THEORETICAL STUDY OF HEAT TRANSFER ON HORIZONTAL TUBES IN THE CONDENSATION OF A DESCENDING VAPOR FLOW

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A theoretical solution is proposed for a problem of heat transfer on horizontal tubes in the condensation of a descending vapor flow. Results are presented from calculation of theoretical heat transfer and are compared with experimental results.

An analytical solution was given in [1] for a problem of heat transfer in the laminar film condensation of a descending vapor flow on single horizontal tube. The following assumptions were made: a) the laminar film of condensate flowed under the force of gravity, pressure, and shear forces at the boundary between the condensate film and the laminar boundary layer of the moving vapor, which was regarded as a boundary layer with suction; the shear forces acted on an arc from the frontal point to the separation point of the vapor boundary layer; beyond this point, the condensation conditions were assumed to be the same as in the condensation of stationary vapor; b) heat flux on the arc Ω was considered to be constant and equal to q_m, while heat flux on the arc $\pi - \Omega$ was also constant and equal to $q_n < q_m$. Thus, on the arc Ω, the rate of suction (blowing) from the vapor boundary layer turns out to be uniform and equal to $(-v_0) = q_m/(\rho_V r)$; c) beyond the boundary layer, the vapor flow was assumed to be a potential flow, meaning that the tangential velocity of the vapor U at the outer boundary of the boundary layer was distributed according to the familiar sine law; d) about the tube periphery, the available temperature head θ and the total heat-transfer resistance of the cooling medium and tube wall R_{gw}, referred to the outer surface of the tube, were constant.

It was shown that the coefficient of heat transfer $\bar{\alpha}_m$ from the moving, condensing vapor on the arc Ω depends on the dimensionless complexes Re_{VC}, Π_m, I_d, Nu_{gw}, and Π_m^* (the effect of Π_m^*, characterizing the pressure forces, is relatively slight), while the ratio $\bar{\alpha}_m/\bar{\alpha}_n$ depends on Π_m, I_d, Nu_{gw}, and Π_m^*.

It is important to note that the complex Π_m in [1] entered into all of the equations for determining the thickness of the condensate film δ_c not independently, but only in the form of the product $\Pi_m I_d^{-1/2}$, while the complex I_d also entered into these equations independently. It can be shown (which was not done in [1]) that $(\Pi_m I_d^{-1/2})^{4/3}$ is the well-known complex $\Pi_{m2} = Fr_V H$ if the vapor—wall temperature head ϑ entering into H is the mean temperature head in the condensation of the stationary vapor under the conditions $q = q_m = \text{const}$, i.e., if

$$\vartheta = \bar{\vartheta}_n = q_m/\bar{\alpha}_n^*, \text{and} \alpha_n^* = a[g\rho_c^2\lambda_c^3 r/(\mu_c d_n q_m)]^{1/3},$$

Then

$$\Pi_{m2} = Fr_V \lambda_c q_m/(\mu_c r\bar{\alpha}_n^*) = \frac{1}{a} U_\infty^2 (q_m/r)^{1/3}/(g^2\rho_c\mu_c d_n)^{2/3} = \frac{1}{a}\Pi_m^{4/3} I_d^{-2/3}. \quad (1)$$

Thus,

$$\bar{\alpha}_m = f(Re_{vc}, \Pi_{m2}, I_d, Nu_{gw}); \; \bar{\alpha}_m/\bar{\alpha}_n = \psi(\Pi_{m2}, I_d, Nu_{gw}). \quad (2)$$

Since, strictly speaking $q \neq \text{const}$ and $\vartheta \neq \text{const}$ with the boundary conditions $\theta = \text{const}$ and $Nu_{gw} = \text{const}$, the solution obtained in [1] is approximate. However, as follows from [2], it agrees rather well with the empirical data.

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An exact solution of the problem posed in [1], i.e., at $q_m \neq$ const and $q_n \neq$ const, can be obtained only through a numerical solution on a computer.

The present work offers such a solution for determining heat transfer in the laminar film condensation of a descending vapor flow on horizontal tubes. The flow of the vapor beyond the boundary layer obeys a more complex law than the sine law assumed in [1].

