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EXPERIMENTS ON HEAT TRANSFER OF A SUPERSONIC GAS FLOW IN A CIRCULAR TUBE AT HIGH TEMPERATURE GRADIENTS

V. A. Mukhin, A. S. Sukomel, and V. I. Velichko

An experimental investigation was made of heat transfer during the flow of a compressible gas in a cooled circular tube at Mach number values from 0.1 to 4 and $T_{a.c}/T_c$ from 1 to 3.1. **A generalization** of the experimental data is given.

Most studies devoted to heat transfer in a flow of a compressible gas in tubes pertain to the case of gas motion at subsonic velocities. Only in a few investigations were heat transfer and drag studied during the flow of a supersonic gas [4-6] and in most of these, the measurements were carried out at small temperature gradients. We made an experimental investigation of heat transfer during supersonic air flow in a circular tube at large temperature gradients.

Local heat transfer was measured by the thick-walled tube method. The experimental heat exchanger was a thick-walled brass tube 575 mm long, 20.2 mm in inside diameter and 87 mm in outside diameter. The air flowed through the tube, the outside of the tube was bathed with water, which moved through the annular gap between the casing of the heat exchanger and the outer surface of the tube.

The temperature close to the inner surface of the tube was measured at 20 points, and the temperature of the outer surface, at 6 points. Thermocouples, which were used to measure the temperature close to the inner surface, were placed in radial drillings 1.5 mm in diameter. The beads of the thermocouples were soldered to the bottom of the drilling by condensor welding. Openings with a diameter of 0.25 mm were made at 17 points along the length in order to sample the static pressure.

The air entered the experimental section from the stagnation chamber through interchangeable nozzles. The experiments were carried out with five nozzles, of which one was subsonic and four were supersonic, calculated for Mach number values of 2.5, 3.0, 3.5, and 4.0.

After preliminary dust, oil, and moisture removal, the air passed through a normal orifice which served to determine the discharge. Further, the air entered an electric shaft furnace where it was heated to the assigned temperature (200-800°C), and then flowed through the chamber in which the flow stagnation parameters were measured. A heater was wound on the wall of the stagnation chamber and a grid mixer was installed at the chamber entrance. This enabled us to determine the stagnation temperature with a sufficient degree of accuracy. The stagnation temperature was measured by a shielded chromel-alumel thermocouple and the stagnation pressure by a class 0.35 standard manometer.

The local density of the heat flux at the tube wall (excluding a small area near the entrance about 5 diameters long) was determined under the assumption that the temperature field in the wall was one-dimensional, i.e., by the formula

$$q_c = A \Delta t_c, \qquad A = \frac{\lambda}{R_0 \ln R_1/R_2}. \tag{1}$$

The heat conduction of the wall was determined experimentally for each thermocouple.

The flow parameters in the given section (velocity coefficient λ and flow temperature T) were found on the basis of a one-dimensional flow model. The stagnation temperature T_0 was determined from the known stagnation temperature at the entrance and from the distribution of the heat fluxes in the wall from the entrance to the section under consideration. The heat-transfer coefficient was calculated by the formula

$$\alpha = \frac{q_c}{T_{a.c} - T_c}, \qquad (2)$$

where

$$\begin{gathered} T_{a.c} = T_0\left[1 - \frac{k-1}{k+1}(1-r)\lambda^2\right], \\ r = \sqrt[3]{\mathrm{Pr}^*} - 7{,}16\cdot 10^{-5} f\left(\frac{x}{D}\right)\mathrm{Re}_x^{0,4}; \\ f\left(\frac{x}{D}\right) = 1 \quad \text{for } \frac{x}{D} = 0\div 15; \\ f\left(\frac{x}{D}\right) = 1 + 0{,}0413\left(\frac{x}{D} - 15\right) \quad \text{for } \frac{x}{D} = 15\div 27; \\ T^* = T + 0{,}5\,(T_c - T) + 0{,}22\,(T_{a.c} - T). \end{gathered} \qquad (3)$$

Formula (3) was derived from the generalization of the experimental data [3]. All physical properties of the air were determined from the thermodynamic temperature of the flow at the given section.

The experiments were carried out at Reynolds number values from $0.4 \cdot 10^5$ to $7.0 \cdot 10^5$ and Mach number values at the tube entrance from 0.1 to 4.0, the entrance temperature varied from 200 to 800°C.

An analysis of the experimental data showed that the velocity and pressure distribution over the tube length mainly corresponded to the one-dimensional flow model. In the experiments with supersonic velocities, we observed oblique shocks which led to a certain disturbance in the smooth change of velocity and pressure over the tube length. The magnitude of the oblique shocks mainly depended on how well the nozzle was coupled with the experimental tube. Therefore, we took measures to couple these units as best as possible.

The character of the lengthwise change of the local heat-transfer coefficients showed that under certain conditions there is an area at the start of the tube which is occupied by laminar and transition boundary layers. We can trace that the length of this area is retained as the Re number increases. Hereafter we will examine only points corresponding to a developed turbulent motion and to complete thermal stabilization of the flow.

At first we carried out a series of experiments with low subsonic velocities at low air temperatures at the inlet (200-300°C).

The results of this series of experiments agree, within an accuracy to 5% with the formula of M. A. Mikheyev for heat transfer in turbulent motion of an incompressible fluid in tubes. After this, we carried out experiments at high temperatures in a subsonic nozzle and experiments on supersonic nozzles both at low and high temperatures.

Figure 1, which was calculated by means of parameter τ, illustrates the effect of gas compressibility on heat transfer:

$$\tau = \frac{T}{T_0} = \left(1 - \frac{k-1}{k+1}\lambda^2\right).$$

It was assumed here that the dependence of heat transfer on the Reynolds and Prandtl numbers is the same as for fluid incompressibility: $Nu \sim Re^{0,8}Pr^{0,43}$.

As we see from Fig. 1, the Nusselt number varies in proportion to $\tau^{0,42}$. Thus the dependence of the Nusselt number on τ was somewhat greater than in Petukhov's study [6], which led to a difference in Nusselt number values which did not exceed 10%. This small divergence can be explained by the different experimental conditions in these studies (different pressure gradients, temperature gradients, exponent in the temperature dependence of viscosity).

Figure 2 confirms the usual dependence of heat transfer on the Reynolds number.

As a result of treating the experimental data, we obtained the dependence

$$\mathrm{Nu} = 0{,}022\mathrm{Re}^{0{,}8}\mathrm{Pr}^{0{,}43}\tau^{0{,}42}. \tag{4}$$

The formula is valid for cooling of a turbulent gas flow in a tube in the section having thermal stabilization at values of

$$\mathrm{Re} = (0{,}4 \div 7{,}0)\cdot 10^5, \quad M = 0{,}1 \div 3{,}0^{*)}, \quad \theta = \frac{T_{\text{a.c}}}{T_{\text{c}}} = 1 \div 3{,}1.$$

The scatter of the experimental points relative to the curve corresponding to formula (4) does not exceed ±10%, which is within the accuracy of the experimental data.

