Using a simulator with $k = 18$ millidarcy, we also performed experiments to determine the effect of a porous medium on the phase transitions of propane in a mixture with a small amount (10-15%) of nitrogen or carbon dioxide. The presence of an inert gas does not affect the above-mentioned phase transitions at partial pressures lower than the saturated-vapor pressure.

In addition to the generally accepted parameters — the porosity, permeability, the particle size distribution curve, and the porometric curve — a porous medium can probably be characterized also by the experimental dependence $s = f(\Delta P)$ or $s = f(P/P_s)$, determined for a certain gas which is readily sorbed.

The above results of the experimental investigation of the effect of porous media on the phase transitions of individual hydrocarbons in the region of direct condensation cannot be used to characterize the processes of retrograde condensation of hydrocarbon mixtures.

NOTATION

P, pressure; P_s, saturated-vapor pressure; ρ, density; V_{si}, pore volume of the simulator; z, compressibility factor; G, quantity of matter; Q, gas volume; s, saturation; σ, weight percentage of the liquid phase; T, t, absolute temperature and temperature in degrees Celsius, respectively; R, gas constant; a, amount of desorbed matter; k, permeability.

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HEAT AND MASS TRANSFER IN UNDEVELOPED BOILING IN HEAT-TRANSMITTING SLOT CHANNELS

P. A. Novikov, L. Ya. Lyubin, and É. K. Snezhko

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One of the variants of the problem of thermostatic control of objects involving quasistationary boiling of a heat-transfer medium in capillary gaps is examined.

In [1, 2] we considered the problem of the temperature control of some objects heated unilaterally by radiation by the use of narrow heat-transmitting channels filled with a subliming heat-transfer medium and obtained equations for engineering calculations of their optimum geometric characteristics.

In the case where the heat regime of operation of such devices is altered and the thermodynamic parameters of the medium are above the triple point, they can function as ordinary heat pipes in which effective heat conduction is obtained by a double phase change and return of the liquid phase to the evaporation zone. The role of the wick in this case is played by the closed slot channels partially filled with liquid. On the heat-receiving region of the channel walls vapor bubbles will arise and will grow. The breakaway of the bubbles

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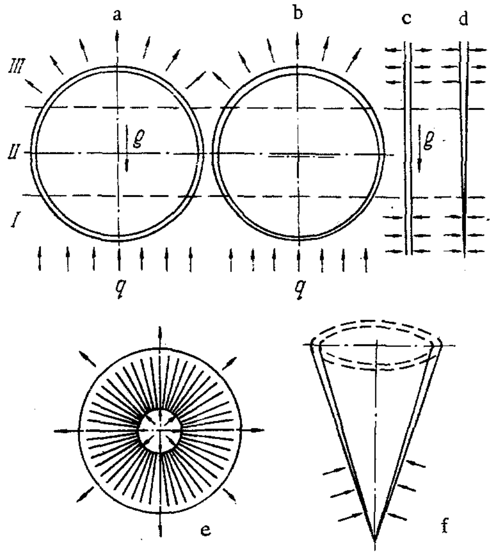


Fig. 1. Heat-transmitting slot channels: a, b) cylindrical and spherical; c, d) flat; e, f) conical and radial.

and their movement into the condensation zone may be due either to the effect of external forces (centrifugal, gravitational, electromagnetic, etc.) or to capillary potential (in wedge-shaped gaps).

Figure 1 shows various forms of frequently encountered slot channels of constant (a, c) and variable (b, d, e, f) cross section with an evaporation zone (I) and condensation zone (III), and the motion of liquid and vapor between these two zones (II).

In the present paper we confine ourselves to an investigation of the effectiveness of such heat-transmitting devices in separation of the heat-transfer medium in the "hot" zone of a plane slot gap. The calculation for the condensation zone will be similar.

To assess the effectiveness of various methods of phase separation we can use the results of [3] and Eqs. (2.3.26) and (2.5.26) from [4], which give the velocity of bubbles flattened between the gap walls, at $Re \ll R_0/h_0$, when the motion of the bubbles can be regarded as quasistationary.

