STUDY OF THE PROCESS OF HEAT AND MASS TRANSFER IN THE REGION OF THE VAPOR—GAS FRONT IN A GAS-REGULABLE HEAT PIPE

V. V. Galaktionov and L. P. Trukhanova

UDC 536.58

Gas-regulable heat pipes (GRHP) are widely used to keep the operating temperature of heat-releasing devices within specific limits under variable thermal loads and changing cooling conditions.

A substantial number of studies have now been published concerning heat and mass transfer in GRHP's. In [1], the authors concluded that diffusion has a negligibly small effect on temperature distribution along the pipe compared to the effect of the thermal conductivity of the body. However, this conclusion pertains to concrete heat pipes. It was assumed in [2] that, depending on the geometric and physical characteristics, the amount of heat transferred as a result of the latent heat of condensation of the diffusing vapor may be significantly greater than the amount transferred as a result of heat conduction by the wick and body of the pipe. Unfortunately, the problem was solved with constant physical properties for the vapor and gas, which does not reflect actual conditions. Also, there was no comparison between calculated data and experimental data, making it difficult to judge the reliability of the results obtained.

In [3] a mathematical model for calculating the temperature distribution along a GRHP was constructed in a unidimensional approximation, and again there was no comparison between theoretical and empirical data. The studies [4-6] gave a mathematical formulation for the study of heat and mass transfer at the vapor—gas front (VGF) in a GRHP, but the results of the solution were not presented.

In the general case, heat and mass transfer in a GRHP are determined by the total effect of the convective and diffusive components, as well as by heat transfer by axial conduction along the wall of the pipe and wick.

Presented below is a mathematical formulation and results of the solution of a two-dimensional problem on heat and mass transfer in the region of the VGF in a GRHP with allowance for convection and diffusion under loads considerably smaller than the sonic limit of heat pipe operation. The results obtained are compared with empirical data.

We will examine the steady laminar motion of a vapor—gas mixture in a planar heat pipe with allowance for gravity and the following assumptions: 1) the vapor and gas and their mixture obey the ideal gas laws; 2) evaporation and condensation occur only at the phase boundary, where the vapor is under an equilibrium partial pressure corresponding to its temperature; 3) the energy equation does not contain terms accounting for dissipation of mechanical energy. It is also assumed that the Lewis number Le = 1; 4) the coefficient of heat transfer from the outside surface of the condenser to the environment is constant.

With allowance for the above assumptions, we can write the system of differential equations — including the equation linking the stream function and vorticity and the momentum, energy, and diffusion equation — as follows:

$$\frac{\partial}{\partial x}\left(\frac{1}{\rho}\frac{\partial\psi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{1}{\rho}\frac{\partial\psi}{\partial y}\right)+\omega=0, \tag{1}$$

$$\frac{\partial}{\partial x}\left(\omega\frac{\partial\psi}{\partial y}\right)-\frac{\partial}{\partial y}\left(\omega\frac{\partial\psi}{\partial x}\right)-\frac{\partial^2(\mu\omega)}{\partial x^2}-\frac{\partial^2(\mu\omega)}{\partial y^2}-$$

$$-\frac{\partial}{\partial x}\left(\frac{u^2+v^2}{2}\right)\frac{\partial\rho}{\partial y}+\frac{\partial}{\partial y}\left(\frac{u^2+v^2}{2}\right)\frac{\partial\rho}{\partial x}+g\cos\beta\frac{\partial\rho}{\partial x}-g\sin\beta\frac{\partial\rho}{\partial y}=0, \tag{2}$$

Moscow Energy Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 48, No. 3, pp. 409-414, March, 1985. Original article submitted February 6, 1984.

 0022-0841/85/4803-0296$09.50

$$\frac{\partial}{\partial x}\left(h\frac{\partial\psi}{\partial y}\right)-\frac{\partial}{\partial y}\left(h\frac{\partial\psi}{\partial x}\right)-\frac{\partial}{\partial x}\left(\Gamma_h\frac{\partial h}{\partial x}\right)-\frac{\partial}{\partial y}\left(\Gamma_h\frac{\partial h}{\partial y}\right)=0, \quad (3)$$

$$\frac{\partial}{\partial x}\left(m_v\frac{\partial\psi}{\partial y}\right)-\frac{\partial}{\partial y}\left(m_v\frac{\partial\psi}{\partial x}\right)-\frac{\partial}{\partial x}\left(\Gamma_m\frac{\partial m_v}{\partial x}\right)-\frac{\partial}{\partial y}\left(\Gamma_m\frac{\partial m_v}{\partial y}\right)=0. \quad (4)$$

Equations (1)-(4) are written in terms of the stream function ψ and vorticity ω, which are determined by the relations

$$\frac{\partial\psi}{\partial y}=\rho u,\quad \frac{\partial\psi}{\partial x}=-\rho v,\ \omega=\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}. \quad (5)$$

The equations Γ_h and Γ_m are the energy and mass transfer coefficients, respectively:

$$\Gamma_h=\lambda/c_p,\ \Gamma_m=\rho\mathscr{D}. \quad (6)$$

The diffusion coefficient $\mathscr{D}$ for the mixture of water vapor and air being examined is calculated on the basis of molecular theory and is approximated by the relation $\mathscr{D}=\mathscr{D}(T)$.