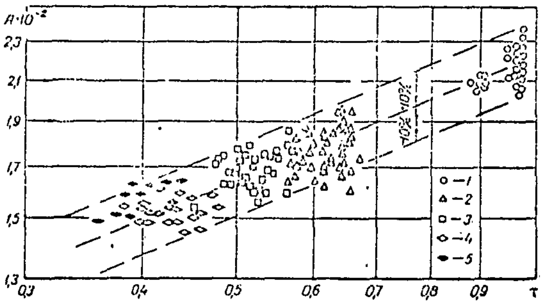

Fig. 1. Effect of gas compressibility on heat transfer $A = \frac{\mathrm{Nu}}{\mathrm{Re}^{0,8}\mathrm{Pr}^{0,43}}$: 1) $M_{in} < 1$; 2, 3, 4, 5) M_{in} respectively equal 2.5, 3.0, 3.5, 4.0.

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* The Mach number at the inlet varied from 0.1 to 4.

Thus, the results of the investigation show that within the accuracy of the experiments, the Nusselt number does not depend on the temperature factor in spite of the fact that the latter varies in wide limits (from 1 to 3.1). Of course, this conclusion is valid only if the gas flow is cooled, and is in conformity with the results obtained in earlier studies [1, 7].

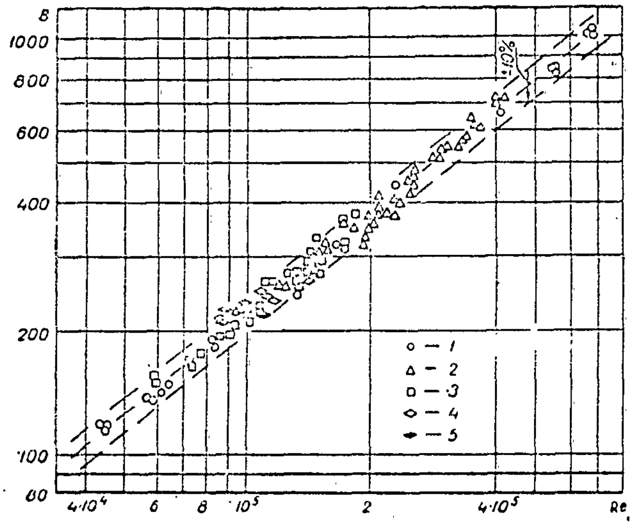

Fig. 2. Dependence of the Reynolds (designations are the same as in Fig. 1), $B = \frac{Nu}{Pr^{0,43} \tau^{0,42}}$.

DESIGNATIONS

Δt_c is the local temperature drop at the tube wall; A is the wall heat conduction; R_0, R_1, R_2 are the radius of the tube and radii of the hot junction inserts of the inside and outside thermocouples; T_c is the wall temperature; $T_{a.c.}$ is the adiabatic wall temperature; $\underline{r}$ is the temperature recovery coefficient; Pr^* is the Prandtl number value taken at a certain temperature T^* calculated by Eckert's method.

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SOLUTION OF AN EQUATION OF GAS MOTION IN A RECTANGULAR CHANNEL WITH FRICTION AND HEAT TRANSFER

Yu. I. Danilov and B. M. Galitseyskiy

A method is given for the solution of a differential equation of gas motion in a rectangular channel under the combined influence of friction and heat transfer.

As a gas moves along a channel, the gas parameters at a given section depend on the friction factor, on the starting parameters of the gas in front of the channel, and also on the law and intensity of the heat effect.

We will assume that the law and the form of the heat effect are already given, therefore the stagnation temperature distribution over the length of the channel is easily determined from the equation of thermal equilibrium

$$dq = \rho wFd(C_pT). \qquad (1)$$

The equation of a uniform steady-state gas flow in a rectangular channel in the presence of friction and heat transfer has the following form:

$$\frac{d}{dx}[P+\rho w^2] + \xi\frac{\rho w^2}{2D} = 0. \qquad (2)$$

Let us change to the parameters of stagnation and the gasdynamic functions

$$P + \rho w^2 = \rho w z \lambda \left(2\frac{k+1}{k} gRT \right)^{1/2}; \quad \rho w^2 = \rho w \lambda \left(\frac{2k}{k+1} gRT \right)^{1/2}. \tag{3}$$

Then Eq. (2) is written:

$$\frac{d}{dx}\left[z(\lambda)\sqrt{T}\right] + \frac{1}{2}\frac{k}{k+1}\xi\frac{1}{D}\lambda\sqrt{T} = 0. \tag{4}$$

To solve Eq. (4) we will make a change of variables

$$\xi\frac{1}{D}\frac{k}{k+1}\lambda\sqrt{T} = -\frac{du}{dx}, \tag{5}$$

where u is the unknown function subject to determination.

Then Eq. (4) is transformed:

$$\left(\frac{du}{dx}\right)^2 + m(u+c)\frac{du}{dx} + m^2T = 0, \qquad m = \xi\frac{1}{D}\frac{k}{k+1}. \tag{6}$$

The constant of integration c can be assumed equal to zero, since the change of variable r = v - c reduces Eq. (6) to the form

$$\left(\frac{dv}{dx}\right)^2 + mv\frac{dv}{dx} + m^2T = 0.$$

Equation (6) at a constant value of m and a linear change of the stagnation temperature over the channel length is easily integrated. Actually, let

$$T = T_1 + \frac{T_2 - T_1}{l}x, \tag{7}$$

where T_1 and T_2 are respectively the stagnation temperature at the channel inlet and exit. Then Eq. (6) is written:

$$\left(\frac{du}{dx}\right)^2 + mu\frac{du}{dx} + m^2\left(T_1 + \frac{T_2 - T_1}{l}x\right) = 0. \tag{8}$$

This equation will be integrated by introducing a parameter. Let us designate $\frac{du}{dx} = z$, then

$$z^2 + muz + m^2\left(T_1 + \frac{T_2 - T_1}{l}x\right) = 0. \tag{9}$$

Differentiating Eq. (9) in terms of u, we derive after appropriate transformations a first-order linear equation

$$\frac{du}{dz} + \frac{zu}{z^2 + m\frac{T_2 - T_1}{l}} + \frac{2z^2}{m\left(z^2 + m\frac{T_2 - T_1}{l}\right)} = 0. \tag{10}$$

The solution of this equation is written in the form

$$u = -\frac{z}{m} + \left[\frac{T_2 - T_1}{l}\sinh^{-1} z\left(m\frac{T_2 - T_1}{l}\right)^{-1/2}\right]\left(z^2 + m\frac{T_2 - T_1}{l}\right)^{-1/2} - \\ - c\left(z^2 + m\frac{T_2 - T_1}{l}\right)^{-1/2}. \tag{11}$$

