PHYSICAL MECHANISMS FOR HEAT AND MOMENTUM TRANSFER IN A SHORT LOW-TEMPERATURE HEAT PIPE. II. VAPOR FLOW STRUCTURE

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The authors present results of an experimental investigation of the structure of flow of a condensible vapor flow in a planar heat pipe. They have examined the influence of volume condensation in the vapor stream and in the vapor–gas front region on heat and mass transfer in the heat pipe.

The calculated characteristics of a heat pipe depend significantly on the vapor state model assumed, which must take account of the complete set of physical processes occurring along the heat pipe. An analysis of design models of vapor state was performed in [1]. Most papers assumed a two-phase equilibrium flow, frozen relative to phase transitions. In general the heat and mass transfer process in a heat pipe must be considered as a sequence of achievable metastable vapor flow states in which new stable phases corresponding to r_{cr} can appear.

Phases arise for which $r < r_{cr}$ are unstable and are broken down in the heat and mass transfer process. A new phase can be formed on prepared nuclei introduced into the vapor flow volume, e.g., drops of heat transfer agent thrown off by vaporization from the wick structure. In accordance with the definition in [2, 3] we shall call this process heterogeneous volume condensation. When there are no extraneous nuclei, i.e., when the condensation centers are formed directly in the supersaturated vapor as a result of heterophase fluctuations [4, 5], homogeneous volume condensation is observed [1]. There are possibly transitional regimes when both homogeneous and heterogeneous volume condensation occurs.

Evaporation processes in a heat pipe play a large role in the delivery of prepared nuclei. Thermochemical decomposition of the heat transfer agent and corrosive breakdown of structural materials in long-term heat pipe operation promote the appearance and accumulation of extraneous impurities in the vapor–liquid channel. Continuous distillation of the heat transfer agent leads to accumulation of impurities in the wick of the evaporation zone. The high relative surface energy of these substances increases the role of absorption processes in the mechanism for stabilizing a nucleation bubble during boiling on a heat pipe wick [6]. The presence of such particles in a liquid heat transfer agent significantly reduce the critical heating and facilitates liquid boiling on the wick, which may be accompanied by ejection of the liquid phase into the vapor flow volume [1, 7].

For large heat flux, $q > 10^6$ W/m^2, the evaporation surface may be broken down due to instability of the interphase boundary in phase transformations [8, 9]. A source of interphase instability of superheated liquid may be the dependence of the rate of evaporation on impurity concentration. Also, thermal-capillary [10] and pressure-capillary [11] instabilities are possible.

In a heat pipe with noncondensible gas when the ratio of the molecular weights of the heat transfer agent and the noncondensible gas μ/μ_{MCG}, the vapor–gas front may be a markedly elongated surface separating the basic vapor stream and the noncondensible gas [12]. The reverse vortex motion of the noncondensible gas then arising promotes considerable supercooling of the vapor stream and high degrees of supersaturation.

In formulating the problem of a theoretical investigation of flow of a moist vapor in a heat pipe one meets considerable difficulties associated with the need to calculate a large number of physical phenomena which interact and occur simultaneously in the heat pipe vapor channel. A considerable role here is played by interphase transfer of mass, momentum and energy, by the polydispersion of the liquid phase, by deformation, subdivision

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and coagulation of drops, and by the interaction of the liquid phase with the virtual surfaces of the gasdynamic channel of the heat pipe formed by the vortex flow structure in the evaporator and the condenser [12].

It is extremely laborious task to create a design model for flow of the two-phase medium in a heat pipe, accounting for all the above special features.