The velocity of a bubble flattened in a slot channel filled with an effectively inviscid liquid (large Re) is determined by the relative strength of the mass and capillary forces. The results of the corresponding experimental investigations are approximated by the formulas [5]

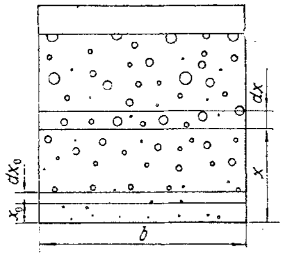
$$\sqrt{Fr} = 6.2\sqrt{Bo} \quad \text{for} \quad Bo < 0.055,$$
$$\sqrt{Fr} = 1.5 \quad \text{for} \quad Bo > 0.055.$$

The effect of capillary forces on the shape of a bubble of large volume is negligibly small. Such a bubble moves in the form of a circular segment, oriented in the direction of motion, with velocity $u = 0.5\sqrt{gR_0}$ [6].

Questions relating to boiling of a liquid medium and the mechanism of bubble formation are dealt with in detail in [8-10].

In some cases the medium used is a liquid whose saturation temperature at the permissible pressures differs only slightly from the temperature maintained in the temperature-controlled volume. If the coefficient of heat transfer to the evaporating liquid is not too high, we can expect that in the latter vapor will be slowly produced by undeveloped boiling, i.e., almost all the space within the gap will be heated practically to the temperature of the surroundings. Only in relatively small volumes close to the bubbles growing by evaporation will the liquid temperature differ slightly from the channel wall temperature T and fall within the limits of the thermal boundary layer to approximately the saturation temperature T_S. It is obvious that the average distances between the bubbles at sufficiently high velocities of translational motion will be much greater than their diameters. We can further assume that the supply of liquid to the considered zone corresponds to its consumption due to evaporation, i.e., in the channel a given liquid level will be maintained.

The study of the kinetics of the considered process reduces in a first (linear) approximation to solution of the following problem. A portion of the capillary gap of width b and length (along the x axis) L is filled with liquid (Fig. 2).



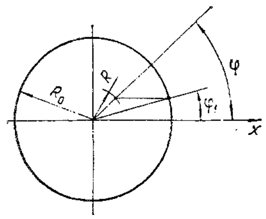


Fig. 2 Fig. 3

Fig. 2. Plan view of portion of channel in evaporation zone.

Fig. 3. Bubble coordinates.

We assume that the inner surface of the gap in the evaporation zone (with the exception of small regions near the bubbles) has a practically constant temperature T_W, exceeding the saturation temperature T_S by an amount ΔT ($\Delta T > 0$) at the given pressure. **Fulfillment of this condition is ensured by the thickness and thermal** conductivity of the wall material, an appropriate method of supplying heat to it, and the small superheat of the liquid in the gap, ensuring at real bubble velocities $u(x)$ sufficiently large ratios of average distances between the bubbles to their diameters over the whole considered portion.

It is obvious that bubbles flattened between the gap walls will be initiated at vapor-forming centers in various regions of the gap, will grow to a certain size, and will then move off in the direction of the acting forces (in the direction of the x axis), becoming larger as they go.

Then the average number of bubbles beginning translational motion in unit time from unit area of wall in contact with the liquid at the considered instant will be given approximately by the formula

$$N = 2kn_0/t_0,$$

where k is a coefficient allowing for the possibility of coalescence of several bubbles formed near close vapor-forming centers ($k \le 1$).

On this surface, characterized by microirregularities, the presence of scale, and gas adsorbed in pores, the number of vapor-forming centers n_0 will be determined by the liquid superheat close to the wall [7-8], and if the superheat is constant along the wall, the number n_0 at a given temperature head ΔT will also be constant.

The period of bubble formation consists of two stages: $t_0 = t_1 + t_2$, where t_1 is the time of nucleation and growth of the bubble to the start of translational movement; t_2 is its time of departure from the vicinity of the vapor-forming center. The bubble growth rate is particularly low at the initial stage, when its radius is very small. Hence, we will assume that the kinetics of bubble formation is almost entirely determined by this initial stage (i.e., along the gap t_0 = const).

