HEAT TRANSFER WITH THE CONDENSATION OF MOVING FREON-21 IN A HORIZONTAL TUBE

I. I. Gogonin and A. R. Dorokhov

The question of the effect of the velocity of a condensing vapor on heat transfer was first investigated theoretically by Nusselt for the laminar flow of a film condensate [1]; the solution can be represented in the form [2]

$$\frac{\alpha}{\alpha_0} = f(\pi) \qquad \left(\pi = \frac{c_f w^2 \gamma'' \alpha_0}{\lambda \gamma' g}\right) \tag{1}$$

Here π is a dimensionless complex; α and α_0 are the heat-transfer coefficients, with the condensation of moving and stationary vapor, respectively, W/(m · deg); λ is the thermal conductivity of the liquid, W/(m · deg); γ'' and γ' are the specific weights of the vapor and liquid, N/m^3; c_f is the local friction coefficient; w is the velocity of the vapor outside of the boundary layer.

For the turbulent flow of a film of condensate, the first sufficiently general investigation was described in [3]. A further development of the theory is contained in [4-6].

However, the results of experimental investigations [4, 7, 8] do not confirm the existence of the single-valued dependence (1), if the friction coefficient, c_f, is taken in accordance with the conditions of flow around a "dry" surface. There is not only a quantitative, but also a qualitative divergence between theory and experiment.

The authors of [9] explain this divergence by the fact that all the preceding investigations did not take account of the considerable effect of suction from the boundary layer of vapor on the friction coefficient of the latter at the surface of the vapor. Actually, in the presence of the strong suction which almost always exists in condensation processes, the friction coefficient cannot be determined using formulas for flow around an impermeable surface. For a laminar boundary layer, the problem of suction was solved in the monograph of Schlichting [10]. In this case, with strong suction, the friction (the tangential stress, N/m^2) on a permeable surface is equal to

$$\tau = jw \quad (j = \rho'' v) \tag{2}$$

Here j is the mass flow rate of the transverse flow of mass through a permeable wall, kg/(sec · m^2); ρ'' is the density of the vapor, kg/m^3; v is the velocity of the vapor at the vapor–liquid interface, m/sec.

For a turbulent boundary layer, the general solution for a permeable surface is given in [11, 12] in the theory of a boundary with a vanishing viscosity.

The relative friction coefficient with flow around a permeable plate with j_w = const and a Reynolds number R_x = idem

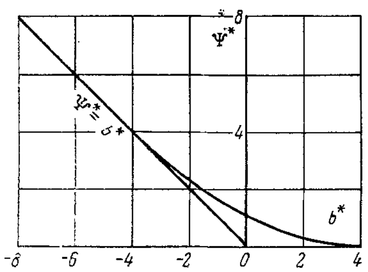

Fig. 1

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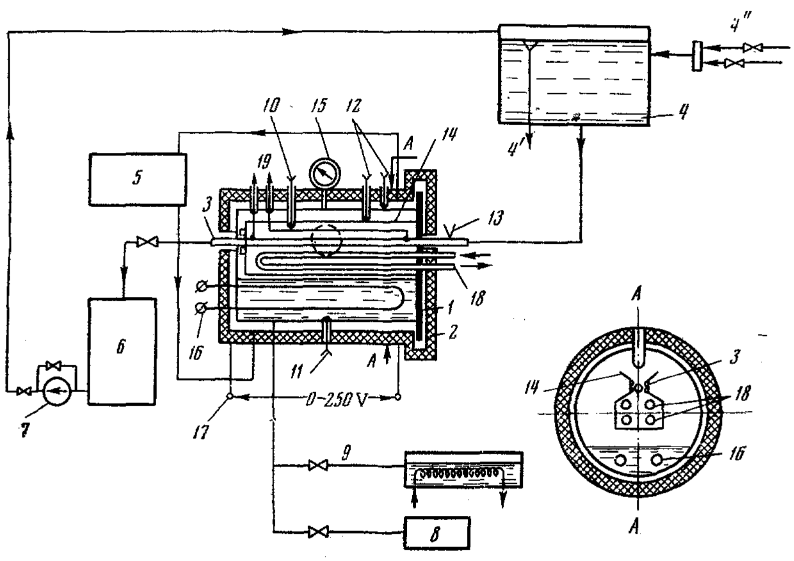