As in [1], the equation for the local thickness of the film of condensate on the tube is written

$$\left(1-\frac{2dp/d\varphi}{\gamma_{c\varphi}d_n}\right)\delta_c^3+\frac{3\tau_0}{2\gamma_{c\varphi}}\delta_c^2=\frac{3\mu_c d_n}{2\gamma_{c\varphi}\rho_c r}\left[\int_0^{\varphi}\frac{\theta d\varphi}{R_{gw}+(\delta_c/\lambda_c)}\right]. \quad (3)$$

The derivative $dp/d\varphi$ in Eq. (3) may be written

$$dp/d\varphi=-\rho_v U(dU/d\varphi). \quad (4)$$

We will determine the shear stress at the phase boundary τ_0 from an expression similar to that in [1]:

$$\frac{\tau_0}{\tau_{0.ac}}=3{,}22\sqrt{\left(\frac{0{,}0682}{\sigma}+0.174\right)\left(2\sigma\frac{I_d}{U_\infty}\frac{dU}{d\varphi}+\frac{0{,}0682}{\sigma}+0{,}174\right)}, \quad (5)$$

obtained as a result of calculation of an incompressible laminar boundary layer with blowing control. The form parameter σ is determined from the equation

$$\sigma^2=\frac{0.22U_\infty}{U^6 I_d}\int_0^{\varphi}U^5(1-2\sigma)\,d\varphi,\ I_d=I(\varphi), \quad (6)$$

while

$$\tau_{0.ac}=Uq/r,\ q=q(\varphi). \quad (7)$$

The vapor velocity distribution about the perimeter of a tube located in a tube bundle and submerged in a potential flow

$$U=U(\varphi,\ S_1,\ S_2) \quad (8)$$

was determined by the method of electrodynamic analogy (see [3], e.g.) on a special unit.

The thus-obtained relation (8) is shown in Fig. 1 for certain corridor and staggered bundles.

System (3)-(8) was solved on a BÉSM-4M computer with a special program in which the values of U from (8) were introduced in tabular form.

As a result of the calculation, after specifying values of the parameters and vapor velocity, law (8), and values of θ and Nu_{gw}, we can obtain the distribution of the quantities $\tau_0/\tau_{0.ac}$, q, ϑ, δ_c, and $\alpha = \lambda_c/\delta_c$ about the tube perimeter, as well as mean values of the heat-transfer coefficient

$$\bar{\alpha}=\left(\sum_1^m q_i\Delta\varphi_i\right)/\left(\sum_1^m \vartheta_i\Delta\varphi_i\right), \quad (9)$$

where m is the number of sections $\Delta\varphi_i$ into which the tube perimeter is divided in the computer calculation, i being the number of the section.

Figure 2 shows the resulting theoretical local values of $\tau_0/\tau_{0.ac}$, q, and ϑ with different values for the parameters of the water vapor and its velocities at the inlet to a staggered bundle. The values of θ and R_{gw} are the same in all cases.

