux corresponds to the boundary of the condensation zone. It is seen that most of the moisture is condensed on the portion of the plate located near the leading edge and adjacent to the cold edge. The reason is that the boundary layer has a small thickness near the leading edge and produces a low diffusion resistance, while the intense condensation near the cold edge is attributed to the large temperature head and, as a consequence, the large difference of the steam densities.

The presence of dry portions on the surface of the plate depends on its dimensions and thermophysical properties and flow parameters. We consider the influence of the elongation per unit length of the plate on the intensity of the process of condensation for a constant area of the plate. As the scale for the value of the total mass flux, we take the value of the mass flow rate of moisture which would pass through the cross section equal to the area of the plate for the prescribed properties of the incoming flow:

$$\bar{J}_\Sigma = \frac{J_\Sigma}{G} = \frac{J_\Sigma}{L_X L_Z u_0 \rho_0} .$$

When the area of the plate is fixed we have such a relation of its sides for which the flux of the condensate mass to the surface is minimum (Fig. 3a). The position of the maximum of the condensate flux is independent of the moisture content of the air. Since the function behaves monotonically, the results obtained can be approximated to a wider range of elongations.

The presence of the maximum of the condensate-mass flux can be explained by considering two limiting cases of the relation between the sides of the plate.

1. If the elongation $L_X/L_Z \to \infty$ (the length is much larger than the width) the entire plate will have the temperature of the cold edge and the temperature drop will be the same at each point while the boundary layer which produces the basic thermal resistance will increase along the plate length. The boundary-layer thickness will be so large at a distance from the leading edge that the process of condensation will cease, in practice. Since here condensation occurs on a portion constituting a small part of the entire area of the plate, this case corresponds to a small mass flux. When the plate is elongated the intensity of the condensation decreases.

2. The relation of the sides is such that the width is much larger than the length $L_X/L_Z \to 0$. The boundary layer has a small thickness and produces a low resistance at all points of the plate, while the temperature of the plate surface changes across the width and will become equal to the flux temperature at a distance from the cold edge. The boundary of the condensation zone will pass near this point. Since in this case the portion of the plate where the flux of the condensate mass is nonzero is smaller than the total area of the plate, we obtain that a nearly zero elongation of the plate corresponds to the minimum of the mass flux. Since the two limiting values of the elongation of the plate correspond to the minimum of the mass flux, we must have a maximum between them.

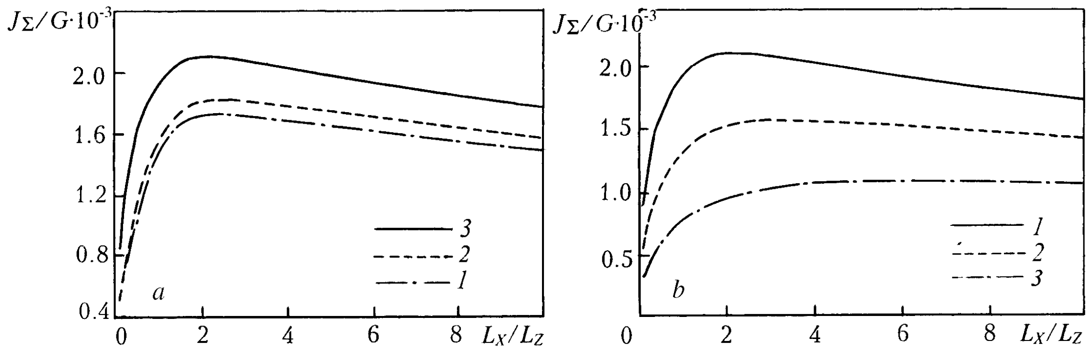

Fig. 3. Relative mass flux vs. elongation of the plate for different moisture contents of the incoming flow [(a) 1) φ = 75, 2) 80, and 3) 100%] and different areas of the plate [(b) 1) 0.1, 2) 0.2, and 3) 0.5 m^2].

With increase in the area the maximum of the flux of the condensate mass shifts to the right (Fig. 3b). The reason is that with increase in the plate width the surface temperature in the region at a distance from the cold edge approaches the flow temperature; consequently, the intensity of the process of condensation will be low.

The dependences obtained show that for the prescribed parameters of the flow and the area of the plate one can find the optimum relation of the sides that ensures the maximum condensate-mass flux for given conditions.