However, in the dependence on the specifics of the heat pipe one can select special factors which appreciably affect the heat and mass transfer processes, and then provide them with a reliable experimental basis. Of these an important factor is the process of blowing which achieves acceleration of the vapor stream and creates conditions for supersaturation of the vapor in the evaporator. Removal of heat in the condenser supercools the vapor flux, which also leads to supersaturation. For the blowing and suction processes with phase transitions there are typically significant gradients of density $d\rho/dy$ in a practically isothermal flow field [12] from the standpoint of thermocouple measurements. An experimental investigation of the kinetics of liquid phase formation in the vapor flow of a heat pipe assumes a reliable determination of: the saturation temperature T_s, the distribution of moisture content $\varepsilon = (1 - \gamma)$ or degree of supersaturation $\chi = P/P_s$ in the vapor flow volume, and the distribution function of drops by size $f(r)$. To evaluate the pressure loss in the vapor channel of a heat pipe one must know the relative mass flow rate of liquid phase in the mixture $\varphi = G_\ell/(G_\ell + G_v)$. The combination of these factors creates prerequisites for an experimental investigation of two-phase flows in a heat pipe using optical methods, which do not distort the operating process. Their chief merits are: low-inertia, high sensitivity, higher accuracy than calorimetric methods, and simultaneous recording of the entire field of optical nonuniformities [13].

The heat pipe for the optical investigations has been described in detail in [12].

Solution of the Problem. In the spontaneous condensation region one can neglect slip of the phases, and for degrees of humidity $\varepsilon < 0.3$, the volume of the liquid phase. Then the density can be calculated from the formula

$$\rho = \rho''\gamma^{-1}. \tag{1}$$

The degree of humidity can be determined from the kinetic relations [4]

$$\bar{\omega} = \frac{d(1-\gamma)}{d\tau} = 4\pi\rho' \int_0^\infty r^2 f(r)\,\dot{r}dr + \frac{4}{3}\pi r^3 \rho' \frac{J}{\rho}; \tag{2}$$

$$J = \left(\frac{P}{kT}\right)^2 \sqrt{\frac{2\sigma\mu}{\pi N_A}} \frac{1}{\rho'} \exp\left(-\frac{4\pi\sigma r_{cr}^2 \beta_c}{3kT}\right). \tag{3}$$

To determine the rate of growth of a drop one can use the Hertz–Knudsen formula

$$\dot{r} = \frac{dr}{d\tau} = \frac{\alpha_c P}{\rho' \sqrt{2\pi RT}} \left[1 - \frac{P_s(T)}{P} \sqrt{\frac{T}{T'}}\right]. \tag{4}$$

The critical radius of nuclei forming as a result of heterophase fluctuations is:

$$r_{cr} = \frac{2\sigma\mu}{\rho' RT \ln \chi}. \tag{5}$$

The unknown quantity in Eq. (4) is the condensation coefficient α_c, which, according to [4], can be written as

$$\alpha_c = \frac{C_p}{\lambda} \frac{T_s - T}{1 - 2\sigma/(r\rho'\lambda)}. \tag{6}$$

Again using the method based on replacing the kinetic equation (2) by a system of differential equations for the moments of the distribution function m_i [13], we have:

$$m_i = \int_0^\infty r^n f(r)\,dr, \tag{7}$$

$$\frac{dm_i}{d\tau} = n\dot{r}m_{i-1} + \frac{J}{\rho} r_{cr}^n \tag{8}$$

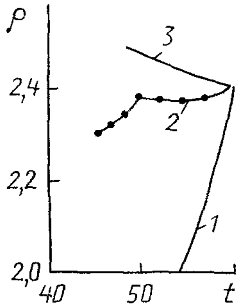

Fig. 1

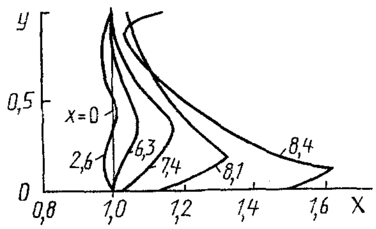

Fig. 2

Fig. 1. Deviation of the vapor flow state parameters from thermodynamic equilibrium: 1) dependence of the density of acetone vapor on the saturation line; 2) experimentally measured value of the density of humid vapor on the axis of the heat pipe, tuning of a Mach–Zender interferometer in an infinitely wide band; 3) calculation of the vapor density on the heat pipe axis on the perfect gas mode; Q = 235 W, initial pressure of noncondensible gas prior to startup $P_{MCG} = 1.2 \cdot 10^4$ Pa, flux of heat transfer agent in the intersleeve gap of the evaporator is similar to the vapor flux, $\Delta t = 3.5°C$, in the condenser the flow is opposite to the vapor flux. The units of ρ are kg/m^3, and t is in °C.