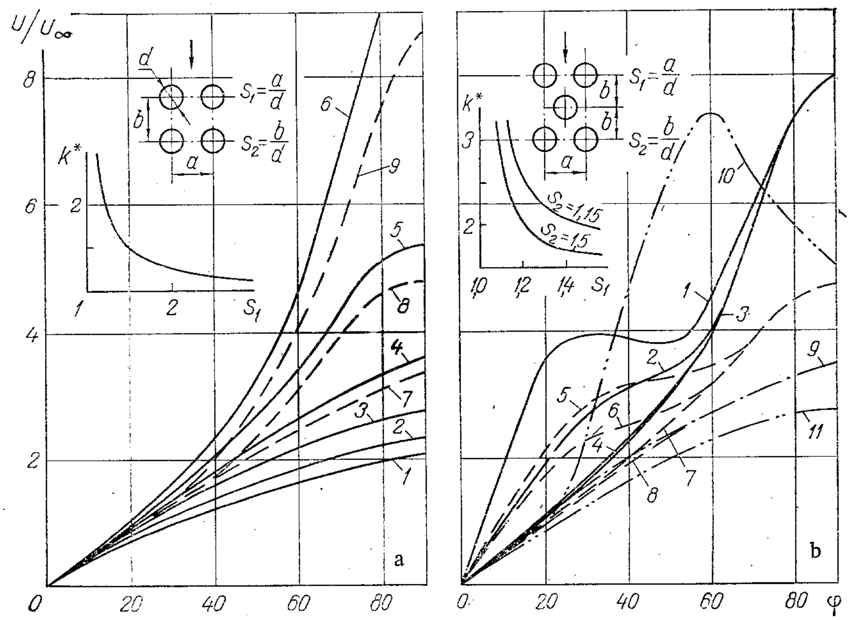

Fig. 1. Velocity distribution about the perimeter of a tube in a bundle submerged in a flow of different vapors: a) corridor bundles: 1, 2, 3, 4, 5, 6) $S_2 \geqslant 5$ with S = 5, 2.5, 2, 1.5, 1.25, and 1.1, respectively; 7, 8, 9) $S_1 = S_2 = 1.5, 1.25$, and 1.14; b) staggered bundles: 1, 2, 3, 4) $S_1 = 1.14$ with $S_2 = 1.14, 1.25, 1.4$, and $\geqslant 1.5$; 5, 6, 7) $S_1 = 1.25$ with $S_2 = 1.1, 1.25$, and $\geqslant 1.5$; 8, 9) $S_1 = 1.5$ with $S_2 = 1.3$ and ≥ 1.5, 10, 11) $S_1 = 1.85$ with $S_2 = 0.54$ and $\geqslant 1.5$, respectively. φ , deg.

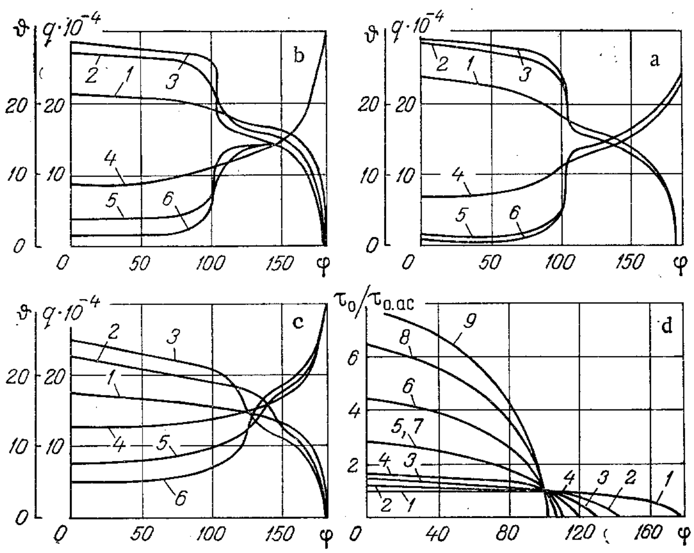

Fig. 2. Local values of q (W/m²·°C), ϑ (°C), and $\tau_0/\tau_{0.ac}$ about the perimeter of a tube with d_n = 19 mm in a staggered bundle with $S_1 = 1.5$ and $S_2 = 1.3$, $R_{gw} = 1 \cdot 10^{-4}$ m²·°C/W, and ϑ = 30°C: a) t_n = 240°C; 1, 2, 3) q; 4, 5, 6) ϑ at U_∞ = 0.6, 6, and 9 m/sec; b) same, but t_n = 140°C and U_∞ = 0.6, 6, and 18 m/sec; c) same as b, but t_n = 40°C; d) $\tau_0/\tau_{0.ac} = f(\varphi)$: 1, 2, 3) t_n = 40°C, U_∞ = 0.6, 6, and 18 m/sec; 4, 5, 6) same, but t_n = 140°C; 7, 8, 9) t_n = 240°C, U_∞ = 0.6, 6, and 9 m/sec.

Some of the calculations were performed for an actual velocity distribution, taken from data in [4, 5]. Here, the mean values $\bar{\alpha}$ for the actual and potential flows did not differ by more than 10% in the cases examined.