Then N will be independent of the local gap width (of the coordinate x, N = const), and the average number of bubbles formed in an elementary layer (of width dx_0 and at a distance x_0 from the initial cross section) and simultaneously occupying an elementary layer dx at a distance x from the initial section will be

$$dN_0 = \frac{bN\zeta(x_0)}{u(x)}\, dx_0 dx. \tag{1}$$

We assume that the motion of the bubbles is quasistationary and depends only on the coordinate x, and the maximum size of the bubbles is small in comparison with the length of the portion in which the parameters η and ζ characterizing the process change appreciably. This means that we are considering undeveloped forms of boiling, corresponding to relatively low temperature heads ΔT and low coefficients of heat transfer from the external medium to the liquid.

Thus, the area of wall surface adjoining the vapor in portion dx will be

$$dS_1 = \left[\int_0^x S(x, x_0)\, \frac{bN\zeta(x_0)}{u(x)}\, dx_0\right] dx.$$

Here $S(x, x_0)$ is the area of contact with the wall of the bubble arriving in section x from section x_0. But $dS_1 = \eta b dx$; hence, the coefficient η will satisfy the following second-order Volterra-type integral equation:

$$\eta(x) + \int_0^x K(x, x_0)\, \eta(x_0)\, dx_0 = F(x), \tag{2}$$

where

$$K(x, x_0) = \frac{NS(x, x_0)}{u(x)}; \quad F(x) = \int_0^x K(x, x_0)\, dx_0.$$

The supply of heat to the phase-change surface is almost entirely determined by large bubbles, whose characteristic size in the plane of symmetry is much greater than the gap width, i.e., a considerable proportion of the bubbles manage to grow to such size during their lifetime. Then we can ignore the change in the mean curvature of the phase interface with bubble growth and assume that vapor production occurs at practically constant pressure and that the rate of growth of a bubble moving in the gap, as in the case of surface [11] and pool [12] boiling, will be determined entirely by the rate of supply of heat to it, i.e.,

$$\frac{dV}{dt} = \frac{Q}{r\rho''}. \tag{3}$$

The surface adjacent to the bubble moving in the channel will be separated from it by a thin layer of liquid. Hence, the heat flux to the bubble is $Q = Q_1 + Q_2$, where Q_1 is the heat supply from the main mass of liquid, and Q_2 is the heat supply from the wall through this thin layer. The flux Q_2 is similar to the heat flux through the microlayer in surface boiling in a large volume [13].

To assess the flux Q_1 we calculate the heat supply to a cylindrical surface of radius R_0, on which a temperature T_s is maintained, from liquid filling a slot channel. Since $R_0 \gg h_0$, then at $Pe \ll 1$ (we have in mind nonmetallic liquids at $Re \ll 1$ and metallic liquids at practically any Re) the determination of Q_1 reduces in practice to the solution of the plane problem for the function T, harmonic in the region $0 < y < \infty$; $-h_0 < Z < h_0$, and satisfying the following boundary conditions:

$$\lambda dT/dZ = \pm\alpha(T_w - T) \text{ when } Z = \pm h_0; \quad T = T_s, \text{ when } y = 0.$$

The first of these conditions is satisfied, for instance, when the liquid is heated through a sufficiently thin wall, and its thermal resistance and the longitudinal flow of heat in it can be neglected. In this case T_W corresponds to the temperature of the surrounding gaseous medium, and the heat flux is

$$Q_1 = 16\pi B\lambda R_0 (T_w - T_s),$$

$$B = \sum_{k=1}^{\infty} \frac{\sin^2 \nu_k}{2\nu_k + \sin 2\nu_k}, \quad \nu_k \operatorname{tg} \nu_k = \mathrm{Bi}.$$

At Taylor numbers $T \lessapprox 3 \cdot 10^{-3}$ the thickness of the layer left behind the receding meniscus can be evaluated analytically [4]. Suo and Griffith [14] gave the results of an experimental investigation of the relative thickness δ/r_0 of the layer of liquid left on the walls of a cylindrical tube of radius r_0 when a large bubble moves through it, in relation to the dimensionless parameters T and $G = \mu^2/(\rho\sigma r_0)$ for a wide range of variation of these parameters. At sufficiently large translational velocities we can assume that in the time of movement of the bubble through a distance $2R_0$ the thickness of the layer is practically unaltered by evaporation.

