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ON CALCULATION OF HEAT EXCHANGE FOR CONDENSATION IN HORIZONTAL TUBES

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UDC 536.246:536.422.4

A method is proposed for determining the intensity of condensation from the value of the equivalent force applied to the condensate film. The asymmetry of film flow along the perimeter of a horizontal tube and the dependence of the characteristic size of the calculation expressions on regime parameters of the condensing flow are demonstrated.

Heat exchange devices using condensation within horizontal tubes are found in many branches of industry. Despite this fact, the heat liberation conditions for such devices have not been studied sufficiently. An analysis of available data [1] shows that for one and the same range of condensing flow regime parameters the calculation expressions presented produce qualitative and quantitative divergences in the principles of heat exchange. The most significant divergences occur in calculated heat liberation coefficients in the range of moderate and low motion velocities and with complete condensation of the vapor. The absence of reliable engineering and theoretical computation equations is retarding the wide use of efficient horizontal heat exchange apparatus.

The causes for the existing divergence in calculation expressions for a comparable thermal flux density range [2-9] were partially analyzed in [1]. The author explained their existence by change in flow regimes of the phases along tube length, neglect of the size of the condensate stream, change in the film flow regime, which can be turbulent over a portion of the tube length. However, it should be considered that within the horizontal tube the force of gravity is perpendicular to the direction of vapor flow, and not aligned thereto, as in vertical tubes and the outer surface of horizontal tubes. This fact changes the film hydrodynamics and principles of heat exchange significantly.

The studies of local heat exchange performed in [10, 11] permitted establishment of some principles of film hydrodynamics within a horizontal tube. Analysis of data on local heat exchange shows that asymmetry of the film condensate thickness along the perimeter of the horizontal tube beginning with its initial segment is intrinsic to the flow of a two-phase medium (Fig. 1a). This effect intensifies with increase in condensation and thermal flux density (thick films) as indicated by the data shown in Fig. 1b. Moreover, significant removal of liquid into the vapor flow, possibly reaching 80%, is inherent to the process of moving vapor condensation [10].

Comparison of experimental values of the heat liberation coefficient (Fig. 2), obtained in a segment $\ell = 0.04$ m long with values calculated from the Nusselt dependence for conditions where the tube position in space has no effect on heat exchange [12], and for the case of dominant friction forces and turbulent film flow [13] shows that the numerical values of the mean heat liberation coefficients on the perimeter of the horizontal tube $\bar{\alpha}_\varphi$ are higher than calculated in all cases. Such a result is the consequence of flow of the condensate film on the lower tube directrix into the condensate stream zone, which increases the intensity of the process on the remaining portion of the perimeter. In turn this feature of the film hydrodynamics causes the condensate in its motion on the lower directrix to traverse a

Kiev Polytechnic Institute. Translated from Inzhenerno-fizicheskii Zhurnal, Vol. 62, No. 1, pp. 16-23, January, 1992. Original article submitted April 1, 1991.

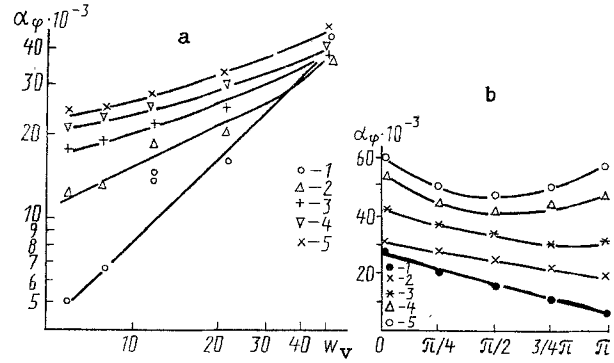

Fig. 1. Effect of vapor motion velocity on local heat liberation coefficients (α_φ over φ) over horizontal tube perimeter: $\bar{q}_\varphi$ = 170 kW/m^2, Re_f = 14, T_s = 373 K; 1) $\varphi = \pi$ (lower directrix of horizontal tube); 2) 3/4π; 3) 1/2π; 4) 1/4π; 5) 0 (upper directrix of tube); b) $\bar{q}_\varphi$ = 440 kW/m^2, Re_f = 28, T_s = 373 K; 1) w_v = 6 m/sec; 2) 15; 3) 28; 4) 54; 5) 65.

