less parameters; $L \equiv d_b/d_0$ and d_b, detachment dimensions of the bubble; d_0, diameter of nozzle orifice; ρ' and ρ'', densities of liquid and vapor; v_0, vapor velocity in nozzle orifice; σ, coefficient of liquid surface tension; a', coefficient of thermal diffusivity for the liquid; c_p', liquid heat capacity; r, heat of vapor formation; $\Delta T = T_s - T_\ell$, temperature head; T_s, saturation temperature; T_ℓ, liquid temperature; R, instantaneous bubble radius; V_0, volumetric vapor blow rate; τ, time; g, gravitational acceleration.

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LAMINAR FLOW REGIME OF A CONDENSATE IN A HORIZONTAL ANNULAR CHANNEL

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UDC 536.248

We present a method for calculating the angle of the rivulet level in the laminar regime encountered in condensation in an annular channel. The theoretical results are compared to the experimental data derived in studying the hydrodynamic and heat exchange in heat-exchange installations of a new type, namely, vapor-dynamic thermosiphons.

Heating tubes and thermosiphons are finding ever-increasing application in the development of contemporary technological processes involving the exploitation of heat from low-potential sources, in the development of cooling systems for equipment subjected to thermal stresses, for radioelectronic installations, and in the exploitation of secondary and alternative energy sources.

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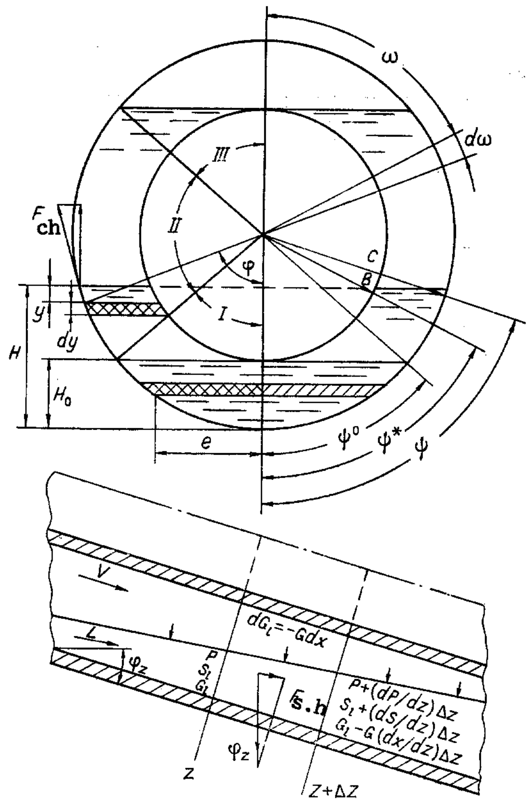

Fig. 1. Geometry of laminar flow in annular channels.

Of significant practical interest are thermosiphons in which the circulation of the coolant can be attributed to their simple and effective design solutions. A particular version of such an installation is represented by the vapor-dynamic thermosiphons (VDTS) proposed in [1]. The advantages of vapor-dynamic thermosiphons become apparent to the greatest extent in the solution of problems involving the heating and thermal stabilization of elongated horizontal objects and in the cooling of radioelectronic equipment [2]. Detailed research into the quantitative relationships governing the hydrodynamics and heat exchange in annular channels in the presence of an outflow of mass (condensation) and with a two-phase characterization of the flow of the vapor-liquid mixture is essential for the successful utilization of the VDTS. In the present study, we will investigate the laminar operational regime for a horizontal VDTS condensor, such as the one most frequently encountered in actual practice.

In one-dimensional approximation, let us examine the model of laminar flow in a coaxial annular channel. Questions dealing with the calculation of the angles of the condensate rivulet levels in horizontal and inclined tubes have been studied in considerable detail in [3, 4]. For the annular channel, we will adopt the same assumptions:

1) the process is treated in one-dimensional approxiation;

2) the film surface is smooth, and its thickness is negligibly small in comparison with the channel diameter;

3) the influence of vapor velocity and that of the condensation intensity on the exchange of heat and on the friction in the flow of the condensing vapor are insignificant;

4) the condensation at the rivulent in the lower portion of the channel is not taken into consideration;

5) nor is any consideration given to the frictional stress at the surface of the condensate rivulet at the boundary separating the vapor from the liquid.