As was shown in [4], when $R_0/h_0 \gg 1$ there is practically no boundary layer in flow around a bubble flattened between gap walls, i.e., the quasipotential form of flow in the gap is not affected by approach to the phase interface and, hence, the characteristic scale of such a flow is the gap width; in a linear approximation the general case of flow of a liquid in a gap with a moving bubble present in it can be represented as the superposition of a flow corresponding to quasipotential flow over a stationary bubble and quasi-two-dimensional flow in planes perpendicular to the plane of symmetry of the gap. The second flow has a distinct boundary layer and the liquid velocity here changes (along the phase interface) from zero to a value on the order of the velocity of the receding meniscus within a layer of thickness $l \ll h_0$.

Since at a distance l the velocity component tangential to the outline of the bubble does not manage to increase significantly from the zero value which it had on the gap walls, we can assume that the formation of the layer left on the walls by the receding meniscus will be affected only by the velocity component normal to the bubble outline. We have in mind that the bubble velocity is not too great, so that the thickness of the residual film is small in comparison with the gap width.

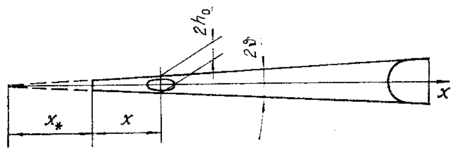


Fig. 4. Diagram for calculation of a wedge-shaped heat-transmitting channel.

Then, taking into account the quasistationary nature of the bubble motion and the inequality $R_0 \gg h_0$, we find, using expression (1.5.15') from [4], the change in thickness of the film separating the bubble from the wall in a direction perpendicular to the bubble velocity,

$$\delta_n = \delta_0 (\cos\varphi_1)^{2/3}, \quad \delta_0 = 1.31 \left(\frac{\rho\mu u}{\rho'\sigma}\right)^{2/3}, \quad \varphi_1 = \arcsin\left(\frac{R}{R_0}\sin\varphi\right),$$

where R, φ are the polar coordinates in the plane of symmetry of the gap with the origin at the center of the bubble (Fig. 3).

The mean (over the area) coefficient of heat transfer from the surrounding medium to the free film surface can be calculated from the formula

$$K_M = \frac{4}{\pi}\int_0^{\pi/2} \frac{\cos^2\varphi_1 d\varphi_1}{\frac{1}{\alpha} + \frac{\delta_n}{\lambda}}.$$

If the heat-transfer coefficient on the outer surface of the wall is not too high ($\alpha \ll \lambda/\delta_n$), then $K_M \approx \alpha$.

The flow of heat to the bubble through the film is

$$Q_2 = 2\pi K_M R_0^2 \Delta T. \tag{4}$$

When $R_0/h \gg 1$, $Q_2 \gg Q_1$ and the component Q_1 can be neglected.

As a first example we consider the process in a plane-parallel gap (h_0 = const, u_0 = const).

The bubble growth rate in this case is given approximately by the expression

$$\frac{dV}{dt} = 4\pi u h_0 R_0 \frac{dR_0}{dx}. \tag{5}$$

Substituting (5) in (3) and solving the obtained equation, we find

$$R_0(x, x_0) = R_1 \exp\left[\frac{\varkappa(x - x_0)}{2}\right], \quad \varkappa = \frac{K_M \Delta T}{u h_0 r \rho''},$$

where R_1 is the radius of the bubble beginning translational motion in section x_0 [$R_1 = R_0(x_0, x_0)$].

