NOTATION

k, σ, absorption and scattering coefficients of an elementary layer of material, m^{-1}; $\Lambda = \sigma/(k + \sigma)$, photon survival probability, scattering criterion; $\bar{k}$, s, ε_e, averaged coefficients of absorption, backscattering, and extinction of an elementary layer, m^{-1}; $\Lambda_e = s/(\bar{k} + s)$, mean effective photon survival probability, Schuster criterion; τ, optical depth; $\varkappa$, γ, constant coefficients; β, ν, real or imaginary numbers; E, radiation flux density, W/m^2; E_0, spatial irradiance, W/m^2; q, resultant flux density, W/m^2; R, T, reflectivity and transmittance of a layer of finite thickness l; R_∞, reflectivity of an optically infinitely thick layer; λ, wavelength, μm. Indices: λ, spectral; i, incident; e, effective.

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HEAT EXCHANGE AND FRICTION IN A SUBSONIC VAPOR FLUX OF HIGH-TEMPERATURE HEAT PIPES

V. N. Fedorov and V. Ya. Sasin

UDC 621.1.016.4-462

The influence of forced vapor convection on heat transport in heat pipes is examined on the basis of the solution of the energy and motion equations. It is shown that radial heat flux due to molecular heat conduction of the vapor in the evaporator is negligible.

High-temperature heat pipes are ordinarily characterized in the literature as isothermal apparatuses. However, depending on the heat-exchange conditions in the surrounding medium and the magnitude of the power being transmitted, modes can exist where the axial temperature profile is characterized by abrupt changes from the maximum value at the beginning of the evaporator to the temperature of the surrounding medium at the end of the condenser. C. A. Busse gave a demarcation of heat pipe operating modes and typical axial temperature profiles. Taken as the viscous flow mode is that for which the vapor pressure at the end of the condensation zone is approximately equal to the vapor pressure at the temperature of the

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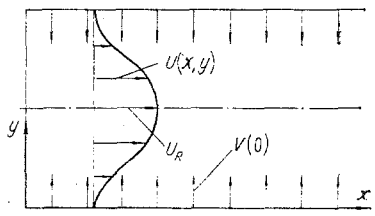

Fig. 1. Coordinate system and decomposition of the velocity according to direction.

ambient medium. An abrupt temperature drop in the condenser without recovery corresponds to the viscous mode. Viscosity forces are negligible in the inertial mode of vapor flow, where the temperature is partially or completely recovered in the condenser. An attempt is made in this paper to solve the equations taking inertial and viscous forces into account.

To a considerable extent, the power transmitted by heat pipes depends on the nature of the vapor generation processes, the liquid and vapor phase motion, and the heat exchange. The heat flux in evaporators of heat pipes is determined mainly by the mass flow rate of the vapor from the evaporation surface. However, heat exchange by heat conduction of the evaporation surface exists with a vapor flux whose velocity is tens and hundreds of meters per second. The purpose of this paper is to estimate the contribution of radial heat conduction to the total heat flux.

The stationary operating mode of a plane heat pipe is considered. The normal vapor velocity from the evaporation surface for a constant density q on the surface is considered constant and given. It is assumed that there are no temperature jumps on the evaporation surface. By neglecting the diameter of the vapor flux as compared with the length, the pressure change along the normal to the evaporation surface can be neglected. For the conditions presented above, the mean vapor velocity across the section is a linear function of the axial coordinate:

$$\bar{U} = \frac{1}{\rho S}\int_S U\rho dS = kx. \tag{1}$$

There are no viscosity and energy exchanges on the axis for a mass, momentum, and energy flow profile symmetric relative to the axis. The axial velocity on the axis U_R is connected with the pressure P by the Bernoulli relationship which is characteristic for an inviscid incompressible fluid flow. The equation of total enthalpy conservation

$$\int \frac{dP}{\rho} + \frac{U_R^2}{2} = \text{const}, \tag{2a}$$

$$t(0) = t(R) + \frac{U_R^2}{2C_p} \tag{2b}$$

is valid in the absence of sinks on the axis. Letting C' denote the ratio between the velocity U_R on the axis and the mean velocity U across the section, and assuming $(C')^2 = C$, we obtain a dependence for the pressure by account:

$$\frac{dP}{dx} = -C\rho\bar{U}\frac{d\bar{U}}{dx}. \tag{3}$$

The mean axial velocity $\bar{U}$ is determined by means of (1), where the proportionality factor k is expressed in terms of the heat flux by starting from the mass balance for the section at the end of the evaporator:

$$k = \frac{Q}{\rho S r x_e}. \tag{4}$$

The vapor density and heat of phase transition are taken for a given temperature of the evaporation surface.

