LAMINAR FILM CONDENSATION ON SPHERE WITH ENTRANCE ZONE

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Film condensation on a sphere is numerically analyzed, taking into account inertia and convection terms. An expression is derived for calculating the mean Nusselt number.

It has been shown elsewhere [1] that the equations of a boundary layer have a self-adjoint solution for the vicinity of the stagnation point on a sphere. That study also dealt with laminar film condensation in the vicinity of such a point.

Let the x-coordinate be read from the upper stagnation point (where x = 0) down along a circle on the surface of the sphere and let the distance y be measured along a normal to this surface. Just as before [1], we assume that the sphere is immersed in a boundless volume of pure vapor at its saturation temperature t_s, while the surface of the sphere is maintained at a constant temperature t_1 lower than t_s. A continuous thin film of condensate is flowing down along the surface of the sphere. Its properties do not depend on the temperature and the viscous dissipation energy is so small that it can be disregarded. Under these assumptions we have the following equations of momentum and energy for steady-state laminar axisymmetric flow of a liquid film on a sphere

$$\frac{\partial (ur)}{\partial x} + \frac{\partial (vr)}{\partial y} = 0; \tag{1}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \nu\frac{\partial^2 u}{\partial y^2} + \frac{g(\rho - \rho_v)}{\rho}\sin\frac{x}{R}; \tag{2}$$

$$u\frac{\partial t}{\partial x} + v\frac{\partial t}{\partial y} = a\frac{\partial^2 t}{\partial y^2}, \tag{3}$$

where R is the radius of the sphere; r = R sin x/R, distance from the axis of symmetry; ρ, density of the condensate; ρ_v, density of the vapor; and a, thermal diffusivity.

The boundary conditions are stipulated as

$$\text{at}\quad y = 0 \quad t = t_1,\ u = v = 0; \tag{4}$$

$$\text{at}\quad y = \delta(x) \quad t = t_s,\ \partial u/\partial y = 0. \tag{5}$$

To these boundary conditions (4) and (5) will be added the condition of energy conservation at the boundary between liquid film and vapor

$$k\left.\frac{\partial t}{\partial y}\right|_{\delta(x)} = \frac{\rho h_0}{r}\frac{d}{dx}\int\limits_0^{\delta(x)} ur\,dy, \tag{6}$$

where k is the thermal conductivity and h_0 is the latent heat of phase transformation.

Using the dimensionless variables

$$x = R\bar{x},\quad u = \frac{\nu c^{1/2}}{R}\bar{u},\quad v = \frac{\nu c^{1/4}}{R}\bar{v},\quad y = \frac{R}{c^{1/4}}\bar{y},$$
$$t = \bar{t}(t_s - t_1) + t_1,\quad r = R\bar{r}, \tag{7}$$

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where $c = [g(\rho - \rho_v)/\rho\nu^2]R^3$, we transform the system of equations (1)-(3) and the boundary conditions (4)-(5) into (dashes above symbols will be omitted)

$$\frac{\partial (ur)}{\partial x} + \frac{\partial (vr)}{\partial y} = 0; \tag{8}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \frac{\partial^2 u}{\partial y^2} + \sin x; \tag{9}$$

$$u\frac{\partial t}{\partial x} + v\frac{\partial t}{\partial y} = \frac{1}{\Pr}\frac{\partial^2 t}{\partial y^2}; \tag{10}$$

$$\text{at } y = 0 \quad t = 0, \quad u = v = 0; \tag{11}$$

$$\text{at } y = \delta(x) \quad t = 1, \quad \partial u/\partial y = 0; \tag{12}$$

$$\text{at } y = \delta(x) \quad \left.\frac{\partial t}{\partial y}\right|_{\delta} = \frac{\Pr}{\gamma}\frac{1}{r}\frac{d}{dx}\int_0^{\delta} urdy, \tag{13}$$

where $\Pr = \nu/a$, $\gamma = c_p(t_s - t_1)/h_0$.

