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BOILING OF A LIQUID ON SURFACES WITH POROUS COATINGS

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A semi-empirical model is suggested for description of the curve of boiling on surfaces with porous coatings. Conditions corresponding to a change in the regime of boiling are found.

To enhance boiling processes, use is made of surfaces with porous coatings, which makes it possible to increase the coefficient of heat transfer by a factor of 5–8 compared to smooth surfaces. At present, along with porous coatings, structurized coatings of Gewa and Thermoexcel types are used, which, in contrast to porous coatings, are characterized by regularity of the structure [1, 2]. Numerous experimental studies of heat transfer in vaporization on surfaces with these coatings indicate the existence of considerable differences in the laws governing heat transfer under these conditions as compared to a smooth surface [3].

Depending on the characteristics of the coating, the type of liquid, and the pressure, the characteristic form of the boiling curve can undergo great changes [2, 4, 5].

Several modes of formation of the vapor phase can be distinguished in liquid boiling on surfaces with a developed structure. In the first case, there is an ordinary cycle of nucleate boiling that includes nucleation, growth, and separation of bubbles floating up through the largest pores [6]. When the separation diameter of a bubble is larger than the size of the pore, a regime of boiling is realized in which vaporization inside the porous coating is continuous due to evaporation of the liquid from the surface of menisci: the menisci "suck" the liquid from the volume and the vapor formed escapes through the mouths of pores to the surrounding volume. The existence of stable vapor zones in porous coatings is proved experimentally [7].

It is known that in boiling on a smooth surface, a substantial portion (about 90%) of the thermal load supplied is not spent on the phase transition, but is taken off by convection, which is intensified by the disturbing effect of the bubbles formed. In boiling on porous and structurized surfaces, a portion of the heat is also taken off by convection, but this portion is much smaller than in boiling on a smooth surface. Nakayama et al. [8] studied the boiling of nitrogen, refrigerant R-11, and water on structurized surfaces, whose thicknesses varied within the range of 0.4 to 0.6 mm, and the diameter of the pores at atmospheric pressure was 0.08-0.2 mm. As a result of direct measurement of the density of formed bubbles on the outer part of the coating, the frequency of their formation, and their diameter, the value of the heat flux q_{lat} spent on the phase transition was calculated. The ratio of q_{lat} to the total heat flux is shown in Fig. 1 as a function of q. The same figure presents data for a smooth surface. Heat transfer by vaporization constitutes a much larger portion for a developed surface as compared to a smooth one, with this portion being noticeably dependent on q. Similar dependences are obtained in work [9] concerned with the boiling of refrigerant R-123 at 26.7°C on a structurized surface with a diameter of the pores ranging from 0.18 to 0.23 mm at a coating thickness from 0.4 to 0.9 mm (Fig. 1). The results of [8, 9] have a qualitatively similar character, although the portion of the heat spent on the phase transition in [9] is smaller than in [8]. Proceeding from these data, we can draw the conclusion that at small heat fluxes the main portion of the thermal load supplied is spent on the phase transition, and the heat removal from the outer surface of the coating can be neglected. A model to allow for heat transfer from the outer surface of the coating due to turbulization of the boundary layer is suggested in [10].

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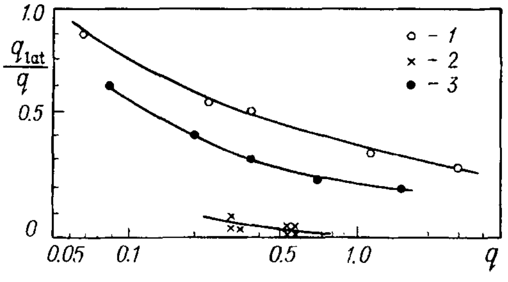

Fig. 1. Contribution of the transfer of latent heat to the total heat flux: 1) $d_p = 0.1$ mm, $h = 0.4$ mm, R-11 [8]; 2) smooth surface, R-11 [8]; 3) $d_p = 0.18$ mm, $h = 0.9$ mm, R-123 [9]. q, W/cm^2.