Analysis of the results of the calculations also showed that the velocity distribution law (8) could be replaced by the sine law

$$U = 2U_{\infty}^{*} \sin\varphi, \text{ where } U_{\infty}^{*} = k^{*}U_{\infty}, \tag{10}$$

with $k^* = f(S_1)$ for corridor bundles and $k^* = f(S_1, S_2)$ for staggered bundles (see Fig. 1). If such a substitution is made, the mean heat-transfer coefficient does not differ from the exact value by more than 5% for a fairly broad range of values of the determining parameters.

The angle $\varphi = \Omega$, at which $\tau_0/\tau_{0.ac} = 0$, corresponds to the separation point of the vapor boundary layer. It follows from Fig. 2d that nonseparated flow ($\Omega = 180°$) is seen only at low parameter values and low vapor velocities ($t_n = 40°C$, $U_\infty = 0.6$ m/sec). At a pressure above atmospheric and with the other conditions remaining the same, the position of the theoretical point of separation changes within a relatively narrow range, 130-123°. It also follows from Fig. 2d that the shear stresses are close to the asymptotic values only for vapor with $t_n = 40°C$ and that they deviate considerably from the latter when the pressure increases. Finally, it is evident from Fig. 2a-c that q and ϑ are fairly close to q = const and ϑ = const on the arc Ω (especially with an increase in the parameters and vapor velocities), which validates the assumptions made in the analytical solution in [1].

To evaluate the proposed theoretical solution, we adopted the coordinates $\overline{Nu}_{ex} = f(Nu_{theo})$ to compare more than 1000 experimental values of heat transfer in the condensation of a descending vapor flow on horizontal tubes without condensate inflow, taken from well-known works, with theoretical values of heat transfer calculated by the method described above for the conditions which prevailed in each experiment: the parameters and velocities of the vapor, values of θ and R_{gw}, and the geometric parameters of the bundles. We determined $R_{gw} = (\bar{t}_w - \bar{t}_g)/q = (\theta - \bar{\vartheta})/q$ from the experiments in which direct measurements were made of the mean temperature of the tube wall $\bar{t}_w$ (all of the experiments in [6-8] and one series in [9]) and the total heat-transfer resistance of the cooling medium and tube wall, which are needed to calculate the theoretical heat transfer. In particular, from [9] we obtained the relation $R_{gw} = f(G_g, \bar{t}_g)$, which was used to determine R_{gw} in those experiments in which wall temperature was not measured.

The results of the comparison are shown in Fig. 3, which provides evidence of the satisfactory agreement between the theoretical and empirical heat-transfer values for both water vapor and vapors of Freon-21 and Freon-12. It should be pointed out that the best agreement between the theoretical and experimental values of heat transfer is seen for the cases of vapor condensation on tubes located in a rectangular channel: vapors of Freon-21 and Freon-12, tube with d_n = 2.5, 6, 16, and 17 mm [7, 8]; water vapor, tube with d_n = 19 mm [9].

Most of the experimental values of heat transfer for the tubes located in staggered bundles [6-9] were about 10-25% higher – and in corridor bundles with $S_2 = 1.25$ [9], about 10% lower – than the theoretical values. This is evidently connected with the features of actual flow in tube bundles. In particular, the poor heat transfer in the above corridor bundle compared to the staggered bundle can be explained by the presence, on each tube in the corridor bundle (apart from the tubes of the first row) in real flow, of two (rather than one) frontal points, shifted by an angle $\beta \leq 30$-40°, and the occurrence, on the arc of β, of reversed currents [4] which act to slow the film of condensate.

In tests in [7-9], not shown in Fig. 3, the experimental values of heat transfer coefficient exceeded the theoretical values by 40-50% at $Re_v > 60{,}000$ and blowing velocities $(-v_0)$ = 0.1-0.3 m/sec (Re_v was calculated for the vapor velocity in a narrow cross section of the bundle). It may be assumed that this is the result of agitation of the vapor boundary layer and condensate film in these tests, whereas the layer and film were assumed to be laminar in deriving the theoretical relations.