Fig. 2. Experimentally measured distribution of the degree of supersaturation of the vapor flow at different sections of the heat pipe; with $\chi = P/P_s$, $y = y_i/\delta$, $x = x_i/\delta$ (the regime of Fig. 1).

Then the expression for the rate of phase transformations reduces to the form

$$\bar{\omega} = \frac{4}{3}\pi\rho' \frac{dm_3}{d\tau} = \frac{4}{3}\pi\rho' r^2 m_2 + \frac{4}{3}\pi\rho' \frac{J}{\rho} r^3, \tag{9}$$

where the first term determines the rate of phase transformations on injected nuclei, and the second term pertains to nuclei formed automatically from heterogeneous fluctuations.

To determine the distribution function of drops by size we use the method proposed in [14], based on measuring the light scattering index at small angles of attack.

Discussion of the Experimental Results. From interferograms of the flow of moist vapor one can determine the distribution of moist vapor [12]. By extending these data one can determine the characteristics of interest to us: the distribution of humidity, and dryness, and also the level of supersaturation at different sections of the pipe (see Figs. 1 and 2). The flow forms a core on the axis of which the humidity and degree of supersaturation are maximum, and increase downstream.

The experimentally measured distribution of drops is shown in Fig. 3 (curve 1). The large scatter of drops with size $1.0 < r < 200$ μ, for which $\Theta = 2\pi r/l_\lambda > 10$, in the evaporation zone and the high density of drops in the condenser, the brightness oscillations in the working channel of the heat pipe when the drops arrive in the plane of the optical knife-edge, and the reflected highlights from the heat pipe walls caused considerable difficulties in the process of photography, development and subsequent data reduction using the technique suggested in [14]. These authors have estimated the error to be quite large (15 to 40%). In addition, for drops with $\Theta > 10$ the error of the method increases considerably due to a sharp asymmetry of the scattering index, since all of the light is transmitted forward at low angles. On the other hand, according to [2], the solution of Eq. (10) for the six moments of the distribution function m_i has exhibited a negligible contribution from the first term in the region of interest to us, i.e., spontaneous homogeneous volume condensation. Figure 4 (curve 3) shows an estimate of the rate of phase transformations, derived from the second term of Eq. (9). Thus, the degree of supersaturation reached in the experiment for a heat pipe with a noncondensible gas proved to be enough to start the process of homogeneous volume condensation.

With increase of heat flux the ejection of drops in the evaporator increased and the role of nuclei injected into the flow became considerable from the standpoint of heterogeneous volume condensation, computed from the first term of Eq. (9). The drops ejected into

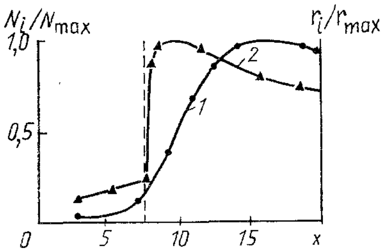

Fig. 3

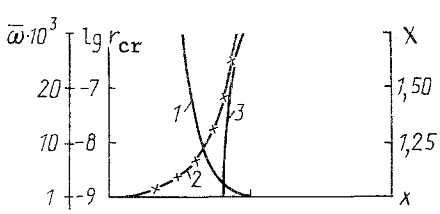

Fig. 4

Fig. 3. Experimentally measured distributions of relative mean drop radius r_i/r_{max} (1) and relative drop concentration N_i/N_{max} (2).

