NOTATION

ℓ, R, length and radius of membrane; r, characteristic pore size; r_0, radius of straight channel; N, r_s, number of channels of model set of capillaries per unit area of the porous medium and their radius; λ, mean free path length; $\bar{u}_{iz}^{ch}$, $\bar{u}_{iz}^{p}$, $\bar{u}_{iz}^{M}$, projection of mean velocity of motion of molecules of the i-th component in the channel, porous medium, and membrane, respectively, onto the membrane axis; p, T, n, pressure, temperature, and number density of mixture particles; m_i, d_i, mass and diameter of molecules of the i-th component; c_i, concentration of i-th component of mixture; η_{12}, η_i, viscosity of mixture and its i-th component, respectively; D_{12}, σ, mutual diffusion coefficient and diffusional-slip coefficient; k, Boltzmann constant; S^{ch}, S^p, S^M, cross-sectional area of channel, porous medium, and membrane; Q^{ch}, Q^p, Q^M, volume flow rate of gas mixture through channel, porous medium, and membrane; Q^e, experimental volume flow rate of gas mixture; Kn^p, Kn^{ch}, Knudsen number in pores and in channel; δ_{12}^p, δ_{12}^{ch}, inverse Knudsen number in pores and in channel; Π, porosity; Δp_m, t_m, maximum magnitude of baric effect and time for its attainment; V, chamber volume.

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HEAT TRANSFER IN VAPOR CONDENSATION ON LAMINAR AND TURBULENT LIQUID JETS, TAKING ACCOUNT OF THE INLET SECTION AND VARIABILITY OF THE FLOW RATE OVER THE JET CROSS SECTION

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The results of numerical modeling of heat transfer in phase transition at jets are outlined.

In [1], condensation at jets was investigated, under the assumption of constant liquid flow rate over the jet cross section. In the present work, this investigation is expanded to the case of variable flow rate.

It is assumed that the liquid jet with initial temperature T_0 and specified (at x = 0) velocity distribution over the cross section of a circular aperture of radius R_0 issues into a space filled with vapor of the same liquid, with saturation temperature T_S; the radical component of the temperature gradient is much larger than the axial component. Correspondingly, the momentum and energy equations for the flow of the liquid jet take the form

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$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial P}{\partial x}+g+\frac{1}{y}\frac{\partial}{\partial y}\left(\nu_{ef}\,y\frac{\partial u}{\partial y}\right); \quad (1)$$

$$\frac{\partial (yu)}{\partial x}+\frac{\partial (yv)}{\partial y}=0; \quad (2)$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{1}{y}\frac{\partial}{\partial y}\left(a_{ef}\,y\frac{\partial T}{\partial y}\right). \quad (3)$$

For laminar flow $\nu_{ef}=\nu$, $a_{ef}=a$; for turbulent flow, $\nu_{ef}=\nu+\nu_t$, $a_{ef}=a+a_t$, where ν_t, a_t are the kinematic coefficients of turbulent transfer. The boundary and initial conditions for Eqs. (1)-(3) are as follows:

$$\text{when } x=0,\ u=u_0,\quad T=T_0, \quad (4)$$

$$\text{when } y=0,\ \frac{\partial u}{\partial y}=v=\frac{\partial T}{\partial y}=0, \quad (5)$$

where Eq. (5) follows from the flow symmetry condition; and

$$\text{when } y=H(x),\quad \frac{\partial u}{\partial y}=\frac{\tau_0}{\mu}=B,\quad T=T_s, \quad (6)$$

$$\lambda\frac{\partial T}{\partial y}=\frac{\rho h_{fg}}{y}\frac{d}{dx}\int\limits_0^{H(x)} yu\,dy. \quad (7)$$

Equation (7) expresses the condition of energy conservation at the phase interface [3].