In accordance with [10], the transition from a laminar boundary with blowing control to a turbulent boundary layer can be expected when

$$\ln Re_{\delta_2} > 34.2 H_{32} - 47.81 \tag{11}$$

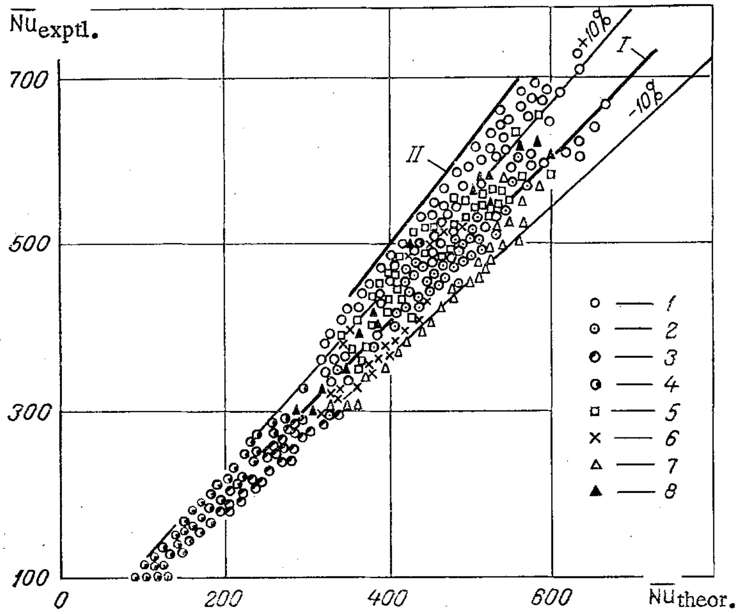

Fig. 3. Comparison of theoretical and experimental heat transfer: I) $\overline{Nu}_{ex} = \overline{Nu}_{theo}$; II) mean lines of tests of series IV with water vapor in [6] and of some of tests with Freon-21 and Freon-12 in [8] (Table 2, tests 26-89, Table 3, tests 13-27); 1) tubes with d_n = 17 and 16 mm in staggered and corridor bundles, Freon-21 and Freon-12 [7, 8]; 2, 4, 6) single tubes with d_n = 17 and 2.5 mm in rectangular channels, Freon-21 [7, 8] and d_n = 19 mm, water vapor [9]; 3) single tube with d_n = 6 mm in a rectangular channel, Freon-21 [7]; 5) tube with d_n = 19 mm in a staggered bundle, water vapor [6]; 7, 8) tube with d_n = 19 mm in corridor and staggered bundles, respectively [9].

and will definitely occur when

$$\ln \mathrm{Re}_{\delta_2} > 34.2H_{32} - 46.78, \qquad (12)$$

where $Re_{\delta_2} = U\delta_2/\nu_v$, and $\delta_2 = \sigma r \mu_v/q$; H_{32} is the form parameter, determined from [10] in relation to δ_2.

Calculations of local values of δ_2 and Re_δ show that, under the above conditions, the appearance of a turbulent vapor boundary layer may be expected when $Re_v > 4 \cdot 10^4$ and guaranteed when $Re_v > 12 \cdot 10^4$, which covers the experimental values of Re_v. Local values of $Re_c = \bar{w}_c \delta_c/\nu_c$ fall within the range 30-50 under the conditions of these experiments.

As follows from [11], agitation of the vapor boundary layer shifts its separation points down the flow and increases the shear stresses, meaning that there is also an increase in the average heat-transfer coefficient about the tube perimeter. The length of the turbulent boundary layer to its separation point and its effect on mean heat transfer will depend on Re_v, the geometry of the bundle, and heat-transfer conditions. This question requires further study.