Eliminating u from equations (9) and (11), we obtain an equation relative to the parameter z:

$$-\frac{mT}{z} = \left[\frac{T_2 - T_1}{l}\sinh^{-1} z\left(m\frac{T_2 - T_1}{l}\right)^{-1/2}\right]\left(z^2 + m\frac{T_2 - T_1}{l}\right)^{-1/2} - \\ - c\left(z^2 + m\frac{T_2 - T_1}{l}\right)^{-1/2}. \tag{12}$$

We will determine the constant of integration c from the condition $z = z_1$ when $T = T_1$, then Eq. (12) is transformed to

$$\frac{mT}{z}\sqrt{z^2 + m\frac{T_2 - T_1}{l}} + \frac{T_2 - T_1}{l}\sinh^{-1} z\left(m\frac{T_2 - T_1}{l}\right)^{-1/2} = \\ = \frac{mT_1}{z}\sqrt{z_1^2 + m\frac{T_2 - T_1}{l}} + \frac{T_2 - T_1}{l}\sinh^{-1} z_1\left(m\frac{T_2 - T_1}{l}\right)^{-1/2}. \tag{13}$$

Since the parameter z is uniquely associated with λ and T: $z = \frac{du}{dx} = -m\lambda\sqrt{T}$, then substituting its value into Eq. (13), we obtain the following expression:

$$\sqrt{A_1^4 + \frac{A_1^2}{\lambda_1^2}} + \sinh^{-1}\lambda_1 A_1 = \sqrt{A_2^4 + \frac{A_2^2}{\lambda_2^2}} + \sinh^{-1}\lambda_2 A_2, \tag{14}$$

where

$$A_1^2 = \xi \frac{k}{k+1} \frac{l}{D} \left(\frac{T_2}{T_1} - 1 \right)^{-1},$$

$$A_2^2 = \xi \frac{k}{k+1} \frac{l}{D} \left(1 - \frac{T_1}{T_2} \right)^{-1},$$

or, designating

$$\sqrt{A^4 + \frac{A^2}{\lambda^2}} + \sinh^{-1} \lambda A = \Phi(\lambda, A),$$

finally for determining the reduced velocity in the channel we will have (Fig. 1)

$$\Phi(\lambda_1, A_1) = \Phi(\lambda_2, A_2). \qquad (16)$$

Analogously for the case of gas cooling according to the linear law, the function

$$\Phi_1(A, \lambda) = \sqrt{\frac{A^2}{\lambda^2} - A^4} + \text{Arc sine } \lambda \ A \qquad (17)$$

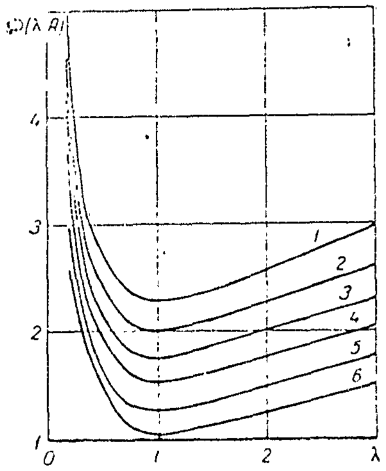

Fig. 1. Dependence of Φ on λ: 1, 2, 3, 4, 5, 6) Values of A equal respectively to 1, 0.9, 0.8, 0.7, 0.6, and 0.5.

Knowing the magnitude of the reduced velocity, we easily determine all remaining gas parameters (static temperature, pressure, density) by using the gasdynamic functions. For example, the pressure losses in the channel will equal:

$$\Delta P = P_1 - P_2 = \frac{\rho w}{m_{cr}}\left[\frac{\sqrt{gRT_1}}{y(\lambda_1)} - \frac{\sqrt{gRT_2}}{y(\lambda_2)}\right],$$

where

$$m_{cr} = \sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}};$$

$$y(\lambda) = \left(\frac{k+1}{2}\right)^{\frac{1}{k-1}}\lambda\left(1 - \frac{k-1}{k+1}\lambda^2\right)^{-1}.$$

When the change of stagnation temperature over the channel length differs from the linear law and the friction factor varies appreciably of the channel length, we recommend the method of dividing the channel into a number of segments in which we can assume approximately a constant value of the friction factor ξ and a linear rise of the stagnation temperature. Thus, we can calculate with any degree of accuracy a channel with any predetermined law of heat transfer.

For sufficiently small values $\lambda \ll 1$, the function $z(\lambda) \approx \frac{1}{2\lambda}$ and Eq. (6) will have the form

$$mu\frac{du}{dx} + m^2T = 0. \tag{18}$$

Solving this equation, we derive the following expressions for determining the reduced velocity at the channel exit relative to the reduced velocity at the channel inlet:

$$\lambda_2 = \sqrt{T_2\left(\frac{T_1}{\lambda_1^2} - 2\int_0^l mTdx\right)^{-1}} \tag{19}$$

When the magnitude $\lambda \gg 1$, function $z(\lambda)$ can be assumed approximately equal to $z(\lambda) \approx \frac{1}{2}\lambda$. Then Eq. (6) is transformed as

$$\frac{du}{dx} + mu = 0. \tag{20}$$

Its solution is written

$$\lambda_2 = \lambda_1 \sqrt{\frac{T_1}{T_2}} \exp\left[-\int_0^l m\,dx\right]. \tag{21}$$

Formulas (19) and (21) are derived in the case of gas flow in a channel with a heat conductor and respectively reflect the character of subsonic and supersonic flow wherein when we derived them we did not permit the assumption of a linear change of stagnation temperature over the channel length, i.e., they are valid for any law of a heat conductor. However, upon using the obtained relationships in practice, it is necessary to take into account the specific conditions of the problem and the required accuracy of the calculations.

The function $\Phi(\lambda, A)$ depends on the two variables A and λ and has a minimum at $\lambda = 1$. Having tabulated the function $\Phi(\lambda, A)$ and compiled a table for a wide range of values of λ and A, we can fairly easily and rapidly determine the distribution of the reduced velocity λ over the channel length and, consequently, all other parameters for any predetermined law of heat exchange.

DESIGNATIONS

q is the heat supplied to the gas in the channel; F is the flow area of the channel; C_p is the heat capacity of the gas at constant pressure; T is the gas stagnation temperature; ρ is the density; w is velocity; x is the abscissa readable along the channel axis from the channel inlet; P is static pressure; ξ is the friction factor; D is the hydraulic diameter; k is the index of the adiabatic curve; λ is the reduced velocity; R is the gas constant; c is the constant of integration; l is the channel length.

HEAT AND MASS TRANSFER IN EVAPORATION PROCESSES

A. V. Lykov

The effect of a crossflow of a substance on heat transfer in a laminar flow past a moist capillary-porous plate is analyzed. Regularities in heat and mass transfer during liquid evaporation from capillary-porous bodies are established. It is shown that in the case of deepening the evaporation surface, the heat-transfer coefficients are larger than those in evaporation at the surface of a body.