As in [1, 2], the system in Eqs. (1)-(3) is converted to a form convenient for numerical integration. Thus, the equations in dimensionless variables for the momentum and energy variation in the turbulent jets take the form

$$u_C\frac{du_C}{dx}=\frac{1}{\mathrm{We}}\frac{1}{H^2(x)}\frac{dH}{dx}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 u_C}{\partial y_C^2}+\frac{1}{y_C}\frac{\partial u_C}{\partial y_C}\right)+\frac{1}{\mathrm{Fr}}+T_1+\Phi(k-1)\frac{\partial u_C}{\partial y_C}; \quad (8)$$

$$\frac{dy_C}{dx}=A_C\frac{B_N}{1-A_N}+B_C; \quad (9)$$

$$u_C\frac{dT_C}{dx}=\frac{1}{\mathrm{RePr}}\left(\frac{\partial^2 T_C}{\partial y_C^2}+\frac{1}{y_C}\frac{\partial T_C}{\partial y_C}\right)+T_2+\Phi(k-1)\frac{\partial T_C}{\partial y_C}, \quad (10)$$

where T_1 and T_2 are terms depending on the kinetic coefficients of turbulent transfer [2], and the significance of $\Phi(x)$ is as follows:

$$\frac{d}{dx}\int\limits_0^{H(x)} yu\,dy=y\Phi(N-1). \quad (11)$$

The function Φ introduced in this way takes account of the variability in flow rate over the jet cross section. Combining Eqs. (11) and (7), the following expression is obtained for Φ:

$$\Phi=\frac{\lambda}{\rho h_{fg}(N-1)}\left.\frac{\partial T}{\partial y}\right|_{y=H(x)},$$

or in dimensionless variables

$$\Phi=\frac{\xi}{\mathrm{Re\,Pr}(N-1)}\left.\frac{\partial T}{\partial y}\right|_{y=H(x)}, \quad (12)$$

where $\xi = c_p(T_s - T_0)/h_{jg}$ is the supercooling parameter.

The system in Eqs. (8)-(10) is integrated numerically by the Runge–Kutta method. At the inlet ($x = 0$), the function Φ is assumed to be specified; when $x > 0$, it is calculated in each step from Eq. (12). As a result of numerical solution of this system, the velocity

TABLE 1. Comparison of the Experimental Data of [4] on the Evaporation of Water from a Turbulent Jet with Results Obtained Using Eq. (15) (ν = 0.0101 cm^2/sec, R_0 = 0.2 cm, Pr = 7, ξ = −0.02, A = 0.45)

$l/(2R_0)$	Re	Fr	St×100		Δ, %
			experiment	calculation	
42,75	7800	775	0,138	0,128	—7,0
42,75	8400	889	0,134	0,127	—4,5
42,75	8900	1009	0,132	0,125	—4,8
42,75	9500	1150	0,129	0,124	—3,6
42,75	10000	1274	0,127	0,123	—2,9
42,75	10500	1405	0,125	0,122	—2,0
42,75	11200	1598	0,123	0,118	—3,8
33,25	6600	555	0,172	0,150	—12,7
33,25	7700	755	0,157	0,146	—7,1
33,25	9000	1032	0,151	0,143	—5,6
33,25	10000	1274	0,144	0,141	—2,1
23,75	6000	459	0,205	0,196	—4,1
23,75	6000	459	0,203	0,196	—3,1
23,75	6700	572	0,205	0,178	—12,8
23,75	7100	642	0,187	0,176	—5,9
23,75	7600	736	0,187	0,175	—6,8
23,75	8200	857	0,184	0,173	—5,9
23,75	8800	987	0,172	0,171	—1,0
23,75	10000	1274	0,170	0,167	—1,7
23,75	10500	1405	0,168	0,166	—1,6
23,75	11100	1570	0,168	0,164	—2,5
23,75	11700	1744	0,159	0,163	2,5

and temperature distribution over the cross section and length of the jet is found and used to determine the heat-transfer coefficient. The expression for the mean heat-transfer coefficient for a liquid jet, taking account of variation in flow rate over the length of the jet, in contrast to [2], is as follows:

$$\alpha = \frac{\rho c_p}{\Pi \Delta T}\left(1+\frac{T_s c_p}{h_{fg}}\right)^{-1}\int_0^x 2\pi \frac{d}{dx}\int_0^{H(x)} yuT\,dy\,dx, \tag{13}$$

or in terms of the Stanton number and dimensionless variables

$$\mathrm{St} = \frac{\int_0^{H(x)} yuT\,dy\big|_0^x}{(1+\xi)\int_0^x H(x)\,dx}. \tag{14}$$