To determine the velocity and temperature profiles it is necessary to solve a system of equations in which are contained the equations of motion, continuity of the mass, and energy:

$$U\frac{\partial U}{\partial x} + V\frac{\partial U}{\partial y} - \nu\frac{\partial^2 U}{\partial y^2} + \frac{1}{\rho}\frac{dP}{dx} = 0, \tag{5a}$$

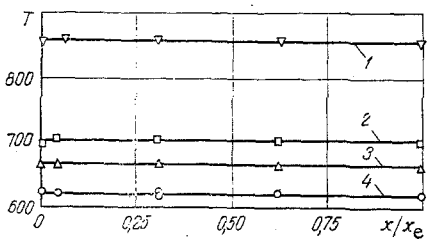

Fig. 2. Typical dependences of the wall temperature of heat pipes on the dimensionless length for an evaporator with a variable axial heat flux; q_x in 10^4 W/m^2; T, °C: 1) $q_x = 1310$; 2) 655; 3) 502; 4) 304.

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0, \tag{5b}$$

$$U\frac{\partial t}{\partial x} + V\frac{\partial t}{\partial y} - a\frac{\partial^2 t}{\partial y^2} = 0. \tag{5c}$$

This system of equations describes the vapor motion in the evaporator of the heat pipe as the laminar flow of a plane-parallel stream. The solution can be applied as an approximation to analyze the heat exchange and friction in heat pipes, since we speak about a thin layer near the evaporation surface. Indeed, the derivative of the temperature with respect to the normal coordinate is sought in computing the heat exchange, and the derivative of the axial velocity with respect to the same coordinate is sought in computing the friction drag. The derivatives are hence taken for the evaporation surface. Therefore, to seek the quantities presented above it is sufficient to have the velocity and temperature distribution near the surface [1].

To extract the particular solution, let us write the boundary conditions. Relying on Fig. 1, we have $t(y) = t(0) = \text{const}$, $U = 0$, $V(y) = V(0) = \text{const}$ for the surface coordinate at $y = 0$. The fundamental boundary-layer approximation — the axial velocity on the surface equals zero — is used here; the blowing law is given. From symmetry conditions, for the axis at $y = R$

$$\frac{\partial t}{\partial y} = 0; \quad \frac{\partial U}{\partial y} = 0, \quad V = 0. \tag{6}$$

The system (5) should be supplemented by relationships for the axis (2) and by the mean velocity distribution. The difference between the present and the classical blowing problems for a homogeneous gas in an independent potential flow is in the assignment of the different boundary conditions. In the present problem it is impossible to know the inviscid flow temperature and velocity in advance, i.e., the flows on the axes; these quantities are to be determined.

It is naturally impossible to make a check in the solution to determine the friction in the absence of blowing, since without blowing there are neither tangential stresses nor friction in the evaporator. New boundary conditions are necessary for such checks, as is a new solution, for instance, for the transport zone of heat pipes which is known well enough [1, 4].

To solve the system (5), let us assume self-similarity of the solution and let us reduce the equations to a more convenient form for seeking the roots. Let us introduce the new dimensionless variables

$$\eta \equiv \frac{y}{\sqrt{vx/\bar{U}}}, \quad \zeta'(\eta) \equiv \frac{U}{\bar{U}}, \quad \theta \equiv \frac{t(\eta) - t(0)}{t_m - t(0)},$$

where t_m is the mean mass temperature across a section for a given coordinate x. The prime denotes differentiation with respect to the variable η. Let us replace the axial and radial velocities in terms of the stream function ψ so that it would satisfy the continuity equation (5b):

$$U = \frac{\partial \psi(x, y)}{\partial y}, \quad V = -\frac{\partial \psi(x, y)}{\partial x}, \quad \zeta = \frac{\psi(x, y)}{\sqrt{vx/\bar{U}}}.$$

Using the stream function and the dimensionless variables, we obtain a new system of ordinary equations (7) in place of the system of partial differential equations (5):

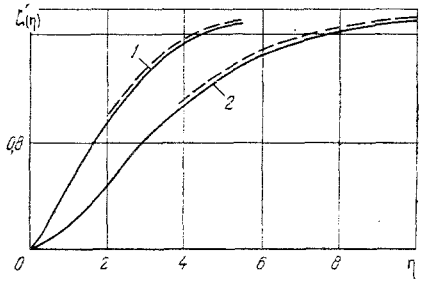

Fig. 3. Velocity profile: 1) $\eta_R = [V(0)\mathrm{Re}_x^{1/2}/\bar{U}] = 5.5$; 2) $\eta_R = 10.0$.