Fig. 2

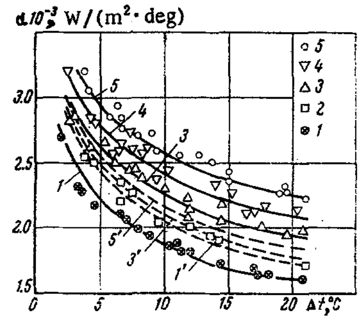

Fig. 3

$$\Psi^* = \left(\frac{c_f}{c_{f_0}}\right)_{R_x} = \frac{(1-0,25b)^2}{(1+0.25b)^{0.2}} \tag{3}$$

$$\left(b = \frac{2j_w}{c_{f_0}};\quad j_w = \frac{\rho'' v}{\rho'' w},\quad -4 < b < 4\right)$$

Here c_{f_0} is the friction coefficient at an impermeable wall.

If we construct the dependence of Ψ^* on the permeability factor, referred to the local friction coefficient (Fig. 1)

$$b^* = \frac{2j_w}{c_{fx}} = \frac{b}{(1+0.25b)^{0.2}} \tag{4}$$

where $-\infty < b^* < 3.45$, then at $b^* \to -\infty$ we have

$$\Psi^* = b^* \tag{5}$$

Thus, at the limit, dependence (2) holds for any given flow conditions in the boundary layer. From this, we have

$$\frac{\alpha}{\alpha_0} \to f(\pi_1 \pi_2),\quad \pi_1 \equiv j_w,\quad \pi_2 = \frac{w^2\gamma''\alpha_0}{\lambda\gamma' g} \tag{6}$$

In the more general case, the dependence $c_f(b)$ must be introduced in accordance with (3). For the process of film condensation

$$\pi_1 = \frac{\rho'' v}{\rho'' w} \equiv \frac{q}{r\gamma'' w}$$

Here q is the density of the heat flux, W/m^2; r is the latent heat of vaporization, J/kg.

In the literature, there are very few experimental data on the condensation of moving water vapor in a horizontal tube; these data exhibit a considerable amount of divergence in a comparable region of vapor flow rates. In view of this, the experimental investigation of the condensation of moving Freon vapor is of definite importance, both to verify the theoretical relationships, and for practical applications.

The use of Freon, whose condensation takes place with an overpressure, makes it comparatively easy to eliminate noncondensing gases from the vapor volume; as is well known [7, 8], these gases constitute the main source of error in measurements of heat transfer with condensation.

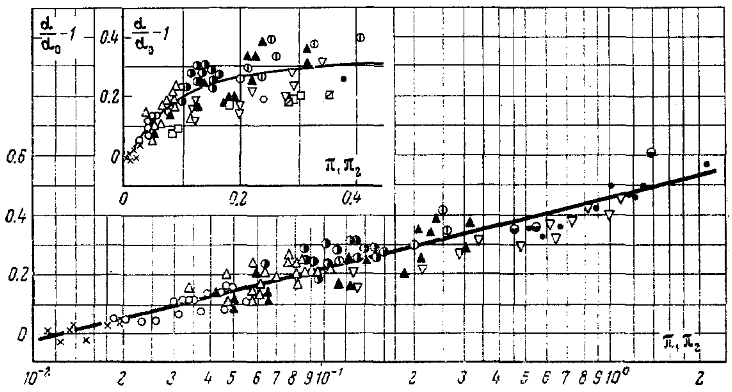

Fig. 4

The investigations were carried out in a unit, a schematic diagram of which is shown in Fig. 2. The unit was designed after the type used in [8]. Its main subassemblies were: the working volume 1, consisting of a stainless-steel cyclinder with a dimater of 350 mm and a length of 600 mm; a closed water recirculation loop, connected to the constant-head tank 4; the experimental section 3; a water pump 7, and a system for charging the Freon 9, and for evacuation 8. The working volume was carefully thermostatted. The thermostatting system consisted in maintaining a zero temperature difference between the working volume and the water jacket 2, in which a constant temperature of the water was maintained using a U-10 ultrathermostat 5, and of a compensating heater 17, wound on the outer shell; all the subunits were carefully insulated thermally, using asbestos.