The density of the mixture is determined from the equation of state

$$\rho=\left(\frac{P}{RT}\right)\Big/\left(\frac{m_v}{M_v}+\frac{m_g}{M_g}\right). \quad (7)$$

The specific heat of the mixture

$$c_p=c_{pv}m_v+c_{pg}m_g. \quad (8)$$

The enthalpy of the mixture

$$h=c_pT. \quad (9)$$

To completely describe the process mathematically, it is necessary to assign boundary conditions. They will be as follows for the velocity components u and v:

$$v(0,\ y)=v(L,\ y)=0,$$
$$u(0,\ y)=u(L,\ y)=u(x,\ 0)=u(x,\ h_v). \quad (10)$$

The value of the normal velocity component at the vapor—liquid boundary for the upper and lower walls of the channel is derived from the energy balance equation. For the evaporator

$$v_e^t=\frac{-q+\lambda\frac{\partial T}{\partial y}}{m_v\rho h_{fg}},\ v_e^b=\frac{q+\lambda\frac{\partial T}{\partial y}}{m_v\rho h_{fg}}, \quad (11)$$

while for the condenser

$$v_e^t=\frac{k(T-T_{e.n})+\lambda\frac{\partial T}{\partial y}}{m_v\rho h_{fg}},\ v_e^b=\frac{-k(T-T_{e.n})+\lambda\frac{\partial T}{\partial y}}{m_v\rho h_{fg}}. \quad (12)$$

Governing relations (5) are integrated to obtain boundary conditions for the stream function. As a result, we have:

TABLE 1. Characteristics of the GRHP

Vapor–gas mixture	L, m	L_e, m	L_a, m	L_c, m	h_v, m	δ_w, m	δ_{wa} m	P, Pa	$T_{e.n}$, K
Water–air	0,424	0,187	0,029	0,208	0,015	0,00053	0,001	101325	288,15

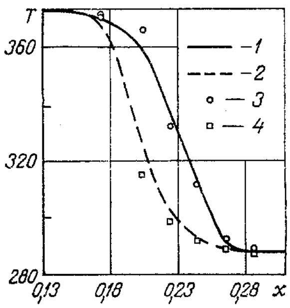

Fig. 1. Distribution of the temperature of the vapor–gas mixture along a GRHP for Q = 130 W: 1 and 3) calculated and experimental data, respectively, near the surface of the upper wall; 2 and 4) the same near the surface of the lower wall. T, °K; x, m.

$$\psi(0,\ y) = \psi(L,\ y) = 0, \tag{13}$$

$$\psi(x,\ 0) = \psi(x,\ h_v) = -\int_0^x \rho m_v v dx.$$

The temperature at the phase boundary is calculated on the basis of the assumption of a parabolic temperature distribution at the boundary. For the other boundaries we take

$$\left.\frac{\partial T}{\partial x}\right|_{x=0,L} = 0. \tag{14}$$

The mass concentration of the vapor at the phase boundary is determined from the relation $m_v = m_v(T, P)$ for the given working fluid. Also, as for temperature, the following condition is satisfied

$$\left.\frac{\partial m_v}{\partial x}\right|_{x=0,L} = 0. \tag{15}$$

The condition for vorticity at the vapor–liquid boundary is derived from the assumption of constancy of ω in the immediate vicinity of the boundary [7]:

$$\omega(x,\ 0) = \omega(x,\ h_v) = -2\,\frac{\psi_{0+1} - \psi_0}{n_{0+1}^2 \rho}, \tag{16}$$

where n_{0+1} is the distance from the phase boundary to the first node of the grid in the radial direction and ψ_{0+1} is the value of the stream function at this node.

The resulting system of equations was solved numerically by the Gauss–Seidel method on an ES-1033 computer.

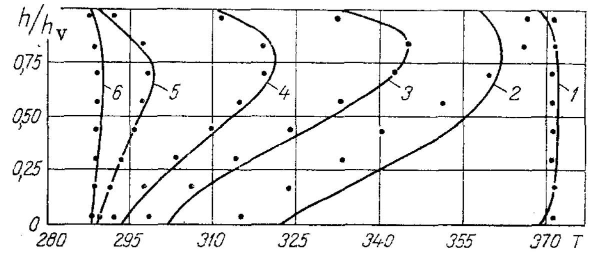

Fig. 2. Temperature distribution over the height of the GRHP for Q = 130 W: the solid lines denote calculated results, while the points denote empirical results: 1) x = 0.174 m; 2) 0.204; 3) 0.224; 4) 0.244; 5) 264; 6) 0.284.