We consider the boiling of a liquid on a horizontal surface turned upward with a porous coating. The liquid in the main volume is motionless and its mass-mean temperature coincides with the saturation temperature. During boiling, vaporization takes place in the porous coating, and the vapor formed is removed to the main volume of the liquid. The porous coating exerts resistance to the filtered vapor; here the pressure drop with respect to the vapor can be found by Darcy's law:

$$\Delta P_2 = \frac{\mu_2 h \omega_2}{k K_2} . \tag{1}$$

The vapor velocity is related to the thermal load by the relationship $\omega_2 = q/L\rho_2$, and then, with account for (1),

$$\Delta P_2 = \frac{\mu_2 h q}{k K_2 L \rho_2} . \tag{2}$$

One of the limiting schemes [11] where it is assumed that the liquid in the porous matrix is at rest and the vapor moves is taken to specify the coefficient of relative phase permeability; in this case,

$$K_2 = \frac{\varepsilon^2 \varphi^2}{(1 - \varepsilon\varphi)^2} . \tag{3}$$

When $\varepsilon\varphi << 1$, Eq. (3) can be simplified: $K_2 = \varepsilon^2\varphi^2$. In boiling of water at atmospheric pressure on a Nichrome coating obtained by the method of deposition [12] $\Delta T = 3^{\circ}$C, $q = 2 \cdot 10^4$ W/m^2, $\varepsilon = 0.3$, $\varphi = 0.5$, $\mu_2 = 12 \cdot 10^{-6}$ Pa·sec, $L = 2257.2 \cdot 10^3$ J/kg, $\rho_2 = 0.6$ kg/m^3, $k = 8 \cdot 10^{-13}$ m^2, $h = 0.8 \cdot 10^{-3}$ m; then in accordance with (2) $\Delta P_2 \approx 0.1 \cdot 10^5$ Pa. With account for [4], at a pressure of 343.06 kPa in the system the pressure drop in boiling of R-134a on a surface with a porous aluminum coating can be evaluated using the following data [4, 13, 14]: $\Delta T = 5^{\circ}$C, $q = 10^4$ W/m^2, $\varepsilon = 0.037$, $\varphi = 1$, $d_p = 3 \cdot 10^{-6}$ m, $L = 193 \cdot 10^3$ J/kg, $\rho_2 = 16.8$ kg/m^3, $k = 31.25 \cdot 10^{-3} d_p^2 \varepsilon = 10^{-14}$ m^2, $h = 50$ μm; then, according to (3), $\Delta P_2 = 1.4 \cdot 10^5$ Pa. It was assumed in the calculation of this estimate that $\varphi \rightarrow 1$, and therefore, the reduced finite value of the pressure drop is a lower estimate.

For the Nichrome coating [12] obtained by the method of deposition, the mean diameter of the pores is $d_p = 15 \cdot 10^{-6}$ m, and the capillary pressure drop for water at atmospheric pressure at this case is $\Delta P_c = 0.16 \cdot 10^5$ Pa. If inside the porous structure at the heating surface the inequality

$$\Delta P_c \geq \Delta P_1 + \Delta P_2 \tag{4}$$

holds, stationary zones of vaporization can be formed in the coating. Through some pores the liquid is supplied by capillary forces to the heating surface, and through the others the vapor is evacuated. It is shown in [15] that $\Delta P_2 \approx 10\Delta P_1$, and therefore, pressure losses in filtration of the liquid in the coating can be neglected. In boiling of water on the Nichrome coating [12], condition (4) holds ($\Delta P_c \approx \Delta P_2$) and stationary phase interfaces where vaporization takes place can be formed in the coating.

In boiling of refrigerant R-134a on the aluminum coating [4] the capillary drop is $0.2 \cdot 10^5$ Pa, which is in conformity with the inequality $\Delta P_c << \Delta P_2$. In this case, the scheme of boiling in which vaporization occurs from motionless phase interfaces inside the porous coating cannot be realized. At the heating surface, the vapor pressure reaches a value that leads to expulsion of the liquid from the coating. These conclusions are confirmed by results of direct visual observation [6]. After the coating is dried, the liquid, under gravity and the capillary head, reaches the heating wall and the process recurs. Here the boiling resembles an ordinary cycle of nucleate boiling on a smooth surface.

If we assume that the vapor and the liquid are in a state of thermodynamic equilibrium, then the change in the vapor temperature in moving from the heating surface to the main volume is caused by the pressure drop in accordance with the line of phase equilibrium (the binodal curve). In boiling of water on the Nichrome coating [12] the pressure drop changes, in accordance with the binodal curve, by $\Delta P_2 = 0.125 \cdot 10^5$ Pa as the temperature changes from 100 to 103°C, and according to the estimate made above, $\Delta P_2 = 0.1 \cdot 10^5$ Pa. For refrigerant R-134a at a pressure of 343 kPa, the saturation temperature is 4.4°C. With a temperature head $\Delta T = 5$°C, the saturation pressure changes by $0.62 \cdot 10^5$ Pa, and according to the estimate made above, this is $1.5 \cdot 10^5$ Pa. Thus, the conclusion of the necessity of allowing for the change in the liquid-vapor temperature in moving from the heating surface to the outer part of the coating due to frictional pressure losses is confirmed.