On the basis of approximation of the numerical solution of Eqs. (8)-(10) with the boundary and initial conditions in Eqs. (4)-(7), for Re = 1500-20,000, Pr = 1-50, Fr = 50-15,000, $|\xi|$ = 0.001-1, using Eq. (14), an expression for the Stanton number is obtained

$$\mathrm{St} = A(0.047 - 0.035\,\xi)\left(\frac{l}{2R_0}\right)^{-0,52}\mathrm{Re}^{-0,033}\,\mathrm{Pr}^{-0,1}\,\mathrm{Fr}^{-0,064}, \tag{15}$$

where A is a parameter taking account of the initial velocity profile; A = 1 if the initial profile is plane and A = 0.18 $(\ell/2R_0)^{0.26}$ if it is parabolic.

It is of interest to compare the formula proposed for the Stanton number with experimental data. In [2], the results were compared with the experimental data of [5]. The initial velocity profile was assumed to be plane from the experimental conditions; the value of the supercooling parameter was in the range 0.04-0.16, and was not taken into account.

Table 1 compares the results of calculation with the experimental data of [4] on the evaporation of water vapor from a circular turbulent jet of radius 0.2 cm. The initial velocity profile is assumed to be parabolic, and therefore A ≈ 0.45; the supercooling parameter ξ = −0.02.

For laminar flow, the approximate formula for the Stanton number takes the form

$$\mathrm{St} = (f_1 f_2 f_3 - 0{,}004\xi)\left(\frac{l}{2R_0}\right)^{-0{,}8} \qquad (16)$$

Here

$$f_1 = 1.25\cdot 10^{-2} - 7.5\cdot 10^{-6}\,\mathrm{Re};\quad f_2 = 1.05 - \mathrm{Pr}\left(8\cdot 10^{-3} - 3\cdot 10^{-4}\,\frac{l}{2R_0}\right);$$

$$f_3 = 1{,}05 - C^{3/4}\,\mathrm{Re}^{-2}\mathrm{We}\left(0.4 + 0.01\,\frac{l}{2R_0}\right) \quad \text{for} \quad \mathrm{We} \geqslant 2{,}5;$$

$$f_3 = 1.05 - C^{3/4}\,\mathrm{Re}^{-2}\,\mathrm{We} + \frac{l}{2R_0}\,(0.03 - 0.6C^{3/4}\,\mathrm{Re}^{-2}\,\mathrm{We}) \quad \text{for} \quad \mathrm{We} < 2{,}5,$$

where $C = gR_0{}^3/\nu^2$.

In [1], the Stanton number was compared with the experimental data of [6]. In this case, the supercooling parameter ξ was no greater than 0.15, and therefore its influence on the heat transfer in condensation at a laminar liquid jet was neglected.

NOTATION

x, y, orthogonal coordinate system related to jet symmetry axis; u, v, components of the velocity vector along the coordinates x and y; T, temperature; ν, kinematic viscosity; a, thermal diffusivity; ρ, density; λ, thermal conductivity; c_p, specific heat at constant pressure; h_{fg}, latent heat of vaporization; $\mathrm{Re} = u_0R_0/\nu$, Reynolds number; $\mathrm{Pr} = \nu/a$, Prandtl number; $\mathrm{Fr} = u_0{}^2/(gR_0)$, Froude number; $\mathrm{We} = \rho u_0{}^2R_0/\gamma$, Weber number.

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