Under these assumptions, for the isolated element of the volume of the annular channel, the continuity equations and those of energy and momentum will be of the same form as in [5]:

$$dG_l = -dG_v = -Gdx, \tag{1}$$

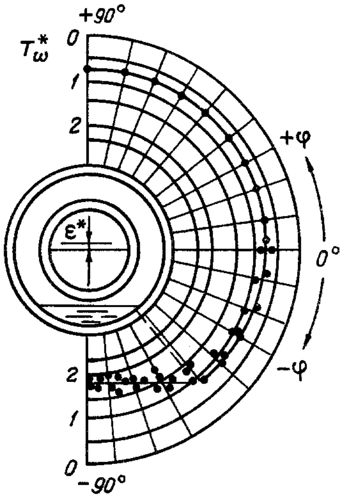

Fig. 2

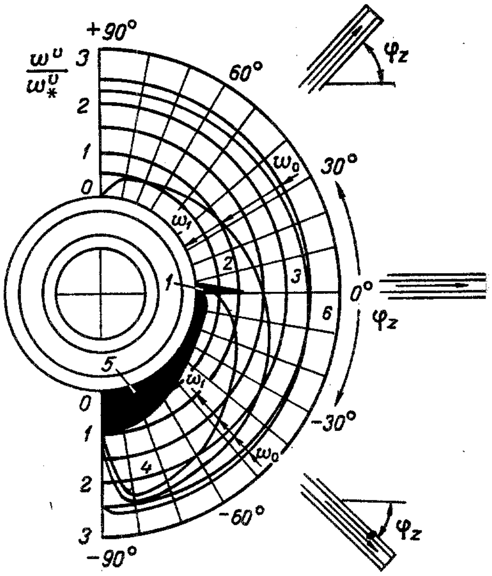

Fig. 3

Fig. 2. Polar diagrams showing temperature distribution about the perimeter of a coaxial condensor at T_v = 120°C and z_i = 3.375 m from the inlet to the channel. Water serves as the coolant.

Fig. 3. Mapping of flow regimes for condensing stream in an inclined annular channel: 1) laminar regime; 2) semiannular; 3) annular, with smooth film; 4) plug-type flow; 5) slug regime; 6) annular regime, with waves at the phase separation surface.

$$2c(\pi-\psi)\alpha\Delta T_{vw}dz = Gr^*dx, \tag{2}$$

$$\frac{\beta_l}{S_l}\frac{d}{dz}(\rho_l S_l w_l^2) - \frac{\beta_v}{S_v}\frac{d}{dz}(\rho_v S_v w_v^2) = \frac{F_l^{\text{fr}}}{S_l} - \frac{F_v^{\text{fr}}}{S_v} + \frac{F_l^{\text{s.h.}}}{S_l}. \tag{3}$$

The latent heat of condensation, with consideration given to the supercooling of the liquid in accordance with Rosenau, is written as follows:

$$r^* = r\left(1 + \frac{0.68c_p\Delta T_{vw}}{r}\right). \tag{4}$$

The heat-transfer coefficient in the case of laminar film condensation is determined from the Nusselt relationship with the Hassan–Jacobi correction factor, taking into consideration the influence exerted by the inclination of the channel

$$\bar{\alpha} = \frac{1}{\pi-\psi}\left[\frac{\rho_l g(\rho_l-\rho_v)r^*\lambda^3}{3c\mu_l\Delta T_{vw}}\right]^{1/4}\left[\frac{4}{3}\int_0^{\pi-\psi}\sin^{1/3}\varphi d\varphi\right]^{3/4}(\cos\varphi_z)^{1/4}. \tag{5}$$

The components of the forces acting on this isolated volume element, according to Fig. 1, are equal to:

1) the force of the rivulet's friction against the wall of the channel

$$F_l^{\text{fr}} = -\xi_l G^2(1-x)^2\,\Pi_l\,(8\rho_l S_l^2)^{-1}, \tag{6}$$

where Π_ℓ and S_ℓ are, respectively, the wetted perimeter and the cross sectional area of the condensate rivulet; ξ is the coefficient of resistance to friction;