NOTATION

α, coefficient in the expression for α_n^*; d, tube diameter, m; d, differential sign; g, acceleration due to gravity, m/sec^2; p, pressure, Pa; q, heat flux, W/m^2; r, heat of condensation, J/kg; R, heat-transfer resistance, m^2·deg C/W; S_1, S_2, relative transverse and longitudinal spacings of tubes in bundle; t, temperature, °C; U, tangential velocity of vapor at the outer boundary of the boundary layer, m/sec; U_∞, U_∞^*, velocity of vapor far from surface, m/sec; v, transverse velocity of vapor in the boundary layer (blowing velocity), m/sec; w, velocity of condensate film, m/sec; α, heat-transfer coefficient, W/(m²·°C); $\gamma_c \varphi = g\rho_c \sin\varphi$, projection of force of gravity onto the tube circumference tangent, kg·m/sec²; δ, thickness of film, layer, m; $\vartheta = t_n - t_w$, vapor–wall temperature head, °C; λ, thermal conductivity, W/m·°C); μ, ν, dynamic and kinematic viscosities, kg/(m·sec) and m²/sec; ρ, density, kg/m³; τ, shear stress, kg/(m·sec²); φ, running angular coordinate, rad; Ω, angular coordinate of separation point of laminar vapor boundary layer, rad; σ, form parameter of boundary layer with blowing; $\theta = t_n - t_g$, available temperature head, °C. Indices: v, vapor; c, condensate; n, stationary (for q, α, ϑ, δ_c, and Nu), external (for d), or saturated (for t); m, moving; g, cooling medium (water); w, wall; 0, phase boundary; ac, asymptotic complexes; $Fr_v = U_\infty^{*2}/(gd_n)$; $H = \lambda_c\vartheta/(\mu_c r)$; $I_d = (U_\infty^* \rho_v \mu_v / d_n)(r/q)^2$; $Nu = \alpha d_n/\lambda_c = d_n/\delta_c$; $Nu_{gw} = d_n(R_{gw}\lambda_c)$; $Re_v = U_v d_n/\nu_v$; $Re_{vc} = U_\infty^* d_n/\lambda_c$; $\Pi_m = Fr_v(\rho_v\mu_v/\rho_c\mu_c)^{1/2}$; $\Pi_m^* = Fr_v\rho_v/\rho_c$; $\Pi_{m2} = Fr_v H$.

LITERATURE CITED

1. V. M. Marushkin and Ya. L. Polynovskii, "Theory of heat transfer in the condensation of moving vapor on a horizontal cylinder," Teploenergetika, No. 4, 59-64 (1979).
2. V. M. Marushkin and Ya. L. Polynovskii, "Agreement of theoretical and experimental data on heat transfer in the condensation of moving vapor on a horizontal cylinder," Teploenergetika, No. 1, 55-58 (1980).
3. N. N. Suntsov, Analogy Methods in Aerohydrodynamics [in Russian], Izd. Fiz.-Mat. Lit., Moscow (1958).
4. A. A. Zhukauskas, V. I. Makaryavichus, and A. A. Shlanchyauskas, Heat Transfer on Tube Bundles in a Transverse Fluid Flow [in Russian], Mintis, Vilnius (1968).
5. Ž. G. Kostić and S. N. Oka, "Fluid flow and heat transfer with two cylinders in cross flow," Int. J. Heat Mass Transfer, 15, 279-299 (1972).
6. L. D. Berman and Yu. A. Tumanov, "Study of heat transfer in the condensation of moving vapor on a horizontal tube," Teploenergetika, No. 10, 77-83 (1962).
7. I. I. Gogonin and A. A. Dorokhov, "Experimental study of heat transfer in the condensation of moving Freon-21 vapor on a horizontal cylinder," Zh. Prikl. Mekh. Tekh. Fiz., No. 2, 133-139 (1976).
8. I. I. Gogonin, A. A. Dorokhov, and V. I. Sosunov, Heat Transfer in the Film Condensation of Moving Vapor [in Russian], ITF Sib. Otd. Akad. Nauk (Institute of Thermophysics, Siberian Branch, Academy of Sciences of the USSR), Novosibirsk (1980) (Preprint No. 66-80).
9. D. W. Nobbs, "The effect of downward vapor velocity and inundation on the condensation rates on horizontal tubes and tube banks," Diss. Univ. of Bristol (1975).
10. R. Eppler, "Praktische Berechnung laminarer und turbulenter Absauge-Grenzschichten," Ing. Arch., 32, 221-245 (1963).
11. G. Schlichting, Boundary Layer Theory, McGraw-Hill (1966).