$$\zeta''' + \zeta''\zeta - (\zeta')^2 + C = 0, \tag{7a}$$

$$\theta'' + \mathrm{Pr}\,\zeta\theta' = 0. \tag{7b}$$

Equation (7a) is the transformed equation of motion and is the dependence of the velocity on the dimensionless coordinate, while (7b) is the energy equation. The Prandtl parameter Pr for the vapor is given by means of the governing temperature. In a first approximation, the given temperature of the evaporation surface can be taken as governing. The undetermined coefficient C is calculated as the square of the dimensionless velocity on the axis:

$$C = [\zeta'(\eta_R)]^2.$$

After manipulation, the boundary conditions (6) for the system (7) become

$$\eta = 0, \quad \theta = 0, \quad \zeta(0) = -\frac{V(0)\,\mathrm{Re}_x^{1/2}}{\bar{U}} = \mathrm{const},$$

$$\eta = \eta_R, \quad \zeta''(\eta_R) = 0, \quad \zeta(\eta_R) = 0, \quad \theta'(\eta_R) = 0. \tag{8}$$

These conditions and the system of equations itself with the dependence (2) taken into account determine the unknowns uniquely. The system has a solution. Equation (7b) is integrated directly by separation of variables, and as the solution we obtain the dependence

$$\theta(\eta) = \theta(\eta_R)\frac{\int_0^{\eta}\exp\left(-\mathrm{Pr}\int_0^{\eta}\zeta d\eta\right)d\eta}{\int_0^{\eta_R}\exp\left(-\mathrm{Pr}\int_0^{\eta}\zeta d\eta\right)d\eta} \equiv \theta(\eta_R)\frac{I(\eta)}{I(\eta_R)}. \tag{9}$$

The temperature dependence takes its final form after transforming the dimensionless temperatures into dimensional temperatures and expressing the velocity on the axis in terms of the mean velocity:

$$t(\eta) = t(0) - \frac{[\zeta'(\eta_R)\,kx]^2}{2C_p}\frac{I(\eta)}{I(\eta_R)}. \tag{10}$$

To seek the temperature profile in the coordinate η, the coordinate dependence of the dimensionless integral velocity is needed, which can be obtained from the motion equation (7a). An approximate analytic solution of (7a) was executed by giving basis functions with undetermined coefficients. The solution is written as

$$\zeta(\eta) = \eta^5\left(-\frac{F}{\eta_R^4} + \frac{C_0}{\eta_R^4} + \frac{FC_2}{\eta_R^3}\right) + \eta^4\left(\frac{4F}{\eta_R^3} - \frac{C_0}{\eta_R^3} + \frac{FC_1}{\eta_R^2} - \frac{3FC_2}{\eta_R^2}\right) +$$

$$+ \eta^3\left(-\frac{6F}{\eta_R^2} - \frac{2FC_1}{\eta_R} + \frac{3FC_2}{\eta_R}\right) + \eta^2\left(\frac{4F}{\eta_R} + FC_1 - FC_2\right) - F\eta_R. \tag{11}$$

Here F is the dimensionless "blowing parameter" referred to the dimensionless axial coordinate:

$$F = \frac{V(0)\,\mathrm{Re}_x^{\frac{1}{2}}}{\bar{U}\,\eta_R}. \tag{12}$$

Transformation of the function results in its explicit dependence on the blowing parameter.

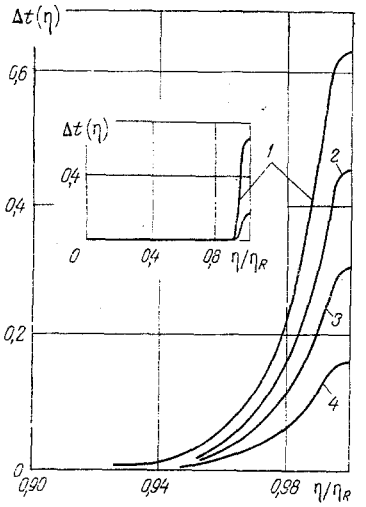

Fig. 4. Axial temperature profiles in different evaporator sections: 1) $x/x_e = 1.0$; 2) 0.85; 3) 0.7; 4) 0.5; Δt, °K.

The coefficient is $|C_0| < 3$ for the case under consideration of computing high-temperature heat pipes with sodium as the heat carrier. The coefficient was refined on an electronic computer (the "Nairi 2"). The accuracy of the solution hence diminished far from the wall. The instability zone is shown by the dash in Fig. 3.

The dependence between the undetermined coefficients is

$$C_1 = -\frac{4C_0\bar{U}}{V(0)\,\mathrm{Re}_x^{1/2}}, \quad C_2 = \frac{2C_0\bar{U}}{V(0)\,\mathrm{Re}_x^{1/2}}.$$

Having obtained the function $\zeta(\eta)$ we can determine the temperature profile in the evaporator of heat pipes by a numerical solution of (10).