Heat and mass transfer in liquid evaporation from an exposed surface and from capillary-porous bodies is not only of theoretical interest, but also of practical value for engineering.

Heat and mass transfer between the surface of a liquid and the ambient medium (moist air) is a single mutually associated process having its own characteristics which differ from a heat transfer process complicated by mass transfer. An even more complex process is heat and mass transfer of a moist capillary-porous body with the flow of a heated gas. In this case, heat and mass transfer of the body surface with the ambient medium is continuously associated with heat and mass transfer inside the capillary-porous body (the interdependence of the external and internal problems). Such processes of heat and mass transfer include those of drying and transpiration cooling by means of liquid evaporation. Heretofore there has been no single opinion not only on the physical mechanism of transfer, but also on the qualitative effect of mass transfer on heat transfer. A number of investigators, based on the theory of gas injection into the boundary layer through a porous plate, consider that in liquid evaporation the heat-transfer coefficient decreases with an increase in evaporation intensity, whereas in condensation of vapor, conversely, the heat-transfer coefficient increases with an increase of condensation intensity.* In the first case the crossflow of a substance is directed to the side opposite the heat flow and in the second case these flows have the same direction. Then the boundary-layer thickness in evaporation increases (the boundary layer "swells"), which leads to a reduction of the heat-transfer coefficient. During condensation the reverse picture occurs, which leads to an increase of the heat-transfer coefficient with an increase of condensation intensity.

However, the experimental investigations of A. V. Nesterenko [1], G. T. Sergeyev [2], and of other investigators on evaporation of various liquids from an exposed surface showed that the heat-transfer coefficients in evaporation are larger than those without mass exchange (dry heat transfer) under the same hydrodynamic conditions and temperature differences. This difference increases with an increase of the relative humidity of the air. When drying moist materials, the heat-transfer coefficients are larger than the corresponding coefficients of a dry body [3-6]. During transpiration cooling, when the fluid is continuously fed into a capillary-porous body, we have different results for different investigators. In some investigations [7] the

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* Here it is assumed that the heat needed for evaporation is mainly transmitted by convection from the heater air.

heat-transfer coefficients increase with an increase of the evaporation intensity and in others, conversely, they decrease [8, 9]. The results, obtained experimentally, are of considerable interest. In our article we will explain them on the basis of the theory of liquid evaporation from capillary-porous bodies.

Transpiration Cooling in Gas Injection into the Boundary Layer

A calculation of transpiration cooling by the method of boundary-layer injection through porous walls was studied in greatest detail by E. Eckert [10]. It is based on a solution of a system of differential equations of heat and mass transfer for a laminar boundary layer with flow of a binary gas mixture past a flat, porous plate. The system of differential equations has the form:

$$\frac{\partial w_x}{\partial x} + \frac{\partial w_y}{\partial y} = 0, \tag{1}$$

$$\rho w_x \frac{\partial w_x}{\partial x} + \rho w_y \frac{\partial w_x}{\partial y} = \frac{\partial}{\partial y}\left(\eta \frac{\partial w_x}{\partial y}\right), \tag{2}$$

$$\begin{aligned} c_p \rho w_x \frac{\partial t}{\partial x} + c_p \rho w_y \frac{\partial t}{\partial y} = \frac{\partial}{\partial y}\left(\lambda \frac{\partial t}{\partial y}\right) + \\ + \rho D (c_{p2} - c_{p1}) \frac{\partial \rho_{10}}{\partial y} \frac{\partial t}{\partial y}, \end{aligned} \tag{3}$$

$$\rho w_x \frac{\partial \rho_{10}}{\partial x} + \rho w_y \frac{\partial \rho_{10}}{\partial y} = \frac{\partial}{\partial y}\left(\rho D \frac{\partial \rho_{10}}{\partial y}\right). \tag{4}$$

Thermal diffusion (Soret effect) and the diffusion thermo effect (Dufour effect) are disregarded as small magnitudes.

The following boundary conditions were taken:

$$\text{when } y = 0 \quad w_x = 0, \; w_y = w_s, \; t = t_s, \; \rho_{10} = \rho_{10s}; \tag{5}$$

$$\text{when } y \to \infty \quad w_x = w_c, \; t = t_c, \; \rho_{10} = \rho_{10c}. \tag{6}$$

In addition, it was assumed that the linear transverse velocity of the change along the surface w_s (in the x-direction) is inversely proportional to $\sqrt{x}$.

The results of the calculations for the case where the mass flow of the injected gas is directed from the body surface (analog of the evaporation process) are shown in Fig. 1. We see from this figure that the heat- and mass-transfer coefficients decrease with an increase of the parameter

$$Z = \frac{w_s}{w_c} \sqrt{\mathrm{Re}_x} = \frac{j_1}{\rho w_c} \sqrt{\mathrm{Re}_x}, \tag{7}$$

where $j_1 = \rho w_s$ is the intensity of mass transfer in a direction normal to the wall surface (the intensity of the mass crossflow of a substance).

Consequently the heat-transfer coefficient α decreases with an increase of intensity of the crossflow of the substance. The ratio $\alpha/\alpha_0 = 0.9$ when $Z = 0.05$ (Fig. 1). Consequently, when $Z = 0.05$ the decrease in the heat-transfer coefficient is 10%. A twofold decrease ($\alpha/\alpha_0 = 0.5$) corresponds to a value of $Z \approx 0.3$. In the study by Shulman [9] with transverse flow past a hollow, porous ceramic cylinder, inside which water was fed to certain regions, the parameter Z varied from 0.015 to 0.05. The evaporation intensity was from 3.6 to 18.6 $\mathrm{kg \cdot m^{-2} \cdot hr^{-1}}$ in the Reynolds number range from $2 \cdot 10^4$ to $8 \cdot 10^4$ and with an air temperature change from 70 to 130°C. Consequently, a decrease in the heat-transfer coefficient caused by crossflow was less than 10%, i.e., it lies within the accuracy of the experiment. Moreover, a decrease in the local heat-transfer coefficient was noted upon an increase in the evaporation intensity by a factor of

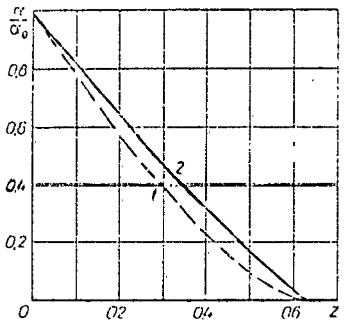

Fig. 1. Dependence of relative heat-transfer coefficient (α/α_0) on parameter Z based on Eckert's data [10]: 1) Pr = = Sc = 1; 2) Pr = = Sc = 0.7.