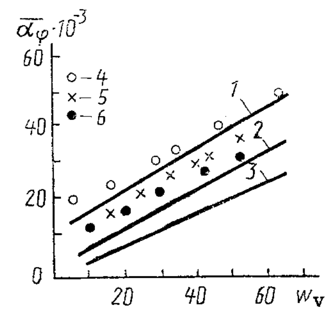

Fig. 2. Comparison of experimental mean $\bar{\alpha}_\varphi$ over φ with calculated dependences of other authors: 1-3) calculation lines, 1) calculation after [12]; 2, 3) [13]; 2) Re_f = 14, 3) Re_f = 360; 4-6) experimental data [10, 11]; 4) Re_f = 14, 5) 90, 6) 360.

different path to the tube output section. Consequently, differences in linear and regime parameters ($\bar{q}_L$; w_v, d_t) existing in the experiments determined the varying numerical values of the characteristic condensate motion path, and hence, heat exchange intensity along tube length.

At the present time no expression exists for calculating the condensate path for various combinations of regime and linear parameters in the process of condensation in horizontal tubes.

It should also be noted that true computation expressions will not permit determination of the change in thickness (force of gravity) of the film due to liquid removal into the vapor flow, which causes an increase in the length of its path through the horizontal tube. It is these features of the two-phase flow hydrodynamics which explain the significant divergences [1] in proposed calculation expressions, since, for example, nonconsideration of turbulence of the condensate film flow would lead to a divergence in heat liberation coefficients of only 21% [12]. In fact the calculated data differ not only by 100% and more, but also in the character of the dependence $\alpha = f(\bar{q}_L)$.

In addition, commencing from Nusselt's theoretical solutions, the intensity of heat exchange in condensation of nonmoving and moving vapor depends on the condensate path along the heat exchange surface and thus the numerical value of the force of gravity or interphase friction.

Considering the above, we will now examine the possibility of obtaining a computation expression for determining the heat liberation coefficients in condensation of a moving vapor with laminar flow of the condensate within a horizontal tube by analyzing the effect on film hydrodynamics of the equivalent forces applied to the liquid volume. We will base the computation method on assumptions, boundary conditions, and Nusselt number dependence for the non-

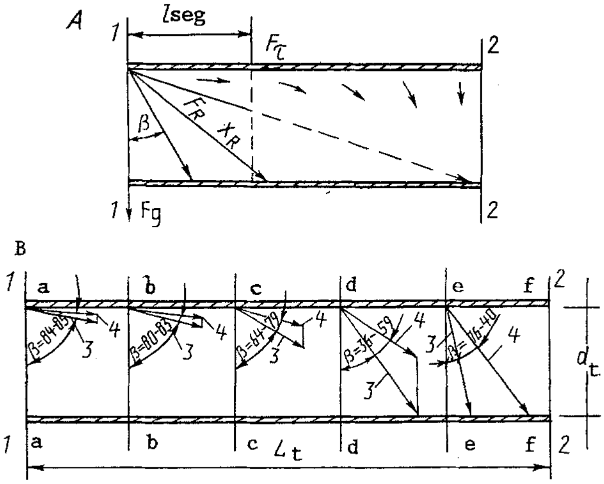

Fig. 3. Diagram of forces acting on condensate film, and direction of its flow along length of horizontal tube for complete condensation of moving vapor: 1–1, tube input section; 2–2, output section; a–a, tube section in which w_V = 60 m/sec; b–b, same, w_V = 40; c–c, 20; d–d, 10; e–e, 5; f–f, 0; 3, value of angle β for q_L = 400 kW/m²; 4, same for q_L = 50 kW/m².

moving vapor in which we replace the force of gravity by equivalent gravitational and tangent forces applied to the condensate film. In this case, the only unique feature of the differential equations describing heat exchange during condensation is that the equation of motion contains the equivalent force F_R, which is assumed constant at the point or along the tube segment under consideration:

$$\frac{\partial^2 t}{\partial y^2} = 0, \tag{1}$$

$$\frac{\mu \partial^2 w_x}{\partial y^2} = -F_R. \tag{2}$$

Solution of this system yields an expression for determination of the film thickness and heat exchange intensity

$$\delta_R = \sqrt[3]{\frac{4\lambda_\ell \mu_\ell (T_s - T_w) x_R}{r\rho_\ell F_R}}, \tag{3}$$

$$\alpha_R = \sqrt[3]{\frac{r\lambda_\ell^2 F_R}{4\nu_\ell (T_s - T_w) x_R}}, \tag{4}$$

where F_R is the force equivalent to gravity and friction; x_R is the condensate path over the heat exchange surface for film motion from the upper to the lower directrix of the tube.

The equations obtained, like Nusselt's solution, reflect the influence on heat exchange intensity of the force F_R, acting along the condensate motion path x_R. The only difference is that we consider not the dominant effect of either gravity or friction, but the value of their equivalent, with consideration of the orientation of the heat exchange surface in space.