Experimental investigations verify the existence of a relative isothermy of the evaporators of high-temperature heat pipes for an optimum operating mode. Typical temperature profiles over the length of the evaporator for variable axial loads are represented in Fig. 2. The surface in the condenser of such pipes is hence also isothermal or almost isothermal. It is shown in Fig. 3 that the velocity profile is deformed with the change in load and the "blowing parameter," correspondingly.

It should be noted that the derivative of the velocity with respect to the coordinate becomes less with the increase in the "blowing parameter," the friction drag on the vapor — fluid interfacial surface therefore being diminished. This results in a drop in the pressure along the evaporator length in heat pipes and to relative constancy of the vapor flux temperature.

To estimate the heat exchange of an evaporation surface with moving flux, the Fourier law must be used in the form

$$q_T = -\frac{\lambda}{\sqrt{\nu/k}}(\mathrm{grad}\, t)_0. \tag{13}$$

The derivative $t'(\eta)$ is determined from (10) at zero. It is shown graphically in Fig. 4 that the heat fraction transmitted in addition by heat conduction from the wall to the vapor increases with the growth in the axial coordinate. Computations performed for a high-temperature heat pipe at t = 575°C show that the heat fraction due to heat exchange of a wall with a moving flux is hundredths of a percent of the phase-transition heat.

NOTATION

U, V, axial and normal velocities, respectively; S, surface; Q, heat flux; R, vapor flux radius; ϑ, excess temperature; P, pressure; t, T, temperatures; F, parameter; C, coefficient; μ, coefficient of dynamic viscosity; ρ, density; λ, thermal-conductivity coefficient; ν, coefficient of kinematic viscosity; q, heat flux density; η, dimensionless coordinate; ζ, velocity function; Re, Reynolds criterion,

$Re_x = (\bar{U}x)/\nu$; Pr, Prandtl criterion; x, y, axial and normal coordinates; r, heat of vapor formation; C_p, specific heat; ψ, stream function. Indices: R, axis; e, evaporator; 0, evaporation surface; t, heat conduction.

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ACTIVATION OF VAPORIZATION CENTERS. I*

V. S. Novikov

UDC 536.243

The influence of dissolved gases on the formation of a vapor-bubble nucleus of critical dimensions is considered. A mathematical model is proposed for the subsequent evolution of the nucleus; it gives a correct description of the growth of the vapor bubble up to the instant at which it is detached from the wall.

Even after allowing for the temperature dependence of the surface tension σ and molecular heat of vaporization λ, the theory of heterophase fluctuations [1] leads [2] to values of the degree of superheating required to vaporize liquids roughly twice as great as those measured experimentally. Harvey [3] noted that microscopic gas nuclei might survive in the indentations of rough, unwetted solid surfaces, causing liquids to boil at very slight superheatings. The idea of micronuclei constitutes the basis for the theory of the deactivation of indentations proposed by Holz and Singer and set out in [4]. It follows from [5, 6] that stable gas micronuclei are absent in the case of organic liquids. For these, as well as for liquid metals which wet adjacent solid surfaces almost completely [4], the deactivation theory cannot explain the fact of early boiling. A new physical model was proposed in [2, 7] for the initial stage of phase transformations in liquids; this model may help in explaining the boiling of organic liquids and molten metals. According to [2, 7], complexes of several vapor molecules formed as a result of the superheating of the boundary layer of liquid are adsorbed in indentations on the surface, forming nuclei of greater than critical size. This paper is a continuation of [2, 7] and considers the influence of dissolved gases on the boiling of liquids, as well as formulating a mathematical model for the further evolution of the nucleus.

One of the reasons for the formation of gas micronuclei on a solid surface is the adsorption of gas dissolved in the liquid on surface indentations. Let us consider an indentation of conical shape with a depth z_0 and a base radius r_0. The number N_a of gas molecules adsorbed in the indentation is determined by the adsorption isotherm derived in [7, 8], which has the following form for a conical indentation:

$$N_a = \frac{\gamma g n_s \pi r_0 P \sqrt{r_0^2 + z_0^2}\left[1 - \left(\frac{P}{P_0}\right)^{\beta}\right] \exp\left(\frac{\Psi}{kT}\right)}{\left(1 - \frac{P}{P_0}\right)\left\{1 - \frac{P}{P_0} + \gamma g P\left[1 - \left(\frac{P}{P_0}\right)^{\beta}\right] \exp\left(\frac{\Psi}{kT}\right)\right\}}. \quad (1)$$

*This paper (together with its second part) was presented to the Fifth All-Union Conference on Heat Transfer and Hydraulic Resistance regarding the motion of two-phase flows in various parts of hydraulic machines and installations; the Conference was held in Leningrad on October 15-18, 1974.

Institute of Technical Heat Physics, Academy of Sciences of the Ukrainian SSR, Kiev.

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