1.5-1.9, i.e., by 50-90%. If we take into account that the process of liquid evaporation is analogous to the process of blowing air into the boundary layer, then under conditions of adiabatic evaporation the parameter Z characterizing the effect of crossflow of a substance on heat and mass transfer can be presented in a different form. The intensity of evaporation is

$$j_1 = \frac{\alpha_x}{r}(t_c - t_M) = \frac{\lambda}{x}\mathrm{Nu}_x(t_c - t_M). \tag{8}$$

In a laminar flow of moist air past a plate $\mathrm{Nu}_x = A_x \sqrt{\mathrm{Re}_x}$, where A_x is a constant. Then the parameter

$$Z = \frac{A_x}{\mathrm{Pr}} \frac{c_p \Delta t}{r}. \tag{9}$$

Consequently, Z does not depend on the velocity of air motion, but depends on the psychrometric difference.

Let us make an approximate calculation. For Z = 0.05 the temperature difference Δt will be $\Delta t = 260°C$. In the calculation we assumed $t_M = 30°C$, $r = 579$ kcal $\cdot$ kg^{-1}, $A_x = 0.33$. Consequently, only at a temperature difference of $\Delta t = 260°$ will the reduction of the heat-transfer coefficient be of the order of 10%.

Similar results were obtained earlier [8] with the flow of hot air past a porous plate. The evaporation intensity in these experiments did not exceed 22.7 kg $\cdot$ m^{-2} $\cdot$ hr^{-1}. The Reynolds number varied from 10^4 to $2 \cdot 10^6$, and the air temperature from 18 to 140°. Evaporation was under adiabatic conditions. The temperature difference did not exceed 79°. Consequently, crossflow of a substance cannot render a perceptible effect on the reduction of the heat-transfer coefficient. However, the authors established that there is a dependence for the average Nusselt number:

$$\mathrm{Nu} = 0{,}00455\,\mathrm{Re}^{0{,}8}\left(\frac{r}{c_p \Delta t}\right)^{0{,}4}. \tag{10}$$

Therefore, the reduction of the heat-transfer coefficient was more than threefold.

Although this analysis of the study [8] is of a somewhat arbitrary nature since evaporation of water from a porous plate occurred in a turbulent flow, the decrease in the heat-transfer coefficient with an increase of the temperature difference Δt is the result of other causes. We must point out that in a turbulent flow, the effect of crossflow of a substance on the change of the heat-transfer coefficient will be less than in a laminar flow.

Drying of moist materials in convective driers is done approximately under adiabatic conditions. The temperature differences in hot-air drying is considerably lower than 250° in the overwhelming majority of cases. Therefore, crossflow of a substance through the boundary layer produced by moisture evaporation has virtually no effect on the value of the heat-transfer coefficient. Numerous experiments established that at a constant drying rate the heat-transfer coefficient is considerably larger than in heat transfer of a dry body, all other conditions being equal. Starting at the critical moisture content, the heat-transfer coefficient decreases during the course of drying time, gradually approaching the value of the heat-transfer coefficient of a dry body. Consequently, the heat-transfer coefficient decreases with a decrease in the drying rate.

Liquid Evaporation from an Exposed Surface

In liquid evaporation from an exposed surface, the heat-transfer coefficient increases with an increase of evaporation intensity. This effect of the evaporation process on heat and mass transfer is characterized by the Gukhman number which is the thermodynamic criterion of evaporation.

The calculation formulas in a criterial form are:

$$Nu = A Pr^{0.33} Re^{n} Gu^{m}, \tag{11}$$

$$Nu' = A' Sc^{0.33} Re^{n'} Gu^{m'}, \tag{12}$$

where A, A', n, n', m, m' are constants, determinable by experiment, dependent on the hydrodynamic flow conditions (Reynolds number range). For example, according to the data of G. T. Sergeyev [2], in the Reynolds number range from $4 \cdot 10^4$ to $16 \cdot 10^4$, the constants in formulas (11) and (12) are equal to: $A = 0.086$; $A' = 0.094$; $n = n' = 0.8$; $m = m' = 0.2$.

The most probable explanation of the evaporation effect on the intensity of heat and mass transfer is the hypothesis of volume evaporation associated with the dynamic character of the processes of sorption and desorption.

The essence of this hypothesis is that minute liquid droplets impinge upon the boundary layer. The main cause of the separation of droplets from an exposed surface of a liquid is the presence of the processes of focal condensation and the interaction of the gas flow with the liquid surface. According to de Bour's dynamic theory of adsorption [11], the evaporation process is a dynamic process of desorption and sorption. The molecules of the liquid not only leave the surface (evaporation) but are continuously returned (condensation). The evaporation rate is proportional to the difference of the molecular flows leaving and returning to the liquid surface. The investigations of N. N. Fedyakin [12] showed that condensation does not occur uniformly along the surface, but on certain sections, and incomplete wetting of the liquid surface by the adsorbed layer of liquified vapor takes place. In the condensation regions drops are formed which, being less strongly bound with the liquid, are carried away by the air flow to the boundary layer.

The evaporation of drops in the boundary layer we will call volume evaporation, it is a volume source of vapor and a negative source of heat in boundary-layer equations. In the presence of volume evaporation, the right-hand part of Eq. (3) will contain a third term equal to rI, where I is the volume capacity of the vapor source ($kg \cdot m^{-3} \cdot hr$). The right-hand part of Eq. (4) must be supplemented by the magnitude of the positive vapor source (I).

By methods of the similarity theory, we will find from Eq. (3) the dimensionless variable

$$K_i = \frac{rIl^2}{\lambda T_c}. \tag{13}$$

The magnitude rI equals the amount of heat which is needed for volume evaporation. If we designate in terms of N_v the number of drops per unit volume of the boundary layer, and $\bar{R}$ is the mean radius of a drop, then an elementary calculation leads to the relationship

$$K_i = 8\pi \bar{R} N_v l^2 \frac{\Delta t}{T_c}. \tag{14}$$

The magnitude $8\pi\bar{R}N_v l^2$ depends on the physical properties of the liquid and the hydrodynamics of the flow.

The temperature difference $\Delta t = [t(x, y) - t_m]$ varies in the direction of the x- and y-coordinates. The relative value $\Delta t/T_c$ characterizes the local thermodynamic intensity of evaporation. The maximal value of this magnitude equals the Gukhman number ($Gu = (T_c - T_m)/T_c$). From this point of view, Gu characterizes the potential possibility of humid air in volume evaporation. The hypothesis of volume evaporation is in need of confirmation by direct experiments.

The second cause of the intensification of the heat-transfer process in evaporation is the disturbance in the near-wall boundary layer by focal evaporation processes. In liquid evaporation at ordinary temperature and pressure, the volume of the substance increases by about 10^3 times, and on condensation of the vapor the same decrease in volume occurs. As a result of focal processes of evaporation and condensation the structure of the laminar boundary layer is disturbed, which leads to an intensification of heat and mass transfer. This effect is especially demonstrated upon evaporation in a vacuum, when the change of volume in phase transformations reaches an order of 10^6. This leads to an increase in the heat-transfer coefficient by about one order.