Fig. 4. Kientics of the formation of the liquid phase in the vapor flow of a heat pipe: 1) computed critical radius of nuclei r_{cr} from Eq. (5); 2) variation of the degree of supersaturation on the heat pipe axis, experiment; 3) rate of phase transformations $\bar{\omega}$, computation from the experimental data (conditions of Fig. 1).

the flow moved in a medium with variable temperature and humidity. Their behavior in the evaporator and the condenser was different. The concentration and the drop size in the evaporator became practically constant, which indicates partial vaporization of the drops. Considering the heat pipe as a nozzle with a thermal and an expansion action, we have [4]

$$\left(\frac{dP}{P}\right)_s - \left(\frac{dP}{P}\right)_v = \frac{kM^2}{M^2-1}\left[\frac{1}{\gamma} - \left(1 + \frac{k-1}{2}M^2\right)\frac{\lambda}{C_pT_0}\right]d\gamma. \tag{10}$$

Since $1/\gamma < 1$, and for the measured pressures $\lambda/C_pT_0 \gg 1$, then the term in the square brackets is negative. If we assume that $d\gamma < 0$ (the degree of dryness decreases), then we have

$$\left[\left(\frac{dP}{P}\right)_s - \left(\frac{dP}{P}\right)_v\right] < 0, \tag{11}$$

i.e., the pressure curve for the moist vapor must fall below the curve for the superheated vapor. The experiments of the other authors contradict this conclusion [4], and therefore, from the standpoint of the results of their experiments one must postulate that in our case there is some drying of the moist vapor under the condition: $(dP/P)_s > (dP/P)_v$, i.e., the pressure curve of the moist vapor must be located somewhat above the curve for the superheated vapor. In fact, the drops in our experiment moved into a region of low temperature, and therefore, partial evaporation of the drops must occur.

The situation in the condenser differs in principle. Two vortices, rotating in the blowing-suction direction at the exist from the evaporator form gasdynamic boundaries compressing the flow [12]. The drops move along these boundaries, which have the profile of a subsonic nozzle. The drop concentration in the flow core increases sharply, the humidity increases, and the process of heterogeneous–homogeneous volume condensation begins and develops along the flow. The concentration of drops and the drop size increase sharply in the computed region of homogeneous volume condensation (Fig. 3).

The adjustment of the vortex flow structure is accomplished at the beginning of the condenser within one caliber (one heat pipe channel width), and a fine-scale vortex motion is established, consisting of a system of longitudinal vortices, of scale 1 mm [12]. In this context it is appropriate to examine the question of possible formation of condensate due to turbulence. In turbulent flow there is always a nonzero probability of forming a metastable state. Considering the development of turbulence as an hierarchy of vortices of different orders, in which the vortices of a given order arise due to loss of stability of larger vortices of the previous order, borrowing energy from them, and in turn giving rise to smaller vortices of the next order, we can hypothesize that in the interior of these vortices the necessary conditions are created to form the condensed phase. To evaluate this possibility we use the results of the development of this idea in [15]. We consider the dimension of the mean radius of a drop of condensate $\langle r_T \rangle$ formed in turbulent flow:

$$\langle r_{\text{т}} \rangle = A\frac{\langle P \rangle}{2\rho_{\ell}\sqrt{2\pi R\langle T \rangle}}\,\frac{\langle \Delta T \rangle}{\langle T \rangle}\,\frac{\Omega}{\eta U}, \tag{12}$$

where A is the experimentally determined value (according to the available data $A \sim 10^{-2}$); $\eta = \psi/U$ is the level of turbulence; and $\psi = \sqrt{1/3\ (\overline{U}'^2 + \overline{V}'^2 + \overline{W}'^2)}$; U is the mean flow velocity.