The numerical value of the force of gravity is determined by the thickness of the film [12], moving under the influence of the acceleration of free fall, by the equation

$$F_g = \rho_\ell g \delta \sin\varphi, \tag{5}$$

where φ is the angle, measured from the upper tube directrix.

The mean value of the gravitational force in the tube section is equal to

$$\bar{F}_g = \int_0^{\pi/2} \rho_\ell g \delta \sin\varphi = 0{,}637 \rho_\ell g \delta. \tag{6}$$

The value of the friction force applied to the condensate film is determined by the stress at the vapor–film boundary and is equal to:

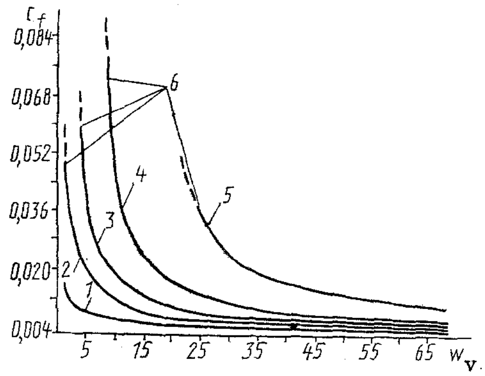

Fig. 4. Numerical value of interphase friction coefficient vs vapor velocity and thermal flux density at T_s = 373 K: 1) interphase friction coefficient for single-phase flow; 2-5) calculation by Eq. (10); 2) $\bar{q}_L$ = 50 kW/m², 3) 100, 4) 200, 5) 400, 6) (dashed lines), regions not calculable by Eq. (10).

$$F_\tau = \frac{c_f \rho_v w_v^2}{2}. \tag{7}$$

The value of the equivalent force (Fig. 3) can be determined analytically from the known F_τ, F_g and β:

$$F_R = \sqrt{F_\tau^2 + F_g^2} = \sqrt{\left(\frac{c_f \rho_v w_v^2}{2}\right)^2 + (\rho_\ell g \delta \sin\varphi)^2} \tag{8}$$

or

$$F_R = \frac{F_g}{\cos\beta}. \tag{9}$$

For condensation of the moving vapor at the beginning of the process, turbulent flow of the vapor flux with laminar film flow is possible. In this case [14, 15], the friction coefficient c_j in Eq. (7) can be determined with consideration of the transverse mass flux occurring due to vapor motion toward the heat exchange surface:

$$c_f = c_{f_0} \frac{(1 - 0{,}25b)^2}{(1 + 0{,}25b)^{0,2}} \tag{10}$$

(c_{f_0} is the friction coefficient for a single-phase flow), while this transverse velocity component $w_{vx} = q/r\rho_v$ for condensation of saturated vapor at atmospheric pressure is an order of magnitude smaller than the longitudinal component, and for example, at q = 500 kW/m² does not exceed w_{vx} = 0.4 m/sec. Such a vapor velocity does not produce any marked dynamic action on the condensate, and the vapor can be regarded as nonmoving in its flow toward the surface.

The vapor flow rate in any section of the heat exchange tube in Eq. (7) can be determined from the expression

$$w_v = 4qL/r\rho_v d_t \, .$$

Analysis of the calculated data of Fig. 4 shows that consideration of the transverse mass flux has a significant effect on the numerical values of the interphase friction coefficient, especially in the area of low and moderate vapor motion velocities. However, the region in which the proposed Eq. (10) can be used is limited by the value $b = -2J/c_{f_0} = 4$, where $J = q/(r\rho_v w_v)$, as actually occurs at low vapor velocities. This means that in the entire range of condensing flow regime parameters, the numerical values of c_f in the analysis presented can be determined without consideration of transverse mass flow. Meanwhile with growth in thermal flux density the absolute value of vapor velocity at which Eq. (10) becomes incorrect increases, and, for example, at $\bar{q}$ = 400 kW/m² reaches w_v = 23 m/sec (Fig. 4). It follows from this that further study of the effect of transverse mass flow on heat exchange principles is required for the process of condensation within tubes. Moreover, for horizontal tubes the ratio of the friction force to the gravitational force, characterized by tan β (Fig. 3A) defines the direction of application of the equalizing force relative to the upper directrix