The hypothesis of stream heat and mass transfer in evaporation in a vacuum was developed by A. A. Gukhman [13]. It is completely natural that these effects intensifying heat and mass transfer take place in evaporation under conditions of ordinary barometric pressure although their influence will be appreciably smaller.

When drying moist materials the effect of volume evaporation on the heat- and mass-transfer process will be smaller as compared with liquid evaporation from a free surface. However, the intensification of heat transfer by the effect of focal evaporation will evidently be greater. This is because sorption and desorption processes take place on the surface of macro- and microcapillaries in capillary-porous bodies. It is important to note here that the external heat and mass transfer of moist materials is continuously associated with physicochemical processes on the surface of a capillary-porous body. Deepening of the evaporation surface is the principal factor influencing heat and mass transfer between moist capillary-porous bodies and hot gas. Thus heat and mass transfer of a body surface with the ambient medium should be considered as a combination of heat and mass transfer in the boundary layer of humid air and in the boundary layer of a capillary-porous body (evaporation zone).

Liquid Evaporation from Capillary-Porous Bodies

To analyze the complex problem of heat and mass transfer in evaporative transpiration cooling, we will use O. Krisher's method [14]. The essence of this method is as follows.

If we neglect the effect of a crossflow of a substance, the differential equation of heat transfer in the boundary layer with laminar flow past a flat plate can be written as*:

$$w_x \frac{\partial t}{\partial x} = a \frac{\partial^2 t}{\partial y^2}. \qquad (15)$$

The boundary conditions are

$$\text{when } y = 0 \quad t(x, 0) = t_s, \text{ when } x = 0 \quad t(0, y) = t_c, \qquad (16)$$
$$\text{when } y \to \infty \quad t(x, \infty) = t_c.$$

The flow velocity w_x is a function of the coordinates which is determined from the solution of the equation of motion. Krisher's method assumes that w_x is a constant magnitude and equal to the average flow velocity in the boundary layer ($w_x = \overline{w}_x$). In actual processes a constant velocity occurs only with liquid flow without friction, i.e., at a very small coefficient of internal friction. In the case of a viscous liquid such an assumption ($w_x = \overline{w}_x = \text{const}$) is a method for solving the problem of heat transfer in the boundary layer.

In the case of flow about an infinitely long plate ($l \to \infty$) the solution of differential equation (15) with boundary conditions (16) has the form

---

* Heat transfer by vapor diffusion can be disregarded as a small magnitude.

$$\frac{t(x,y)-t_s}{t_c-t_s}=\mathrm{erf}\left(\frac{y\sqrt{\bar{w}_x}}{2\sqrt{ax}}\right). \tag{17}$$

The local Nusselt number is

$$\mathrm{Nu}_x=\frac{\alpha_x x}{\lambda}=\frac{x}{(t_c-t_s)}\frac{\partial t(x,0)}{\partial y}. \tag{18}$$

Differentiating solution (17) with respect to y and assuming y = 0, we obtain

$$\mathrm{Nu}_x=\frac{1}{\sqrt{\pi}}\left(\frac{\bar{w}_x x}{a}\right)^{1/2}=\frac{1}{\sqrt{\pi}}\sqrt{\mathrm{Pe}_x}. \tag{19}$$

The average Nusselt number over the surface is

$$\mathrm{Nu}=\frac{1}{l}\int_0^l \mathrm{Nu}_x\,dx=\frac{2}{\sqrt{\pi}}\sqrt{\mathrm{Pe}}. \tag{20}$$

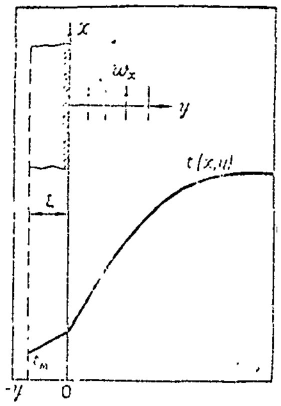

Fig. 2. Diagram of the calculation of heat transfer in evaporative transpiration cooling.

In order to compare the obtained results with known formulas for the Nusselt number in a laminar flow past a plate, we must determine the magnitude $\bar{w}_x$. If the velocity profile $w_x(y)$ is assumed a cubic parabola, the mean integral velocity [15] is

$$\bar{w}_x = w_c \frac{1}{\delta} \int_0^\delta w_x(y)\,dy = \frac{5}{8} w_c, \tag{21}$$

where δ is the boundary-layer thickness.

Then for moist air ($Pr = 0.7$)

$$\mathrm{Nu} = 0{,}625 \frac{2}{\sqrt{\pi}} \sqrt{\mathrm{Pe}} = 0{,}74 \sqrt{\mathrm{Re}}. \tag{22}$$

This result differs only by 12% from the known empirical formula $Nu = 0.66 \sqrt{Re}$. The assumption of the constancy of velocity w_x in the solution of the differential equation of the boundary layer is thus fully admissible.

Our problem concerning evaporative transpiration cooling can be set up as in Fig. 2.

The differential equation of heat transfer stays the same (15).

The boundary conditions will be:

$$t(0, y) = t_c, \quad t(x, \infty) = t_c, \quad t(x, \xi) = t_{\text{м}}; \tag{23}$$

$$-\lambda \frac{\partial t(0, x)}{\partial y} = -\lambda_{\text{т}} \frac{\partial t(0, x)}{\partial y} = \frac{\lambda_{\text{т}}}{\xi} [t(0, x) - t_{\text{м}}]. \tag{24}$$

Here it is assumed that the temperature at the evaporation surface equals the temperature of a wet-bulb thermometer. Owing to the small boundary-layer thickness of a body the temperature distribution in it follows the linear law. In this case (24) we can write:

$$-\frac{\partial t(0, x)}{\partial y} + H [t(0, x) - t_{\text{м}}] = 0, \tag{25}$$

where $H = \frac{\lambda_t}{\lambda_\xi}$ is some magnitude analogous to the relative heat transfer coefficient.