The mean radius $\langle r_T \rangle$ obtained from Eq. (12) should be comparable with the value r_{cr} of the critical radius for a mean value of supercooling in the medium $\langle \Delta T \rangle$; if it turns out that $\langle r_T \rangle < r_{cr}$, then the drops formed due to turbulence will break down as time goes on. The adjustment of the vortex flow structure is accompanied by breakdown of the large-scale vortices in the region coincident with the region of heterogeneous—homogeneous volume condensation. The transverse vortices of dimension Ω_s are transformed into longitudinal vortices in the condenser. It was established experimentally that: $\Omega_s \approx 6.5\Omega_c$. The laminar influence of suction begins from the heat transfer surface of the condenser and propagates to the flow core as the boundary layer grows. Here one should consider that the degree of turbulence accomplished in the flow core in the evaporator remains constant until the boundary layers join in the condenser. Therefore we can estimate the influence of turbulence on the process of generation of condensed phase in the condenser from the characteristics of turbulence generated in the evaporator, i.e., $\Omega_s = 6.5\Omega_c \approx 6.5$ mm, $\eta \sim 10^{-2}$-10^{-3}:

$$\langle r_T \rangle = 1{,}1 \cdot 10^{-6}\Omega/\eta. \tag{13}$$

For these values $\langle r_T \rangle \approx r_{cr}$, which indicates that condensate may form due to turbulence. Thus, this mechanism has an independent value in the general picture of generating conditions to stimulate the development of volume condensation in the vapor flow of a heat pipe.

A positive pressure gradient in a heat pipe condenser leads to the formation of reverse flow, physically identical in flow structure with that of the near wake behind a poorly streamlined body. The formation of the condensed phase in the near wake for comparatively small degrees of vapor supercooling outside the wake of $\Delta T \simeq 5°C$, which has excluded any appreciable spontaneous formation of condensate, was established experimentally in [15]. The convergence of these structures means that the necessary degree of metastability can be reached for active formation of condensate in the interior of a reverse flow vortex structure. A detailed analysis of possible formation of condensate in the vortex of a near wake was described in [15]:

$$P_0(\tau) = P_\infty - \frac{\rho\Gamma}{8\pi^2\nu(\tau_0 + \tau)}, \tag{14}$$

i.e., the pressure on the vortex axis is less than outside it. Therefore, the temperature also in the vortex will decrease towards the axis. In addition:

$$\left(\frac{d\ln P}{dT}\right)_s > \left(\frac{d\ln P}{dT}\right)_\rho. \tag{15}$$

Therefore, if there is some supercooling outside a vortex, this will only increase within it. Our experiments confirm that condensed phase can form in reverse-vortex flow.

The experiment showed that the region of developed heterogenous—homogeneous volume condensation falls between the sections x = 7.6, the end of the evaporator, and x = 10, in the condenser. The growth of the number of particles and the rate of growth of the mean radius are stopped, but, beginning at x = 16, the mean radius of drops begins to decrease (Fig. 3, curve 1). Therefore, the vapor flux leaves the metastable state in the region of section x = 10, and from then on one can assume that J = 0. The subsequent growth of drops from x = 10 to x = 16 is due to the process of coagulation, for which we can write the following expression for the variation of the mean drop size, according to [16]:

$$\frac{r}{r_{10}} = (1 + \beta_1\tau)^2;\ \beta_1 = 8\varepsilon_{10}\sqrt{\frac{\Gamma}{4\pi^2\rho'_{10}\Omega r_{10}}}, \tag{16}$$

where r_{10}, ε_{10}, ρ_{10}' correspond to the vapor state at section x = 10, for which J = 0.