2) the force of friction in the flow against the wall of the annular channel

$$F_v^{\text{fr}} = \xi_v G^2 x^2 \Pi_v (8\rho_v S_v^2)^{-1}; \tag{7}$$

3) the static-head force

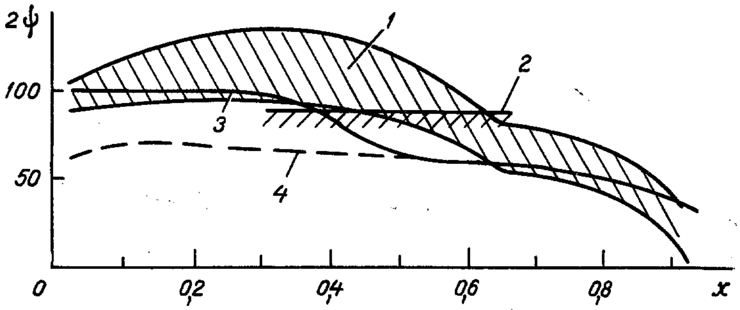

Fig. 4. The angle of the rivulet level as a function of the mass vapor-content flow rate in the condensation of water in a horizontal annular channel when T_v = 120°C; ΔT_{vw} = 1.0°C; Q = 875 W; Bo = 0.9; 1) experimental data; 2) angle of rivulet incidence at the inner tube of the channel; 3) calculation according to (12) with and without consideration of capillary forces, respectively.

$$F_l^{\text{s.h.}} = g\rho_l \sin\varphi_z S_l \Delta z - \frac{d}{dz}\left(\int_0^H 2ey\rho_l g \cos\varphi_z dy\right)\Delta z. \tag{8}$$

When the VDTS condensor is oriented horizontally, the first term (8) is equal to 0, since $\sin\varphi_z = 0$, while the second term may be written as follows

$$F_l^{\text{s.h}} = -\rho_l g c \sin\psi S_l \frac{d\psi}{dz}\Delta z. \tag{9}$$

In this range of geometric dimensions for the channels, with $\psi > \psi_0$, the capillary forces take hold since the Bond number [Bo = $g(\rho_\ell - \rho_v)\delta^2/\sigma$] is less than unity. Their effect becomes apparent when $\psi_0 < \psi < -(\psi_0 - \pi)$. In this case, the second term in expression (8) for the static-head force is written as follows

$$F_l^{\text{s.h}} = F_g^{\text{s.h}} + F_\sigma^{\text{s.h}};$$
$$F_l^{\text{s.h}} = -\frac{d}{dz}\left(\int_0^{H_0} 2eg\rho_l y dy\right)\Delta z + \frac{d}{dz}\left[\int_{H_0}^{H-H_0} \frac{4\sigma\sin\psi}{(c-b)} e dy\right]\Delta z. \tag{10}$$

The capillary component of the static-head force for the second zone of the laminar flow (Fig. 1, II) can be represented in the form

$$F_\sigma^{\text{s.h}} = \frac{2\sigma}{(c-b)}\left[(S_l - S_0)\cos\psi + \sin\psi\frac{dS_l}{d\psi}\right]\frac{d\psi}{dz}\Delta z. \tag{11}$$

After substitution of (5)-(7) and (10) into (1)-(3), and after solution for $d\psi/dz$ and transposition of the terms, we obtain

$$\frac{d\psi}{dz} = \frac{1}{8}\,\frac{\xi_l(1-x)^2 S_v^3 \Pi_l - \xi_v \frac{\rho_l}{\rho_v} x^2 S_l^3 \Pi_v}{\left[(1-x)^2 S_v^3 + \frac{\rho_l}{\rho_v} x^2 S_l^3\right]\frac{dS_l}{d\psi} - \left[\frac{g\rho_l^2}{G^2}(c\sin\psi - \Phi_\sigma)\right] S_v^3 S_l^3} -$$
$$- \frac{2\left[(1-x)^2 S_l S_v^3 + \frac{\rho_l}{\rho_v} x^2 S_v S_l^3\right]\frac{dx}{dz}}{\left[(1-x)^2 S_v^3 + \frac{\rho_l}{\rho_v} x^2 S_l^3\right]\frac{dS_l}{d\psi} - \left[\frac{g\rho_l^2}{G^2}(c\sin\psi - \Phi_\sigma)\right] S_v^3 S_l^3}; \tag{12}$$
$$-\frac{dx}{dz} = \frac{2}{G}\left[\left(\frac{c\lambda_l \Delta T_{vw}}{r^*}\right)^3 \frac{g\rho_l(\rho_l - \rho_v)}{3\mu_l}\right]^{1/4}\left[\frac{4}{3}\int_0^{\pi-\psi}\sin^{1/3}\varphi d\varphi\right]^{3/4}(\cos\varphi_z)^{1/4}.$$