The experimental section 3 was a smooth nickel tube with a diameter of D = 17 mm and a length of 520 mm. It was installed between the inner walls of the shell 14, thanks to which there was set up a directed flow of vapor from the top downward, as the result of condensation of the vapor in the auxiliary condenser 18, consisting of six finned tubes. In the salts, additional half-tubes were used to simulate flow of the vapor around the experimental section along a second series of staggered bundles with a spacing S/D = 1.6. The width of the narrowest part between the additional tubes was equal to 10 mm. The distance between the walls of the shell at the vapor inlet was 27 mm. The Freon was vaporized using heater 16, installed in the lower part of the volume; the heater had a power of 10 kW. The velocity of the vapor was calculated from the power developed by the heater, and was referred to a narrow cross section of the bundle. The heat flux was determined from the mass flow rate and the heating up of the water in the experimental tube. The mass flow rate of the water was measured by the weight method, and the heating up using the differential thermocouple 13. The temperature of the saturated vapor was determined with the two-junction thermocouple 10, and the wall temperature of the experimental section from the change in its resistance. For this purpose it was connected into the circuit of an R329 bridge. A correction for the change in the wall temperature over the thickness was introduced using a known formula [13]. All the thermocouples were individually calibrated, and the experimental section was calibrated after its installation in the volume. The calibration was verified before each series of measurements. After the unit had been filled with Freon, the system was purged repeatedly to eliminate any traces of air; this was monitored from the corresponding dependence of the pressure on the temperature. The accuracy of the measurements was determined on the basis of the accuracy in determination of the heat flux and, with a temperature head between the vapor and the wall $\Delta t \geq 5°C$, was not less than 7%.

Figure 3 gives the experimental data on heat transfer, and the values of the heat-transfer coefficient $\alpha^* = \alpha \cdot 10^{-3}$, W/(m^2·deg), as a function of Δt °C, with the condensation of stationary Freon-21 vapor (points 1) and moving vapor (points 2, 3, 4, 5), at vapor flow velocities w = 0.11, 0.22, 0.37, and 0.56 m/sec, respectively. The dotted lines 1', 3', 5' represent calculation using a theoretical formula obtained in [9], at w = 0, 0.22, and 0.56 m/sec. The experiments were made at a pressure of 5.2 bar. In this case, the temperature head varied from 2 to 20°C. The experimental points for stationary vapor differ from those calculated using the Nusselt theory by not more than 10%. In experiments in the presence of velocity, the form of the dependences is in qualitative agreement with the experimental data of other authors on water vapor. While, for stationary vapor the heat-transfer coefficient is proportional to the temperature head to the −0.25 power, for moving vapor the exponent of Δt decreases with an increase in the velocity and, in our experiments, attains −0.18.

A correlation of the experimental data obtained, carried out in accordance with dependence (6), is presented in Fig. 4, on which points 1, 2, 3, 4 correspond to the velocities $w = 0.11$, 0.22, 0.37, and 0.56 m/sec for Freon vapor, and points 5, 6, 7, 8, and 9 to velocities from 1 up to 16 m/sec for water vapor; points 5 and 6 were obtained at $p = 0.48$ atm, and points 7, 8, and 9 at $p = 0.88$. The data on the condensation of water vapor were taken from [8]. It is evident that dependence (6) correlates sufficiently well the experimental data obtained with the condensation of water and Freon vapors.

Here, it must be borne in mind that a majority of the experimental data of [8] was obtained at small temperature heads ($\Delta t = 1.2$-4°C) and, therefore, cannot pretend to a high degree of accuracy in determination of the heat-transfer coefficient. In addition, there is a stratification of these data with respect to pressure which is evidently connected with the rising concentration of air at a lowered pressure.

It is evident from Fig. 4 that the strongest effect of suction on heat transfer is observed at values of the determining parameter lower than 0.15.

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