As a rule, we have $\beta\tau_1 \gg 1$, and therefore the final drop size is practically independent of r_{10}. It is known [16] that if, as a result of coagulation, the mean drop size reached in turbulent flow exceeds the critical size R_{cr}, then these drops will break up, with a large probability. The expression for R_{cr}, obtained by processing the experimental data, takes the form:

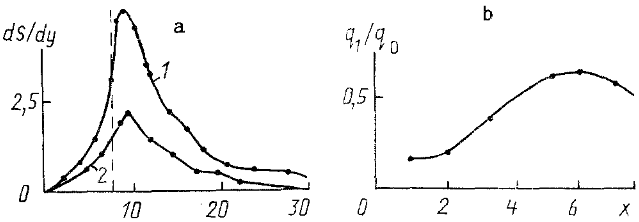

Fig. 5. Simultaneous influence of hydrodynamics and volume condensation in the vapor flow on heat and mass transfer in the heat pipe (conditions as for Fig. 1): a) variation of the gradients of optical nonuniformities along the heat pipe filter; 1) y = 0; 2) y = 1; b) ratio of the specific heat fluxes in the evaporator q_1/q_0; q_1 for y = 1, q_0 for y = 0.

$$R_{cr} = 0{,}09\delta \left[\frac{1}{U} \sqrt{\frac{2\sigma}{\rho'\delta}} \right]^{6/7} (\rho''/\rho')^{1/7}. \tag{17}$$

From Eqs. (16) and (17) we find the distance from x = 10 at which the drops grow to the critical size:

$$L = \frac{0{,}016\mu_v^{3/14}\,(2\sigma)^{9/14}\,\delta^{31/28}}{\varepsilon_{10}\Gamma^{1/2}\,(\rho')^{6/7}U^{29/28}(\rho'')^{1/28}} > L_c. \tag{18}$$

Our experimental conditions were: acetone, T = 323 K; $U = 5\cdot10^{-2}$ m/sec; $\Gamma = 10^{-19}$ J; $\delta = 19\cdot10^{-3}$ m; $\varepsilon_{10} \approx 0.07$. From these data we obtain $L > L_k$, where L_k is the condenser length. Therefore, drop breakdown did not occur in our experiment.

Besides breakdown, the decrease of mean drop radius, starting at x = 16, can be explained by partial evaporation of drops superheated as a result of condensate growth of drops moving to the cold part of the condenser, and also to their falling out of the flow. We consider the behavior of a drop in the field of the aerodynamic and gravitational forces [17]:

$$R'_{cr} = (3U^2\rho\xi)/[8g(\rho' - \rho_s)], \tag{19}$$

where $\xi = 0.4$ for $10^3 < Re_0 < 2\cdot10^5$. Drops for which $r_i > R_{cr}'$ will fall out of the flow.

Figure 5 shows the development of heat and mass transfer according to the set of phenomena described above: hydrodynamics and heterogeneous—homogeneous volume condensation in the vapor flow of a heat pipe.

Conclusions. I. Processes of heterogeneous—homogeneous volume condensation were achieved in the vapor flow of a heat pipe:

1) accumulation of condensation nuclei due to ejection of drops from the evaporator wick structure;

2) supersaturation due to processes of blowing in the evaporator;

3) supercooling due to efflux of vapor in the condenser;

4) heat removal with rearrangement of the vortex structure in the condenser.

II. The process of volume condensation, growth and coagulation of drops occurring in the field of aerodynamic and gravitational forces lead to a drain of condensed phase, unsymmetrical relative to the heat pipe axis, on the surface of the condenser.

Notation. J) rate of nuclei formation, $1/(m^3\cdot sec)$; μ) molecular mass, kg/mole; λ) heat of vapor formation, J/kg; l_λ) wavelength of the light, m; r) drop radius, m; Ω) scale of turbulence, size of vortex, m; ρ', ρ") density of the liquid and the vapor, kg/m^3; δ) channel height, m; Γ) Gamaker constant, j; β_c) coefficient of accommodation, correction to the operation of formation of the new phase; v) viscosity, m^3/sec; ξ) coefficient of friction; M) Mach number. Subscripts: v) vapor, ℓ) liquid; cr) critical; t) turbulent; e) evaporator; c) condenser.

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