In the first of the equations in system (12) the notation of the relationships for the parameters (Π_i), the cross sectional areas (S_i), and the derivative of the rivulet cross sectional area ($dS_\ell/d\psi$), as well as the expressions for the capillary component of the static head, are different in three characteristic regions in which laminar flow exists:

1) for region (I) with $\psi < \arccos(b/c)$

$$\Pi_l = 2c\psi;\ \ S_l = c^2\left(\psi - \frac{1}{2}\sin 2\psi\right);\ S_v = S_{ch} - S_l; \tag{13}$$

$$\Pi_v = \Pi_{ch} - \Pi_l;\ \frac{dS_l}{d\psi} = 2c^2\sin^2\psi;\ \Phi_\sigma = 0,$$

where Π_{ch} and S_{ch} are, respectively, the perimeter and cross sectional area of the annular channel;

2) for region (II) with $\arccos(b/c) < \psi < [\pi - \arccos(b/c)]$

$$\Pi_l = 2c\psi - 2b\psi^*;\ \Pi_v = \Pi_\text{к} - \Pi_l;\ \psi^* = \arccos\left(\frac{c\cos\psi}{b}\right); \tag{14}$$

$$S_l = c^2\left(\psi - \frac{1}{2}\sin 2\psi\right) - b^2\left(\psi^* - \frac{1}{2}\sin 2\psi^*\right);\ S_v = S_\text{к} - S_l;$$

$$\frac{dS_l}{d\psi} = 2c^2\sin\psi\left[1 - \left(\frac{b^2}{c^2\sin^2\psi} - \frac{1}{\operatorname{tg}^2\psi}\right)^{1/2}\right];\ \psi_0 = \arccos\left(\frac{b}{c}\right);$$

$$\Phi_\sigma = \frac{2\sigma}{(c-b)g\rho_l S_l}\left[(S_l - S_0) - \sin\psi\,\frac{dS_l}{d\psi}\right];\ S_0 = c^2\left(\psi_0 - \frac{1}{2}\sin 2\psi_0\right);$$

3) for region (III) with $\psi > [\pi - \arccos(b/c)]$

$$\Pi_v = 2c(\pi - \psi);\ S_v = c^2\left[(\pi - \psi) - \frac{1}{2}\sin 2\psi\right];\ S_l = S_{ch} - S_v; \tag{15}$$

$$\frac{dS_l}{d\psi} = -\frac{dS_v}{d\psi} = 2c^2\sin^2\psi;\ \Phi_\sigma = 0.$$

The solution of the system of differential equations (12) with consideration of relationships (13)-(15) was performed by Runge—Kutta method at the 4-th order of accuracy. The computational program made provision for preliminary investigation of the region in which solutions for system (12) existed, as well as of the requirements imposed on the specification of the initial conditions and the stability of the calculation. For this, in accordance with the recommendations made in [4, 6], the directional fields were calculated, as well as the form of the curves for the zeros of the numerator and the zeros of the denominator in the right-hand sides of system (12). On the basis of an analysis of the nature of the change in the right-hand sides in the range of regime perimeters being studied here we determined the requirements to be imposed on the specifications of uniquely-defined conditions to ensure the stability of the chosen integration method.

The experimental study of liquid film distribution in a horizontal annual channel was performed on a model of a vapor-dynamic thermosiphon with a condensor made of 12Kh18N9T steel with a length of 4.321 m and a diameter of 0.020 m. The dimensions of the inside tube were 0.012 × 0.001 m. The design of the unit by means of which the condensor of the VDTS model was attached allowed for the turning of the coaxial tubes about the longitudinal axis, thus ensuring the ability to obtain detailed information regarding the distribution of temperature over the condensor surface at various points by means of 24 copper-constantan thermocouples and an automated microcomputer-based calculation-measuring system.