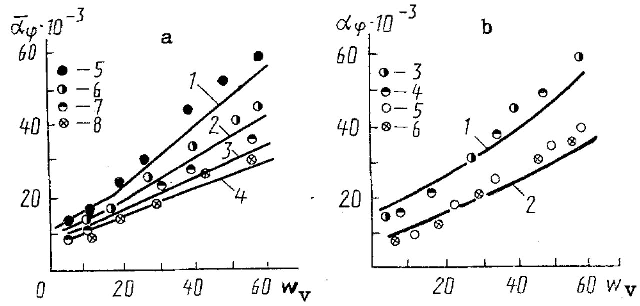

Fig. 5. Comparison of values computed with Eq. (4) with experimental data [10, 11] for various lengths of horizontal tube: lines, calculated heat liberation coefficients; points, experimental data. a) initial tube segment, 1, 5) $\bar{q}_L$ = 50 kW/m²; 2, 6) 100; 3, 7) 200; 4, 8) 400. b) In presence of condensate stream; 1, 3, 4) $\bar{q}_L$ = 50 kW/m²; 2, 5, 6) 200; 3) Re_f = 45; 4) 183; 5) 80; 6) 218.

$$\tan \beta = \frac{F_\tau}{F_g} = \frac{c_f \rho_v w_v^2}{2\rho_\ell g \delta \sin \varphi}. \tag{11}$$

It must be considered that removal of the condensate film changes the numerical value of the gravitational force. At present, the literature lacks data which would permit evaluating the effect of liquid removal and deposition upon change in thickness, and thus, upon the direction of the condensate film flow. This is an important problem requiring its own solution.

We will now analyze the change in the angle β at which the resultant force F_R is applied and correspondingly, the value of the condensate path length x_R along the heat exchange surface using the example of water vapor condensation at atmospheric pressure for attached film flow with change in thermal flux density $\bar{q}_L$ = 50...400 kW/m².

According to the calculated data (Fig. 3B) the greatest change in angle occurs upon increase in vapor motion velocity to w_v = 20 m/sec and reaches the value Δβ = 40°. Thus, the maximum change in β occurs in the range of condensing flow velocity most often found in industry. Increase in vapor velocity above w_v > 20 m/sec causes an insignificant increase in the numerical values of the angle β. For example, at q_L = 50 kW/m² increase in vapor velocity in the range w_v = 20...40 m/sec produces an increase Δβ = 18°, while further increase to 60 m/sec causes an increase Δβ = 5° to the maximum value β = 85°.

The effect of thermal flux density for w_v = const on the value of β is most significant in the same vapor velocity range (w_v = 0.5...20 m/sec). Thus, an increase $\bar{q}_L$ = 50...400 kW/m² reduces β by 15...20°, since with increase in q_L the effect of gravitational forces on the film increases. It follows from Eq. (11) that for any increase in vapor velocity the angle will not reach the value β = 90° because the increasing role of gravity is not eliminated during condensation.

The above indicates that within a horizontal tube a purely annular regime of condensate film flow does not exist, and in all condensation regimes there is a cross-flow of condensate along the tube perimeter due to the effect of gravity. Only the length of the tube section over which the film reaches the region of the condensate stream varies. One can consider the condensate film flow regime annular only with a certain arbitrariness, if that assumption provides the necessary accuracy in determining heat exchange intensity.

On the other hand, knowing the range of change of the angle at which the equivalent force is applied, one can characterize the change in condensate path along the heat exchange surface under the action of the applied forces

$$x_R = \frac{\pi D}{2 \cos \beta}. \tag{12}$$

Calculations performed with Eq. (12) for two tube diameters d_{in} = 18 and 30 mm for β = 0...85° and $\bar{q}_L$ = 50...400 kW/m² show that the value of the characteristic path changes (x_R = πD/2...6πD) by a factor of more than 10 times.

In case it is necessary to determine the heat exchange intensity on a segment of the tube shorther than the value x_R (Fig. 3A), its characteristic dimension can be determined from the expression

$$l_{seg} = \frac{x_R}{\sin\beta}. \quad (13)$$

It follows from the analysis presented that with growth in vapor velocity the condensate path increases due to its inevitable motion to the lower directrix. In other respects, the principle of increase in film thickness and reduction in heat liberation intensity is analogous to the situation for a vertical surface. In connection with this, calculation of a mean heat liberation coefficient α_{x_R} within a horizontal tube can be accomplished analogously to the calculation for a vertical tube of length L:

$$\bar{\alpha}_{x_R} = \frac{1}{x_R}\int_{x_1=0}^{x_2=x_R} \alpha_{x_R} d_{x_R} = 0{,}945\sqrt[3]{\frac{r\rho_\ell \lambda_\ell^2 F_R}{\mu_\ell (T_s - T_w) x_R}}. \quad (14)$$

At the current state of studies the validity of the proposed method can be confirmed by comparing experimental heat liberation coefficients [10, 11] with values computed with Eq. (4) for various horizontal tube sections, as shown in Fig. 5. According to the data presented, calculated and experimental α_φ values agree for a tube with diameter $d_{in} = 0.018$ m for vapor velocities $w_v < 15$ m/sec when a condensate stream is absent (Fig. 5a) while the computations are elevated as the stream perimeter increases (Fig. 5b).