We then solve the system of equations (8)-(10) for the boundary conditions (11)-(13) by the method described in earlier studies [2, 3]. In accordance with that method, we introduce lines of constant flow rate $y_k = y_x(x)$ into the full flow and let

$$u_k(x) = u[x, y_k(x)], \quad v_k(x) = v[x, y_k(x)], \quad t_k(x) = t[x, y_k(x)].$$

The solution to the system of differential equations (8)-(10) will be sought in the form

$$u_k = \sum_{j=1}^{N} A_j(x)U_j(y_k(x)); \quad t_k = \sum_{j=1}^{N} A1_j(x)N1_j(y_k(x)), \tag{14}$$

where N is the consecutive number of an approximation and U_j, $N1_j$ are complete systems of functions characterizing the surfaces $y_k(x)$ through which the liquid does not flow. The values of u_k, t_k, y_k are determined from the system of differential equations

$$u_k\frac{du_k}{dx} = \Phi(k-1)\frac{\partial u_k}{\partial y_k} + \frac{\partial^2 u_k}{\partial y_k^2} + \sin x; \tag{15}$$

$$\frac{dy_k}{dx} = \frac{dy_{k-1}}{dx} + \frac{2\Phi - (y_k - y_{k-1})\left[\frac{du_k}{dx} + \frac{du_{k-1}}{dx} + (u_k + u_{k-1})\operatorname{ctg} x\right]}{u_k + u_{k-1}}; \tag{16}$$

$$u_k\frac{dt_k}{dx} = \Phi(k-1)\frac{\partial t_k}{\partial y_k} + \frac{1}{\Pr}\frac{\partial^2 t_k}{\partial y_k^2}. \tag{17}$$

Function Φ in Eqs. (15)-(17) is determined from the solution to the equation

$$\frac{d}{dx}\int_0^{y_k} urdy = r\Phi(k-1). \tag{18}$$

In the entrance section (x = 0) were stipulated initial conditions for the distributions of velocity and temperature, also a grid of mathematical lines. The value of function Φ in the entrance section was assumed to be zero.

The system of nonlinear differential equations (15)-(17) with these boundary and initial conditions was solved by the Runge-Kutta method, with the value of function Φ calculated according to Eq. (18) from the preceding step as well as by the method of iterations. Both procedures yielded values with negligible difference already after a few integration steps.

This algorithm was then used for determining the velocity field, the temperature field, and the thickness of the liquid film at various distances from the stagnation point, whereupon the mean Nusselt number was calculated on the basis of those data according to the relation

$$\mathrm{Nu} = \bar{\alpha}2R/k. \tag{19}$$

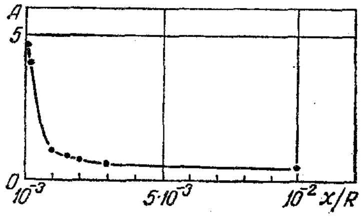

Fig. 1. Variation of ratio Nu/Nu_o = A with increasing distance from stagnation point: dots indicate calculated values; curve based on expressions (24) and (25).

The mean heat-transfer coefficient to the surface $\bar{\alpha}$ was calculated from the local coefficient α according to the relation

$$\bar{\alpha}=\frac{1}{x}\int_0^x \alpha dx, \tag{20}$$

where

$$\alpha=\frac{k}{t_1-t_s}\left(\frac{dt}{dy}\right)_0. \tag{21}$$

The local heat-transfer coefficient can be calculated by integrating Eq. (10) over the thickness of the condensate film, taking into account relations (13) and (18). This yields

$$-ra\left(\frac{dt}{dy}\right)_0=\frac{d}{dx}\int_0^{\delta(x)} ur\left(t-t_s-\frac{h_0}{c_p}\right)dy. \tag{22}$$

From relations (21) and (22) follows the expression for the local heat-transfer coefficient

$$\alpha=\frac{\rho c_p v c^{1/4}}{Rr}\frac{d}{dx}\int_0^{\delta(x)} ur\left(t-1-\frac{1}{\gamma}\right)dy;$$

this expression together with relations (19) and (20) giving the mean Nusselt number in the form

$$\mathrm{Nu}=2\mathrm{Pr}\,c^{1/4}\frac{1}{x}\int_0^x \frac{1}{r}\frac{d}{dx}\int_0^{\delta(x)} ur\left(t-1-\frac{1}{\gamma}\right)dydx. \tag{23}$$

The other expression for the Nusselt number [1]

$$\mathrm{Nu}_0=k_1c^{1/4}(\mathrm{Pr}/\gamma)^{1/4} \tag{24}$$

contains a certain mean proportionality factor k_1 which has been found to be k_1 = 1.35 on the basis of empirical data.