Thus, if we neglect the heat flux Q_1, the coefficient η in this case will satisfy the following convolution integral equation:

$$\eta(x) + M\int_0^x \exp[\varkappa(x - x_0)]\,\eta(x_0)\,dx_0 = \frac{M}{\varkappa}[\exp(\varkappa x) - 1]. \tag{6}$$

Here $M = \pi N R_1^2/u$.

The solution of Eq. (6) has the form

$$\eta = \frac{M}{\varkappa - M}\{\exp[\varkappa - M)\,x] - 1\}. \tag{7}$$

This formulation of the problem assumes that an increase in the surface area of contact between the wall and bubbles is mainly due to bubble growth, and not to the appearance of new bubbles at early stages of growth, i.e., the total area of surface of small bubbles can be neglected. Hence, $M \ll \varkappa$.

The amount of heat supplied in the portion of the channel from 0 to x in unit time is

$$Q_3 = 2r\rho'' bu\eta(x)\, h.$$

The distribution of the time-averaged specific heat fluxes has the form

$$q(x) = 2\pi r\rho'' h_0 R_1^2 N \exp[(\varkappa - M)\, x].$$

In the case of a wedge-shaped slot [$h = (x + x_*) \tan\vartheta$, Fig. 4] the bubble growth rate is approximately

$$\frac{dV}{dt} = u\left(4\pi h_0 R_0 \frac{dR_0}{dx} + 2\pi R_0^2 \frac{dh_0}{dx}\right). \tag{8}$$

If the plane of symmetry of the channel is horizontal, the velocity of the flattened bubble in the direction of the x axis will be practically constant when $\mathrm{Re} \ll R_0/h_0$ ($u = u_M \approx \sigma\vartheta/3\mu$ [3]). When mass forces obviously predominate, the bubble velocity is proportional to $h^2(x)$ [$u \approx u_g = g\rho h_0^2(x)/3\mu$, if $u_g \gg u_M$]. In view of relations (3), (4), and (8) the bubble radius in the first and second case will vary in the following way:

$$\frac{R_0^2(x, x_0)}{R_1^2} = \left(\frac{x + x_*}{x_0 + x_*}\right)^{\Omega},$$

$$\frac{R_0^2(x, x_0)}{R_1^2} = \frac{x_0 + x_*}{x + x_*} \exp\left[\frac{\Lambda}{(x_0 + x_*)^2} - \frac{\Lambda}{(x + x_*)^2}\right].$$

We can assume that in the case of a wedge-shaped channel $R_1 = \Pi_1 h_0(x_0) = \Pi(x_0 + x_*)$, where $\Pi = \mathrm{const}$, $\Pi_1 = 0$ (1).

If, as in the treatment of the process in a channel with parallel walls, we neglect the heat flux Q_1, then for the coefficient ζ we can write a Volterra-type integral equation:

$$\zeta(x) - \omega \int_0^x K(x, x_0)\, \zeta(x_0)\, dx_0 = 1. \tag{9}$$

The parameter ω and the kernel $K(x, x_0)$ in the first and second cases, respectively, have the form

$$\omega = -\omega_M = -\pi\Pi^2 N/u_M,$$

$$K(x, x_0) = (x_0 + x_*)^{\Omega}/(x + x_*)^{\Omega - 2}.$$

$$\omega = -\omega_g = -3\pi\Pi^2 N\mu/\rho g\, \mathrm{tg}^2\vartheta,$$

$$K(x, x_0) = [(x_0 + x_*)/(x + x_*)] \exp[\Lambda/(x_0 + x_*)^2 - \Lambda/(x + x_*)^2].$$