To relate the temperature head and the pressure drop with respect to the vapor, we used the Clausius–Clapeyron law

$$\Delta T = \frac{T_s (\rho_l - \rho_2) \Delta P_2}{\rho_l \rho_2 L} . \tag{5}$$

The model considered below is based on the following assumptions:

- the thermal resistances on the side of the porous skeleton and the liquid film on the capillary walls are disregarded;
- the temperatures of the vapor, the liquid, and the porous skeleton coincide and vary along the height of the coating;
- during boiling the entire thermal load supplied is spent only on the phase transition;
- the coating is a porous matrix, so that to relate the pressure drop and the velocity of the escaping vapor we can use Darcy's law.

On the assumption that the entire thermal load supplied is spent only on the phase transition, we have

$$q = \Psi_1 \omega_2 \rho_2 L , \tag{6}$$

where Ψ_1 is an empirical factor that takes into account that a certain portion of the thermal load supplied is not spent on the phase transition.

The averaged value of the vapor fraction in the porous coating can be represented in the form

$$\varphi \approx \frac{h^2}{\varepsilon} n . \tag{7}$$

Relation (7) is obtained on the basis of the fact that n zones of vaporization are present on unit area of the heating surface. The volume of a vaporization zone in the porous coating is approximately equal to h^3, and the total reference volume without the porous skeleton is equal to the product of unit area by h and ε. To evaluate the vapor fraction, we must determine the density of the vaporization zones on the heating surface. The assumption that with increase in the thermal load their number is determined by the density the active cavities on the heating surface holds. The diameter of an active cavity can be found by the known formula

$$d_a = \frac{4T_s\sigma \cos \Omega}{L\rho_2 \Delta T} . \tag{8}$$

For water at atmospheric pressure at ΔT = 2–5°C, in accordance with (8), d_a = 6–12 μm, and in [12] porous coatings with diameter of the pores in the range d_p = 15–90 μm were used, and therefore, the inequality $d_a < d_p$ holds. In boiling of refrigerant R-134a [4] at T_s = 4.4°C and ΔT = 4.5°C, d_a = 0.68 μm, and the diameter of the pores was d_p = 3 μm. Therefore we can assume that in the process of activation of cavities on the heating surface, the dynamics of their increase does not depend on the presence of the porous coating. The absolute number of active cavities, conversely, is related to the presence of the porous skeleton, since it leads to an increase in the portions of deteriorated wettability, an increase in the roughness of the heating surface, etc. The dynamics of the increase in the number of active cavities as a function of the temperature head on the heating surface is well reflected by the known formula [16]

$$n = \Psi_2 \left(\frac{L\rho_2 \Delta T}{\sigma T_s} \right)^2 . \tag{9}$$

In [17], the number of vapor-withdrawing pores in boiling of refrigerant R-11 on a surface with a structurized coating was counted. Grooves of rectangular cross section were made in the copper substrate, and then the substrate was covered by a thin plate with drilled rows of holes. The experiments were conducted with a pressure variation from 0.04 to 0.23 MPa. As is noted in [17], up to a pressure of 0.1 MPa in the system the process of boiling corresponded to a mode of flooding where one vapor-withdrawing pore corresponds to one nucleation center on the heating surface. The rate of vaporization increases with the pressure in the system, the bubbles in the coating spread, and the number of vapor-withdrawing pores becomes uncorrelated with the number of active nucleation centers. The experimental data of [17] within the pressure range of 0.04–0.1 MPa are well generalized by relation (9), if we take $\Psi_2 = 0.8 \cdot 10^{-5}$.

In evaluation of the pressure drop by Darcy's law, the coefficient of relative phase permeability is taken in the linearized form [18]

$$K_2 \approx \varphi . \tag{10}$$

From relations (1) and (5)-(10), the expression for the law of heat transfer

$$q = \Psi_3 \frac{L^4 k \rho_2^4 \rho_1 h}{\mu_2 \sigma^2 T_s^3 (\rho_1 - \rho_2)\, \varepsilon} \Delta T^3 , \tag{11}$$

follows, where Ψ_3 is an empirical factor that depends on the thermal conductivity of the porous coating.

For the data of [12] in boiling of water at atmospheric pressure on a Nichrome coating obtained by the method of sintering and the results of [4] in boiling of refrigerant R-134a on porous coatings of copper, molybdenum, aluminum, and zinc at a pressure of 343.06 kPa we can take $\Psi_3 = 1.1 \cdot 10^{-5}$.