The ratio Nu/Nu_o = A of the Nusselt number based on relation (23) to the Nusselt number based on relation (24) varies as the distance from the stagnation point increases. The trend of this variation is shown in Fig. 1, it being evident here that the ratio A decreases asymptotically to a constant value as the distance from the stagnation point increases.

The data can be approximated with the expression

$$\mathrm{Nu}=\mathrm{Nu}_0 A,$$

where

$$\begin{aligned} &A=5-8\cdot10^3(x/R)\ \text{for}\ 0<x/R\leqslant5\cdot10^{-4};\\ &A=1-2\cdot10^2(x/R)\ \text{for}\ 5\cdot10^{-4}\leqslant x/R\leqslant3\cdot10^{-3};\\ &A=0{,}4\ \text{for}\ x/R\geqslant3\cdot10^{-3}. \end{aligned} \tag{25}$$

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INTENSIFICATION OF HEAT TRANSFER DURING VAPOR CONDENSATION ON THE OUTSIDE SURFACE OF VERTICAL TUBES WITH ANNULAR SWIRL VANES

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It is shown that it is possible to significantly intensity heat exchange in vertical condensers consisting of tubes with annular membranes inside and grooves on the outside.

Reducing the weight and size of condensers used in power engineering, the chemicals industry, and other areas of technology is an important scientific—technical and economic problem. One of the most promising methods of doing this is to intensify heat transfer as a result of artificial agitation of the flow.

Extensive data has by now been accumulated on intensifying heat transfer during condensation on the outer surface of tubes. However, practical realization of most of the schemes proposed has been made difficult by the lack of technology for mass-producing the investigated heat-transfer surfaces, the need to develop special technology for assembling the heat exchangers, their comparatively low efficiency, and the fact that heat exchange is not simultaneously intensified inside the tube.

The Moscow Aviation Institute has developed a highly efficient method of intensifying heat exchange in tubular heat exchangers and has conducted extensive studies of the method in tubes and annular channels and with longitudinal flow about tube bundles for both liquids and gases [1]. The essence of the proposed method is the rolling of periodically arranged annular grooves on the outer surface of the tubes (Fig. 1). These grooves, together with annular membranes with a planar configuration formed on the inside surface of the tubes, agitate the flow in the boundary layer and intensify heat transfer both outside and inside the tubes. In this case, there is no increase in the outside diameter of the tubes, which means that it is not necessary to significantly alter the existing technology for assembling tubular heat exchangers. The above tubes are also not highly susceptible to obstruction, and salt deposits in them are minimal. The grooved tubes are made on standard equipment.

This article presents findings from new experimental studies of intensifying heat transfer during film condensation on the outer surface of vertical tubes with annular swirl vanes.

The studies were performed on water and solvent spirit. Tubes with an outside diameter D_t = 16 mm, inside diameter D = 14 mm, length of 1.5 m, and four groove variants ($1 - d_t/D_t$ = 0.876; t/D_t = 0.248; 2 — 0.938 and 0.248; 3 — 0.938 and 0.437; 4 — 0.91 and 0.437) were placed in a smooth tube with an inside diameter of 26 mm. The width of the annular grooves was 2 mm. We also studied one section with a smooth tube for comparison. Coolant water was fed inside the tube from bottom to top, while condensing vapor was sent in the opposite direction in the annular channel (Fig. 1).

We varied the following parameters: temperature of the vapor at the condenser inlet, temperature of the outside surface of the tube wall at nine points along the height, tempera-

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