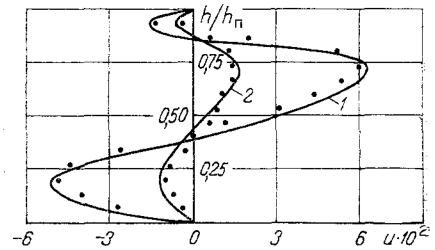

Fig. 3. Change in the axial component of velocity over the height of the GRHP for Q = 240 W: solid lines denote calculated results, while the points denote empirical results: 1) x = 0.291 m; 2) 0.321. $u/10^2$, m/sec.

To check the theoretical data obtained, we designed a planar gas-regulable heat pipe with transparent windows to permit observation of the hydrodynamic of the processes taking place. The parameters of the GRHP that was studied are shown in Table 1. The experimental unit was similar to that described in [5]. All of the tests were performed with the unit in the horizontal position. The condenser was cooled by running water. To study the velocity fields we used a laser Doppler anemometer made by the DISA company. Temperature along the pipe walls was measured by 30 Chromel—Alumel thermocouples welded to the outside surface of the upper and lower walls. The temperature of the vapor—gas mixture was measured by a movable thermocouple probe.

The graphs (Fig. 1) clearly show the presence of a fairly long vapor—gas transition zone. The difference in the temperature profiles for the upper and lower walls is evidence of the substantial effect of gravity on heat and mass transfer in the GRHP. It can be seen from Fig. 2 that the changes in temperature in the axial and radial directions are comparable, which clearly demonstrates the two-dimensional character of the problem. With an increase in the thermal load the qualitative character of the temperature change in the vapor—gas region remains the same, but convection has more of an effect on the formation of the vapor—gas front.

The velocity profiles obtained by theoretical and experimental means for the vapor—gas transition region (Fig. 3) confirm the presence of the reverse flows observed visually on the unit and seen earlier by other authors [5]. The reverse flows develop in the region of the vapor—gas front and are quite nonsymmetrical. Figure 3 shows the distribution of the axial component of velocity over the height of the GRHP for two different sections with a thermal load Q = 240 W.

Comparison of the theoretical and experimental data shows that they agree well, which indicates that the assumptions used in the theoretical model are correct.

NOTATION

c_p, specific heat; $\mathcal{D}$, diffusion coefficient of the bindary vapor—gas mixture; Γ_h, energy transfer coefficient; Γ_m, mass transfer coefficient; g, acceleration due to gravity; h, enthalpy of the mixture; h_v, height of the vapor space; h_{fg}, latent heat of vaporization; k, heat transfer coefficient; L, length of the heat pipe; M, molecular weight; m, concentration; P, pressure; Q, total heat flux; q, heat flux; R, universal gas constant; x, y, coordinates; T, temperature; $T_{e.n}$, ambient temperature; u, axial component of velocity; v, radial component of velocity; β, angle of inclination of the pipe; λ, thermal conductivity of the mixture; μ, absolute viscosity; ρ, density of the mixture; ψ, stream function; ω, vorticity. Indices: a, adiabatic zone; t, top; g, gas; v, vapor; e, evaporator; c, condenser; b, bottom; 0, phase boundary; wa, wall; w, wick.

LITERATURE CITED

1. Edwards and Markus, "Heat and mass transfer in the region of the vapor–gas front in a gas-filled heat pipe," Trans. ASME Ser. C. Heat Transfer, 94, No. 2, 31-40 (1972).
2. Rohani and Tuen, "Steady heat and mass transfer in the vapor–gas region of a heat pipe filled with a noncondensing gas (two-dimensional approximation)," Trans. ASME Ser. C. Heat Transfer, 95, No. 3, 92-98 (1973).
3. I. G. Shekriladze, Z. V. Arkaniya, G. I. Zhorzholiani, and I. I. Topuriya, "Analysis of thermal operating regimes of a gas-regulable heat pipe," Teplofiz. Vys. Temp., 14, No. 5, 1126-1129 (1976).
4. V. V. Galaktionov and O. M. Shalya, "Two-dimensional problem of heat and mass transfer in the region of the vapor–gas front in a gas-regulable heat pipe," in: Tr. Mosk. Ordena Lenina Energet. Inst. Teplo- i Massoobmennye Protsessy i Apparaty, 268, 121-128 (1975).
5. V. V. Galaktionov, A. A. Parfent'eva, V. D. Portnov, and V. Ya. Sasin, "Study of the boundary of the vapor–gas front in the condenser of a planar gas-regulable heat pipe," Inzh.-Fiz. Zh., 42, No. 3, 387-392 (1982).
6. S. Chi, Heat Pipes: Theory and Practice [Russian translation], Mashinostroenie, Moscow (1981).
7. A. D. Gosman, W. M. Pun, A. K. Runchal, D. B. Spalding, and M. Wolfshtein, Heat and Mass Transfer in Recirculating Flows, Academic Press, New York (1969).