According to (11), with increase in the heat load the coefficient of heat transfer increases with the coating thickness and the pore diameter. Here, an increasing number of cavities are activated and the vapor fraction in the coating increases. However, it is limited by the inequality $\varphi < 1$; therefore, from relation (7) we can obtain the limit of the value of the temperature head up to which the vapor fraction in the coating can grow:

$$\Delta T < \frac{\sqrt{\varepsilon}\, \sigma T_s}{\sqrt{\Psi_2}\, h L \rho_2} . \tag{12}$$

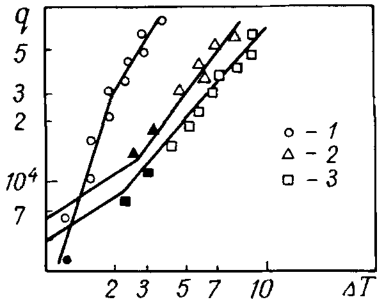

Fig. 2. Curve of boiling on the initial portion for water at $P = 0.1$ MPa on specimens with various porous coatings [12]: 1) sintered coating (h = 0.8 mm); 2, 3) sprayed coatings (h = 0.2 and 0.5 mm); dark symbols, appearance of the first bubbles on the outer surface of the coatings. q, W/m^2; ΔT, °C.

When the value of the temperature head determined by inequality (12) is exceeded, the vapor fraction in the coating reaches its limiting value.

If we use simplified formula (3) [11] to specify the coefficient of relative phase permeability and assume $\varphi \to 1$, then in this limiting case relation (1) takes the form

$$\omega_2 = \frac{k\varepsilon^2 \Delta P_2}{\mu_2 h} . \qquad (13)$$

The assumption $\varphi \to 1$ does not mean that the entire porous coating is occupied by the vapor only. Relation (13) has an asymptotic character (were the liquid absent in the coating, the coefficient of relative phase permeability would be equal to $K_2 = 1$).

From relations (5), (6), and (13) an expression for the law of heat transfer when inequality (12) is not met follows:

$$q = \Psi_4 \frac{k\varepsilon^2 \rho_1 (\rho_2 L)^2}{\mu_2 h T_s (\rho_1 - \rho_2)} \Delta T . \qquad (14)$$

For the data of [4, 12] we can take $\Psi_4 = 0.1$.

Figure 2 presents experimental data [12] for the initial portion of the curve of boiling on two types of coating. The coating obtained by the method of sintering is distinguished by a rather loose structure (porosity 0.4–0.5), whereas a denser structure was produced by the method of plasma spraying (porosity 0.32–0.36). The initial portion of the boiling curve for the loose coating is steeper than for the dense coating.

According to inequality (12), with increase in the thickness of the coating and with decrease in the porosity, the value of the temperature head that corresponds to the limiting value of the vapor fraction in the coating decreases. Therefore, for dense coatings, the law of heat transfer (14) can be satisfied immediately after boiling-up. This conclusion is confirmed by experimental results [19] obtained in boiling of nitrogen and Freon-113. In particular, in this work it is noted that for coatings with a thickness larger than ten diameters of the particles, the character of the dependence of the law of heat transfer approaches a linear one.

According to relations (11) and (14), the effect of the coating thickness on the coefficient of heat transfer is nonunique. At small thicknesses of the coating, when inequality (12) is satisfied, according to (11), an increase in h leads to enhancement of heat transfer. With increase in the coating thickness, the sign of inequality (12) changes and the heat transfer is determined by Eq. (14). According to (14), as h grows, the heat-transfer intensity becomes weaker. This is caused by the increase in the hydraulic resistance of the coating. The conclusions drawn are confirmed by the experimental data of [20]. In [20], boiling heat transfer of propane on tubes with porous coatings deposited by the method of electric-arc spraying was studied. In specimens with the same porosity of the coating ($\varepsilon \sim 12.6$–12.8%) and thicknesses of 0.1 to 0.3 mm, the highest coefficient of heat transfer was obtained on the specimen with a thickness of the porous layer of 0.2 mm. Both an increase and a decrease in the coating thickness led to a reduction in the coefficients of heat transfer. These results are in qualitative agreement with the obtained relations (11) and (14).

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NOTATION

d_a, diameter of the active cavities; d_p, diameter of the pores; h, thickness of the porous coating; L, latent heat of the phase transition; k, coefficient of absolute phase permeability; K_2, coefficient of relative phase permeability with respect to the vapor; n, number of zones of vaporization in the porous coating per unit area of the heating surface; ΔP_2, pressure drop with respect to the vapor; ΔP_1, pressure drop in filtration of the liquid; ΔP_c, capillary pressure drop; q, heat-flux density; q_{lat}, heat flux spent on the phase transition; ΔT, temperature head; T_s, saturation temperature; Ψ_i, empirical constants; Ω, wetting angle; ε, porosity; φ, vapor fraction; μ, dynamic viscosity; ρ, density; σ, coefficient of surface tension; ω, velocity. Subscripts: 1, liquid; 2, vapor.

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