With growth in vapor velocity, increase in the divergence is the result of deviation of x_R from the value $x_R = 40$ mm, which was used for determining local α_φ, and the presence of liquid removal into the vapor flow.

The proposed calculation method not only permits determination of the efficiency of heat exchange for various ratios of the gravitational force to friction, but also allows determination (Eq. (12)) of the optimum horizontal tube length as a function of the condensing flow parameters. However, the computation equation (14) is valid for a laminar attached condensate film flow ($Re_f < 450$, $w_v < 15$ m/sec) and in the case where the effect of the condensate stream may be neglected.

A deeper and more universal experimental and theoretical study of the principles of condensate film hydrodynamics will permit determination of the characteristic dimension and condensate film thickness which determine the intensity of the process within a horizontal tube, thus supplementing the computation method proposed herein.

NOTATION

λ, thermal conductivity coefficient, W/m·K; α, heat liberation coefficient, W/(m²·K); r, heat of phase transition, kJ/kg; q, thermal flux density, W/m²; T, t, medium temperature, K (°C); ρ, medium density, kg/m³; δ, condensate film thickness, m; w, velocity of medium motion, m/sec; g, acceleration of gravity, m/sec²; μ, coefficient of dynamic viscosity, Pa·sec; ν, coefficient of kinematic viscosity, m²/sec; d, tube diameter, m; L, ℓ, length, m; φ, β, angular coordinates; c, interphase friction coefficient; F, force, N/m²; x, current coordinate, m; b, permittivity parameter; j, dimensionless substance flow. Dimensionless complexes: $Nu = \alpha\delta/\lambda_\ell$, Nusselt number; $Re_f = qL/(r\mu)$, film Reynolds number. Subscripts: in, internal; f, film; v, vapor; ℓ, liquid; w, wall; s, saturation; o, one-phase; inp, value at tube input; seg, tube segment; R, value with equivalent force acting; g, value with gravitational force acting; τ, value with interphase friction acting; x, transverse velocity of vapor motion toward heat exchange surface.

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ANALYTIC SOLUTIONS FOR THE PROBLEM OF CONDENSATION OF A TWO-COMPONENT GAS MIXTURE

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UDC 536.48

For describing the process of condensation of a two-component gas mixture on a cryopanel, a mathematical model is proposed, for which analytic solutions are obtained under various proposed simplifications. Numerically computed solutions are compared with experimental results.

1. Investigations of the process of condensation of a two-component gas mixture into the liquid phase have been made in a series of studies [1-5]. Experimental studies have, for example, been described in [1, 3, 4]. Based on these, the qualitative behavior of the system has been studied and empirical formulae proposed.

In [2], a study was made of gravity-flow film condensation, in which the liquid condensate flowed down an inclined cooled surface and was removed from the system. This enables a study to be made of the steady-state working of such a system, and the construction of a self-modeling or numerical solution of the boundary value problem. In [5], a similar problem is considered on the basis of correlations deriving from the conservation laws. The solution is less accurate, but the basic qualitative rules can be traced from it. In these investigations, however, a steady-state process was studied, in which the concentration field, temperature and other parameters of the boundary layer were time independent. In practice, on the cryopanels of cryogenic condensation pumps, cooled to 15-17 K, there is formed during the condensation of gas mixtures a liquid film precipitating on the cryodeposit. The layer of condensate grows continuously, which brings about an important change with time in the remaining characteristics of the system. Therefore, it is necessary to construct a nonstationary model for describing the process.

2. The construction of a mathematical model of such a process, describing its dynamics for a wide range of input parameters (system pressure, temperature head, overall running time of the process), must in its implementation take into consideration free convection in the gaseous region [2]. If, however, the temperature head in the system (temperature difference between that of the gas far removed from the cryopanel T_∞ and that of the cryopanel T_p is sufficiently great then the contribution from free convection may be neglected, and we can limit ourselves to a spatially uniform model.

For describing the processes of heat and mass transport in the gaseous region, we take a system of coordinates, the origin of which coincides with the boundary of the phase interface (Oy-axis in Fig. 1). Although such a system is not static ($h(\tau)$ being the law of motion

Balashikhin Scientific Production Organization of Cryogenic Machine Construction. Translated from Inzhenerno-fizicheskii Zhurnal, Vol. 62, No. 1, pp. 24-30, January, 1992. Original article submitted January 31, 1991.

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