The solution of differential equation (25) with boundary conditions (23) and (25) has the form

$$\frac{t(x,y)-t_{\text{м}}}{t_c-t_{\text{м}}}=\operatorname{erf}\left(\frac{y\sqrt{\overline{w}_x}}{2\sqrt{ax}}\right)+$$
$$+\exp\left(Hy+H^2\frac{ax}{\overline{w}_x}\right)\operatorname{erfc}\left(\frac{y\sqrt{\overline{w}_x}}{2\sqrt{ax}}+H\sqrt{\frac{ax}{\overline{w}_x}}\right). \tag{26}$$

From (26) we derive the solution of (17) as a particular case. If evaporation occurs at the body surface ($\xi = 0$), then the second term of the right-hand part of (26) equals zero, because when

$$\xi=0 \quad H\to\infty.$$

The temperature at the body surface ($y = 0$) will not be constant, but varies along the x-axis:

$$\frac{t(x,0)-t_{\text{м}}}{t_c-t_{\text{м}}}=\exp\left(H^2\frac{ax}{\overline{w}_x}\right)\operatorname{erfc}\left(H\sqrt{\frac{ax}{\overline{w}_x}}\right). \tag{27}$$

At the plate edge ($x = 0$) the temperature of the body surface t_s [$t_s = t(x, 0)$] equals the air temperature ($t_s = t_c$), whereas at an appreciable distance ($x \to \infty$) it equals the temperature of the wet-bulb thermometer ($t_s = t_m$). Consequently, the temperature difference Δt ($\Delta t = t_c - t_s$) varies from zero at the plate edge to a constant magnitude ($t_c - t_m$). This is a very important fact determining the specific characteristics of heat and mass transfer upon deepening of the evaporation surface into the body. If evaporation occurs at the body surface, then under adiabatic conditions the body surface temperature is constant and equals the temperature of the wet-bulb thermometer.

It is known from the general theory of heat transfer that if the temperature difference Δt increases in the flow direction, the heat-transfer coefficient becomes larger than at a constant temperature [14]. Consequently, upon deepening the evaporation surface the heat-transfer coefficient is larger than when evaporation occurs at the surface. If in the first approximation we assume that the heat-transfer coefficient in evaporation at a body surface equals the heat-transfer coefficient of a dry body, then on drying with a deepening of the evaporation surface the heat-transfer coefficient will be larger as compared with the heat-transfer coefficient of a dry body. This increase of the heat-transfer coefficient must be reflected in the calculation formulas Nu = f(Re) by introducing an additional generalized argument. Since Δt is the cause of the change in the heat-transfer coefficient, then, naturally, the parametric criterion (generalized argument) will be the Gukhman number or $\left(\frac{T_c}{T_m}\right)$. We will dwell on this in detail.

The local Nusselt number is

$$\mathrm{Nu}_x = \frac{x}{t_c - t(0,\ x)} \frac{\partial t(x,\ 0)}{\partial y} = \sqrt{\mathrm{Pe}_x}\, K \exp K^2 \operatorname{erfc} K \times \\ \times [1 - \exp K^2 \operatorname{erfc} K]^{-1}, \tag{28}$$

where the dimensionless variable

$$K = \frac{Hx}{\sqrt{\mathrm{Pe}_x}} = \frac{\lambda_T}{\lambda\xi} \sqrt{\frac{ax}{w_x}} \tag{29}$$

characterizes the effect of deepening the evaporation surface on heat and mass transfer of capillary-porous bodies.

We will designate

$$f(K) = \sqrt{\pi}\, K \exp K^2 \operatorname{erfc} K. \tag{30}$$

Then

$$N = \frac{\sqrt{\pi}\, \mathrm{Nu}_x}{\sqrt{\mathrm{Pe}_x}} = \frac{f(K)}{1 - \frac{1}{K\sqrt{\pi}} f(K)}. \tag{31}$$

We derive (19) from the solution of (31). If evaporation occurs at the surface ($K = \infty$), then $N = 1$ since $f(K) = 1$, i.e.,

$$\mathrm{Nu}_x = \frac{1}{\sqrt{\pi}} \sqrt{\mathrm{Pe}_x},$$

which fully agrees with solution (19).

Thus the dimensionless magnitude N characterizes the relative increase of the local Nusselt number during evaporation of moisture from capillary-porous bodies as compared with evaporation of moisture at the body surface.

Figure 3 shows a graph of $N = f(K)$ from where we see that the magnitude N decreases with an increase of K, gradually approaching unity.

In the range of K values from 0.1 to 5 ($0.1 < K < 5.0$) the dependence $N = f(K)$ can be approximated by the relationship

$$N = 1{,}31\, K^{-0{,}12}. \qquad (32)$$

It is known from the theory of drying moist materials that the distance of the evaporation surface from the body surface in the constant drying rate period in the first approximation is directly proportional to the psychrometric difference $\Delta t (\Delta t = t_c - t_m)$. Then the dimensionless variable K will be inversely proportional to $(t_c - t_m)$ and consequently inversely proportional to the Gukhman number ($K \sim \sim \mathrm{Gu}^{-1}$). Hence it follows that $N \sim \mathrm{Gu}^{0,1}$, which takes place in experiments on heat and mass transfer in the drying process [2].

Only at the constant drying rate period is the temperature of the evaporation surface $t(x, -\xi)$ constant, starting with the critical moisture content its temperature increases with time of drying, gradually approaching the air temperature which it achieves at equilibrium moisture content. Hence it follows that Nu_x in the falling drying rate period will decrease with time, gradually approaching the Nusselt number for a dry body.

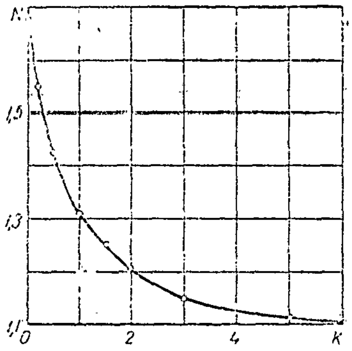

Fig. 3. Dependence of coefficient N on parameter K.

It is of interest to determine approximately the magnitude ξ. For a porous ceramic (λ = 0.2 kcal $\cdot$ m^{-1} $\cdot$ hr^{-1} $\cdot$ deg^{-1}) when Re = $= 6 \cdot 10^4$ and for x = 20 mm and parameter K = 1.5, the magnitude ξ = 0.5 mm. In this case, according to the graph in Fig. 3, coefficient N = 1.25, i.e., the heat-transfer coefficient in drying is approximately 25% greater than that for a dry body, if we consider that heat transfer in evaporation at the surface of a body is identical to heat transfer of a dry body. For the parameter K = 0.25, the magnitude ξ = 3 mm, and the coefficient N = 1.55, i.e., the heat transfer coefficient is about 25% greater during drying as compared with heat transfer of a dry body.

It is completely natural that at small values of ξ it is practically impossible to measure accurately the temperature of the body surface. Thermocouples embedded on the "surface" of a body for all practical purposes show the wet-bulb temperature. Therefore the heat-transfer coefficient can be considered as the ratio of heat flow to the psychrometric differences $(t_c - t_m)$:

$$\alpha_{xм} = \frac{q}{(t_c - t_м)} . \qquad (33)$$

Then the local Nusselt number is

$$\mathrm{Nu}_{xм} = \frac{\alpha_{xм} x}{\lambda} = \frac{x}{(t_c - t_м)} \; \frac{\partial t(x, 0)}{\partial y} . \qquad (34)$$

After simple transformations we obtain

$$N_м = \frac{\sqrt{\pi}\,\mathrm{Nu}_{xм}}{\sqrt{\mathrm{Pe}_x}} = f(K), \qquad (35)$$

where N_m is the coefficient showing the relative change of the number Nu_{xm} and consequently, of the coefficient α_{xm} owing to deepening of the evaporation surface. The graph $N_m = f(K)$, given in Fig. 4, shows that the coefficient N increases with increase of K.