In the general case a spatially nonuniform temperature field is formed in the wall as a result of the heat influxes from the air flow and of the condensation. For certain values of the parameters of the problem in the steady regime the degree of nonuniformity is low and the wall can be considered to be isothermal. An attempt has been made to find the corresponding region of the values of these parameters. We select the maximum deviation of the wall temperature from the cold-edge temperature as the criterion of isothermality (selection of the value of the criterion is quite arbitrary). We assume that the wall can be considered to be isothermal if

$$T_w^{max} \le 0.1 \,. \tag{7}$$

We consider such a steady regime of condensation that the temperature of the plate surface at each point differs little from the temperature of the cold edge. In this case the relative temperature drop between the air flowing about the plate and the cold edge is rather large, and we can believe that the dimensionless values of the temperature of the plate surface T_w and of the steam density at the plate surface $\bar{\rho}_w$ are much less than unity. Then the heat-conduction equation for the plate in the steady regime will take the form

$$\left(\frac{\partial^2 T_w}{\partial X^2} + \frac{\partial^2 T_w}{\partial Z^2}\right) + A_2 \frac{1}{\sqrt{X}} (A_1 + 1) = 0 \,. \tag{8}$$

We introduce the variable $\theta_w = T_w/[A_2(A_1 + 1)]$ and transform (8):

$$\left(\frac{\partial^2 \theta_w}{\partial X^2} + \frac{\partial^2 \theta_w}{\partial Z^2}\right) + \frac{1}{\sqrt{X}} = 0 \,. \tag{9}$$

By solving Eq. (9) numerically we can find the distribution of the parameter θ_w in the plane x, z. Since the value of the deviation of the maximum value of the dimensionless plate temperature has been selected as the isothermality criterion, we will show the manner in which the parameter θ_w^{max} behaves. From Fig. 4 it is clear that the maximum value of θ_w changes as a function of the elongation of the plate. The surface temperature of the plate depends very strongly on the elongation for $L_X/L_Z < 1$. When the length of the plate becomes larger than the width $L_X/L_Z \ge 1$ the transfers of heat along the wall reduce temperature nonuniformities on the portion near the leading edge and θ_w depends weakly on the elongation of the plate and its value is close to unity.

For the case in question the criterion will be the expression

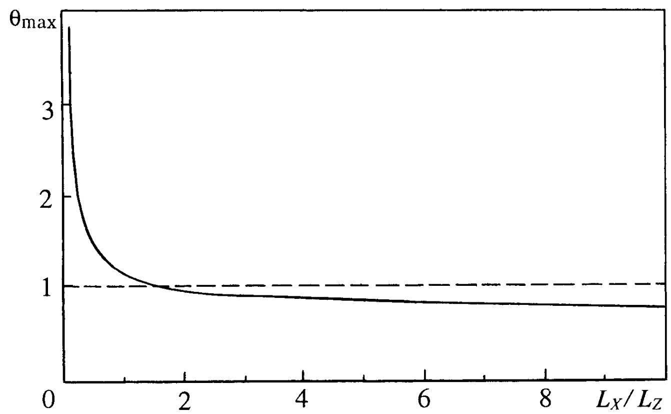

Fig. 4. Influence of the degree of elongation of the plate on its temperature regime. The coordinates are dimensionless.

$$A_2\,(A_1+1)\,\theta_w \le 0.1\,.$$

In the region where $L_X/L_Z \ge 1$, the plate is isothermal on condition that $A_1(A_1+1) \le 0.1$.

As a result of the investigation of the dependence of the condensation intensity on different parameters of the process we have obtained:

1) when the moisture content of the incoming flow is lower than 100% we can have regions on the plate surface where no condensation occurs;

2) for each fixed area of the plate we have such a length-to-width relation for which the flux of the condensate mass is maximum;

3) the relation of the sides of the plate for which the flux of the condensate mass is maximum is independent of the moisture content of the incoming flow;

4) transition of the plate to a regime in which it can be considered to be isothermal is determined by the value of the relative elongation of the plate and by the dimensionless parameter $A_2(A_1+1)$;

5) the range of parameters of the process in which the temperature nonuniformities in the plate can be considered to be small has been found.

NOTATION

u_0, longitudinal component of the velocity in the undisturbed flow; t_0, temperature of the undisturbed flow; t_w, local temperature of the plate surface; t_{w0}, temperature of the cold edge; $t_0 - t_{w0}$, characteristic temperature drop; ρ_0, concentration of the steam in the undisturbed flow; ρ_w, concentration of the steam near the cold wall; j, mass flux of the condensate formed on the film surface in a unit time on a unit area; φ, moisture content of the incoming flow; h_D, mass-transfer coefficient; α, heat-exchange coefficient; r, heat of phase transition; ν, coefficient of kinematic viscosity of the air; λ, thermal conductivity of the air; D, diffusion coefficient; q, local heat flux to the wall; c_p and λ_w, heat capacity and thermal conductivity of the wall material; δ_w, thickness of the wall; L_Z, width of the plate, characteristic dimension; L_X, length of the plate; Re = u_0L_Z/ν, Reynolds number; Pr = ν/a, Prandtl number; Sc = ν/D, Schmidt number; $T_w = (t_w - t_{w0})/(t_\infty - T_{w0})$, dimensionless increase in the temperature on the plate surface; $\overline{\rho}_w = \rho_w/\rho_0$, dimensionless concentration of the steam above the plate surface. Subscripts: max, maximum; w, variables with this subscript correspond to the values on the wall (condensation surface).

REFERENCES

1. E. R. G. Eckert and R. Drake, *Heat and Mass Transfer Theory* [Russian translation], Moscow (1961).