As a result of these experiments, it was established that with an annular shape to the flow of the condensate film in this range of heat flows the temperature distribution about the perimeter of the wall is described by the equation for a circle with a radius R^* displaced downward relative to the axis of the channel through a quantity ε^*:

$$T_w(\varphi) = \varepsilon^*\sin\varphi + [(\varepsilon^*\sin\varphi)^2 + R^{*2} - \varepsilon^{*2}]^{0,5}, \tag{16}$$

where

$$\varepsilon^* = 0.5\left[T_w\left(\frac{\pi}{2}\right) - T_w\left(-\frac{\pi}{2}\right)\right];\ R^* = 0.5\left[T_w\left(\frac{\pi}{2}\right) + T_w\left(-\frac{\pi}{2}\right)\right]$$

are the parameters of Eq. (16), calculated on the basis of the temperatures at the upper and lower generatrices of the annular channel walls at some cross section z_i. Based on geometric considerations, we can generalize the expressions for ε^* and R^* for any values of the angles φ_1 and φ_2. Then

$$\varepsilon^* = \frac{T_w^2(\varphi_2) - T_w^2(\varphi_1)}{2\,[T_w(\varphi_2)\sin\varphi_2 - T_w(\varphi_1)\sin\varphi_1]}\,; \qquad (17)$$
$$R^* = \{[T_w(\varphi_1) - \varepsilon^*\sin\varphi_1]^2 + (\varepsilon^*\cos\varphi_1)^2\}^{0.5}.$$

The experimental data confirmed that when relationships (17) are used Eq. (16) approximates the temperature distribution over the channel wall, even with a a laminar flow regime for the condensate film, with the exception of that portion of the wall perimeter in the sector of the rivulet level angle in the lower portion of the annular channel.

The best interpretation of the flow regime in the construction of the polar diagrams is achieved by the utilization, in place of the quantity $T_w(\varphi)$, of the relationship

$$T_w^*(\varphi) = \{M - [T_v - T_w(\varphi)]\}, \qquad (18)$$

where M is a number expressed in °C to determine the scale of the diagram. In such a representation (see Fig. 2) the course of the curve $T_w^*(\varphi)$ on the polar diagram is similar to the distribution of the condensate film over the perimeter of the annular channel wall. The sharp change in the character of the function $T_w^*(\varphi)$ on transition from the region of laminar film condensation to the region of rivulet flow indicates the weak influence exerted by return flows of heat over the outside condensor wall as a consequence of the low thermal conductivity of the stainless steel. From these considerations, we have the conditions and methology for temperature diagnostics of the structure of a two-phase condensing flow. This involves the following.

1. In using the temperature information to diagnose the condensation regime of a two-phase flow, it is essential, as a minimum, at the section z_i to have three measurements of the wall temperature over the perimeter of the cross section for the angles φ_1, φ_2 and φ_3. In this case, the angles φ_1 and φ_2 are selected in the upper portion of the wall perimeter, e.g., $\varphi_1 = \pi/2$; $\varphi_2 = 0$, while the angle $\varphi_3 = -\pi/2$. Thus, we have to have average values for $T_w(\varphi_1)$, $T_w(\varphi_2)$ and $T_w(\varphi_3)$ and an estimate of the mean-square error in the temperature data at the point $\varphi_3 = -\pi/2$.

2. From the value of the quantities φ_1; $T_w(\varphi_1)$; φ_2; $T_w(\varphi_2)$ in the section $z = z_i$, in conjunction with utilization of relationship (17), we calculate the parameters of the curve showing the temperature distributions of ε_i^* and R_i^*.

3. The criterion for the annular (semiannular) structure of film flow in this case will be the expression

$$\left|\left[T_w\left(-\frac{\pi}{2}\right) - (R_i^* - \varepsilon_i^*)\right]\right| \leqslant \sigma_i^*, \qquad (19)$$

where σ_i^* is the mean-square error in the measurement of the temperature. Failure to satisfy (19) indicates (a necessary condition) the presence of a laminar regime. In this case

$$\left[R_i^* - \varepsilon_i^* - T_w\left(-\frac{\pi}{2}\right)\right] > \sigma_i^*. \qquad (20)$$

4. The extent to which the flow structure approximates an annular structure in fulfilment of condition (19) is determined by the magnitude of ε_i^*, with consideration given to the error in the measurement of the temperature σ_i^*:

$$\begin{aligned} &\varepsilon_i^* \leqslant \sigma_i^* \quad \text{annular regime;} \\ &\varepsilon_i^* > \sigma_i^* \quad \text{semiannular regime.} \end{aligned} \qquad (21)$$

5. The coordinate for the transition from the annular to the laminar flow regime can be determined from the sharp change in the shape of the temperature curve $T_w(z)|_{\varphi=-\pi/2}$ at the lower generatrix of the annular channel.