Equation (9) for any values of parameter ω has a single summable value, which can be expressed in terms of the resolvent

$$\left[\zeta(x) = 1 + \omega \int_0^x \Gamma(x, s, \omega)\, ds\right],$$

represented by the convergent power series

$$\Gamma(x, s, \omega) = \sum_{m=1}^{\infty} \omega^{m-1} K_m(x, s), \quad K_1(x, s) = K(x, s),$$

$$K_m(x, s) = \int_0^x K(x, s_1)\, K_{m-1}(s_1, s)\, ds_1.$$

For instance, in the case of horizontal placement of the channel, which is of most practical interest,

$$\eta(x) = 1 - \zeta = \frac{\omega_M x_*^3}{\Omega - 3}(X^{\Omega} - X^3) - \frac{\omega_M^2 x_*^6}{3}\left(\frac{X^{\Omega-3} - X^6}{\Omega - 3} - \frac{X^{\Omega} - X^6}{\Omega - 6}\right) + \ldots, \quad X = 1 + \frac{x}{x_*}. \tag{10}$$

In evaluating the velocity $u(x)$ of the bubbles throughout the above treatment we ignored their drift due to back flow of the liquid. The velocity of the back flow, averaged over time and over the channel width, is

$$v(x) = \frac{\rho''}{\rho'} \cdot \frac{\eta(x)}{1 - \eta(x)}\, u(x).$$

Hence, neglecting the bubble drift, we introduce a relative error [4]:

$$\frac{\Delta u(x)}{u(x)} = -2\,\frac{\rho''}{\rho'}\cdot\frac{\eta(x)}{1-\eta(x)},$$

which is obviously negligible if the coefficient $\eta(x)$ is not too close to unity.

NOTATION

T_s, saturation temperature; T_ω, temperature of inner surface of channel walls except for small regions adjacent to bubbles (it is practically the same as the temperature of the evaporation zone outside); $\Delta T = T_\omega - T_s$; ρ', ρ'', densities of liquid and vapor, respectively; μ, a, λ, dynamic viscosity, thermal diffusivity, and thermal conductivity of liquid; α, coefficient of heat transfer from external medium to inner surface of channel wall; Q, heat flux to bubble; Q_1, Q_2, components of heat flux from main mass of liquid and through thin layer separating bubble from wall, respectively; Q_3, amount of heat supplied in region from 0 to x; K_M, mean heat transfer coefficient, averaged over bubble area; n_0, number of vapor-forming centers per unit area of wall; N, mean number of bubbles beginning translational movement from unit area of wall in contact with liquid in unit time; t_0, period of bubble formation; t_2, time of bubble departure from vicinity of vapor-forming center; x, distance from start of evaporation zone; x_0, bubble nucleation coordinate; b, channel width; $2h_0(x)$, height of slot channel at cross section x; ϑ, angle of inclination of walls of channel of variable section; σ_n, thickness of layer of liquid left on walls; R_0, R_1, instantaneous radius of bubble flattened in the channel and at the start of its translational movement; V, bubble volume; r_0, tube radius; r, heat of evaporation; σ, surface tension coefficient; u, bubble velocity; v(x), velocity of back flow of liquid in section x, averaged over time and channel width; q(x), specific heat flux per unit area of median channel surface, averaged over time and channel width; $g_1 = g + 0.925\sigma\nu/(h_0^2\rho')$; g, density of mass forces in x direction; ζ, η, fractions of wall surface adjacent to liquid and vapor, respectively ($\zeta + \eta = 1$); Π, Π_1, dimensionless constants; z, distance from plane of symmetry of channel; $Re = \rho' uh_0/\mu$; $Pe = uh_0/a$; $Fr = u^2/(g_1 2h_0)$; $Bo = 4h_0^2\rho' g_1/\sigma$; $Bi = \alpha h_0/\lambda$; $T = \mu u/\sigma$; $G = \mu^2/\rho'\sigma r_0$; $M = \pi N R_1^2/u$; $\varkappa = (K_M\Delta T)/(2uh_0 r\rho'')$; $\Omega = (2K_M\Delta T)/(r\rho'' u_M - 2\tan\nu)$; $\Lambda = (K_M\Delta T)/(2r\rho''\gamma\tan\nu)$; $\gamma = (g\rho'\tan^2\nu)/(3\mu)$; $y = R - R_0$; $\omega_g = (3\pi\Pi^2 N\mu)/(\rho' g\tan^2\nu)$; $\omega_M = \pi\Pi^2 N/u_M$; $X = 1 + (x/x_*)$.

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