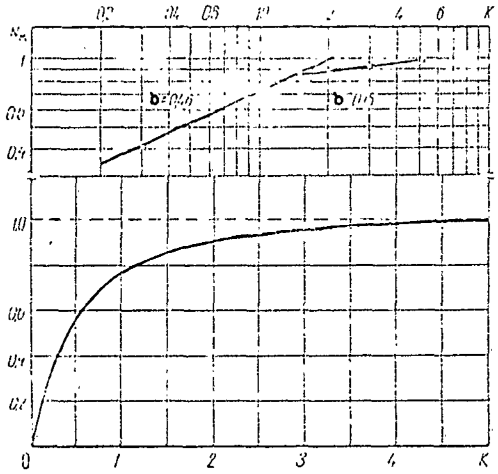

Fig. 4. Dependence of the coefficient N_m on parameter K in ordinary and logarithmic scales.

Since the dimensionless variable K is inversely proportional to the psychrometric difference $(t_c - t_m)$, the Nusselt number will decrease with its increase or that of the Gukhman number.

In a small variation range of K the dependence $N_m = f(K)$ can be presented as

$$N_m = BK^b, \tag{36}$$

where B and b are constants ($0 \leq q \leq 1$). For example, in the range $0.3 < < K < 1.5$ $B = 0.73$, $b = 0.46$, and when $1.5 < K < 5$ $B = 0.80$, $b = 0.15$ (Fig. 4).

Consequently, in the range $0.3 < K < 1.5$ the coefficient N_m will be directly proportional to $Gu^{-0.46}$. A similar relation occurred in Shull'man's work [9] in which he experimentally obtained the relationship

$$\frac{Nu_x}{\sqrt{Re_x}} = f(\overline{X})\, Gu^{-0.4}, \tag{37}$$

where $f(\overline{X})$ is a function of a dimensionless coordinate.

By its structure $N_m = 0.73K^{0.46}$ is close to empirical formula (10), since the variable K is inversely proportional to Δt. We can note that the authors of article [8] carried out their experiments using the Gukhman number. In this treatment they obtained the following dependence:

$$Nu = 0.00695\, Re^{0.8}\, Gu^{0.5}, \tag{38}$$

which almost agrees with the calculation formula with respect to the exponent q in spite of the different conditions of flow past the plate. We should point out that a comparison of these formulas is tentative since the magnitude ξ depends on the capillary-porous structure of the body, its physicochemical properties, and in the general case is a function of the parametric criterion (T_c/T_m). However formulas (31), (35) convincingly show that in these studies [8-9], evaporation of water occurred at a certain depth from the body surface and the decrease in the heat-transfer coefficient with an increase of evaporation rate, established from experimental data, is explained not by the crossflow of mass to the boundary layer but by the method of calculating the value of the heat-transfer coefficient. The heat-transfer coefficient in drying of capillary-porous bodies or in evaporative transpiration cooling is greater than the heat-transfer coefficient of a dry body at ordinary temperature differences ($\Delta t = 250-300^\circ C$).

Actually in these studies [8-9] not the heat-transfer coefficient α_x was determined but the over-all heat transfer coefficient

$$k = \alpha_{xM} = \left(\frac{1}{\alpha} + \frac{\xi}{\lambda}\right)^{-1}. \tag{39}$$

It is completely natural that the coefficient <u>k</u> decreases with an increase of the psychrometric difference ($t_c - t_m$) since ξ in the first approximation is proportional to ($t_c - t_m$).

The basic conclusion of our investigation is that heat and mass transfer of capillary-porous bodies with the ambient medium is a single interrelated process of heat and mass transfer in the boundary layer of the body and in the boundary layer of the medium.

CONCLUSIONS

In this article we have analyzed Eckert's solution [10] on the effect of mass transfer on heat transfer with gas injection into a laminar boundary layer. It was shown that with liquid evaporation from a porous plate, a decrease in the heat transfer coefficient of more than 10% can take place only at a temperature head $\Delta t \geq 260^\circ C$.

Under the usual conditions of drying and evaporative transpiration cooling this effect can be neglected. An analysis of the experimental data on liquid evaporation from a free surface shows that an additional argument, the Gukhman number, should be introduced into the impirical relations Nu = f(Re, Pr) (see formulas (11), (12)). The Gukhman number characterizes the thermodynamic intensity of volume evaporation in a boundary layer.

The problem of heat transfer in evaporative transpiration cooling with the presence of deepening of the evaporation surface is solved by Krisher's method [14] (see Eq. (26)). It was shown that in this case the heat-transfer coefficients are larger than those with evaporation at a body surface, all other things being equal. A decrease in the heat-transfer coefficient with a decrease of the temperature difference is accounted for in the calculation formulas by the Gukhman number.

It was shown that in other studies [8, 9] not the heat-transfer coefficient α was calculated but the over-all heat-transfer coefficient k (see formula (39)), which decreases with an increase of the temperature difference. Heat and mass transfer of capillary-porous bodies with the ambient medium is a single interconnected process of heat and mass transfer in the layers of the body and gas.

DESIGNATIONS

a — coefficient of thermal diffusivity (m^2/hr); c_p — specific isobaric heat capacity of moist air (kcal/kg · deg); j_1 — evaporation rate for flow density of a substance ($kg/m^2 \cdot hr$); l — a characteristic dimension (m); D — mass (kg); p — over-all air pressure (mm Hg); q — density of heat flow ($kcal/m^2 \cdot hr$); t — temperature (°C); T — absolute temperature (°K); r — specific heat of evaporation (kcal/kg); w — velocity of air motion (m/sec); α — heat-transfer coefficient ($kcal/m^2 \cdot hr$); η — coefficient of dynamic viscosity (kg/m · sec); λ — coefficient of thermal conductivity (kcal/m · hr · deg); ν — coefficient of kirematic viscosity (m^2/sec); ρ — density (kg/m^3);

ρ_{10} — relative vapor concentrations; τ — time; ξ — distance of the evaporation surface from the body surface (m).

Similarity numbers: Re_x — local Reynolds number value; Pe_x — local Peclet number value; $\overline{Pe}_x$ — local Peclet number value referred to the average integral velocity $\overline{w}_x$ in the boundary layer ($\overline{Pe}_x = \overline{w}_x x/a$); Nu_x — local Nusselt number; Sc — Schmidt number; Pr — Prandtl number; Gu — Gukhman number.

Indexes: c — ambient medium (moist air); m — state of adiabatic saturation; s — surface; t — solid body; x — local value depending on the x-coordinate; 1 — vapor; 2 — dry air; 0 — heat transfer without mass transfer.

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