6. The coordinate for the transition from the laminar flow structure to a plug-type regime in the case of computer processing of the data can be determined (a necessary condition) from the sharp increase in the magnitude of the mean-square error in the temperature changes in this section at the lower and upper generatrices of the channel, due to the pulsations in the temperature of the wall.

7. A sufficient condition for the transitions, based on items 5 and 6 for positive and zero angles of condensor inclination is the prehistory of the flow, i.e., the existence of corresponding film flow regimes in the earlier cross section.

By means of the temperature-diagnostic method for the flow regime of a two-phase flow, we have constructed an approximate chart of the flow regimes in the coaxial condensors of vapor-dynamic thermosiphons in the coordinates φ_z and w_v/w_v^*, as shown in Fig. 3. The quantity w_v^*, in this case, was determined experimentally on a glass model of VDTS to be the minimum velocity of the vapor which results in the reversal of the flow ("flooding") in a vertically rising flow for the channel geometry under consideration. The theoretical validation of the method for the calculation of w_v^* can be found in [7, 8].

Figure 4 shows a comparison of the numerical calculation of the the relationship between the angle of the laminar-flow level in an annular channel and the mass vapor-content flow rate and the experimental results derived on the basis of the above temperature-diagnostic method for the condensation regimes of a two-phase flow relative to the operating conditions of the vapor-dynamic thermosiphons. Analysis of the curves shows that, in calculating the hydrodynamics and heat exchange in a horizontal coaxial condensor, unlike cylindrical channels, a necessary condition for adequate theoretical description of the mechanism of the condensation process is consideration of the capillary forces in the system of differential equations (12). Therefore, the one-dimensional model proposed in this paper, based on the theoretical research of [4-6], additionally makes provision for the unique features of the geometry of the annual channel, as well as of the contribution of the capillary forces in the case of Bond numbers lower than unity. As we can see in Fig. 4, the model describes the experimental results in zones I and II of the laminar flow regime in the condensor of the vapor-dynamic thermosiphon rather well. In the transition region, the experimental data for I lie above the theoretical curve 3. This is explained by the fact that, as a consequence of the periodic formation of the wave on the smooth surface of the rivulet, the surface of the rivulet comes into contact with the surface of the inside tube of the coaxial channel and the liquid is kept in a position that had not been predicted, a fact which can be ascribed to the action of the capillary forces. This same mechanism leads to a delay in the reverse transition between zones I and II, when the theoretical angle of the rivulet level is reduced below the critical value $\psi = \psi_0$. As demonstrated by the experimental data in Fig. 4, the spontaneous change in the nature of laminar flow is possible when $x \le 0.65$, where the probability of the formation of waves of adequate amplitude above the smooth surface of the rivulet is increased. This analysis is in complete agreement with results from visual observations of the dynamics in the development of laminar flow on the glass model of the vapor-dynamic thermosiphon. The results make it possible, with greater validity, to approach the calculation of operational characteristics of vapor-dynamic thermosiphons, as well as for horizontal condensors with annular channel geometry.

NOTATION

a, b, and c are radii of the annular channel, m; G, mass flow rate, kg/sec; H, height, m; F, force, M; S, cross sectional area, m^2; g, gravitational acceleration, m/sec^2; T, temperature; Q, heat flow, W; x, mass vapor-content flow rate; z, axial coordinate, m; w, velocity, m/sec; Π, perimeter, m; α, heat transfer coefficient, $W/(m^2 \cdot K)$; ξ, coefficient of frictional resistance; δ, film thickness, m; ρ, density, kg/m^3; λ, π, and σ are coefficients of thermal conductivity, W/m·K, dynamic viscosity, Pa·sec, surface tension, N/m; ψ, φ, and ω are azimuthal angles, rad; φ_z angle of condensor inclination, rad. Subscripts: v, vapor, ℓ, liquid; w, wall; ch, annular channel.

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