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VAPOR CONDENSATION ON HORIZONTAL WIRE-PROFILED TUBES

V. G. Rifert and A. B. Golubev UDC 536.423.4

The effect of the profiling parameters and the nonisothermality of the walls on the vapor condensation rate is analyzed for the case of a horizontal tube profiled by means of a wire spiral.

The degree of intensification of the process of vapor condensation on profiled as compared with smooth horizontal tubes depends to a large extent on the flooding by the condensate of the lower surface of the tube [1] and, moreover, on the thermal conductivity and thickness of the wall [2]. Our investigations [1] of the bottom flooding of horizontal tubes showed that tubes with spiral wire fins have a much smaller bottom flooding angle φ_f than do tubes profiled by deformation of the wall, for similar values of the We number which in this case characterizes the ratio of the forces of surface tension drawing the condensate to the base of the fins to the force of gravity.

In [3] an approximate theoretical solution of the problem was given for the case of vapor condensation on a horizontal wire-profiled tube. In the present article certain assumptions of [3] are refined and the effect of the profiling parameters and the physical properties of the condensate on the vapor condensation enhancement is analyzed in detail.

The assumptions made in [3] for the purposes of a theoretical solution of the problem of vapor condensation on a horizontal tube with spiral wire finning are analogous to the Nusselt theory for the condensation of stationary vapor on a vertical wall. Here we will consider the case where We > 5 and over the entire perimeter of the tube, except for the zone with the bottom layer, the condensate flows from the center between the wires under the action of surface tension predominantly along the normal to the wires and, under the force of gravity, drains along the wires to the bottom of the tube (Fig. 1).

In order to refine the inequality We > 5 we will estimate the radius of curvature of the film R, which enters into We, from the expression obtained in [4] by studying the geometry of the liquid film forming the meniscus (near the wire):

$$R = l_2^2/2d. \tag{1}$$

From the solution of the differential equation for the growth of the stream width (see below), over a broad interval of variation of the heat flux q and the tube diameter and for various liquids $l_2 \leqslant d/2$. Taking $l_2 = d/2$, we determine the pitch of the spiral S ensuring We $\geqslant$ 5:

$$S \leqslant 3.2\sigma/\rho g d. \tag{2}$$

At d = 1.5 mm for water vapor and ammonia S $\leqslant$ 13 mm, for freons S $\leqslant$ 2.5-4 mm. For this pitch and the tube diameters D > 16 mm encountered in practice the maximum inclination of the spiral relative to the vertical for a uniform single-thread winding is ≈16°. Accordingly, in the solution of the problem we can neglect the inclination of the spiral and treat the finning as annular (transverse). In [3] the following differential equation was obtained for the flooding by the condensate of the interwire space in terms of the angular coordinate φ for a tube with spiral wire fins at a wall temperature constant with respect to φ and x:

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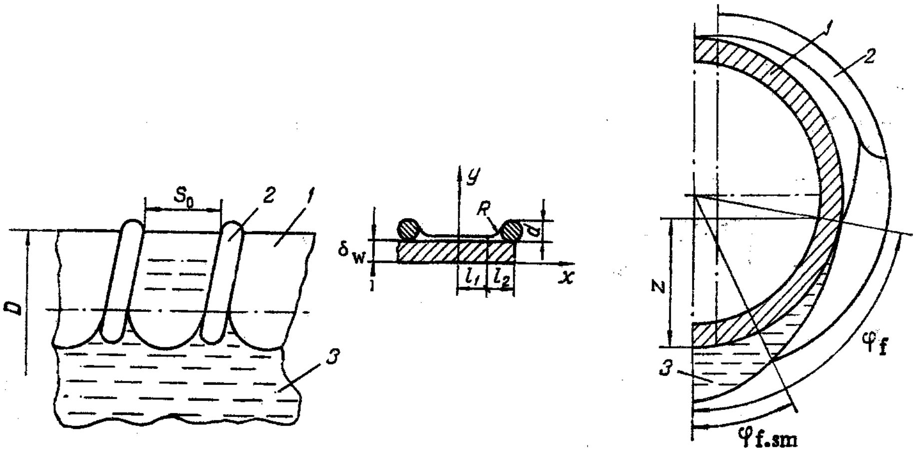

Fig. 1. Pattern of condensate flow along a horizontal wire-profiled tube: 1) tube wall; 2) wire; 3) condensate.

$$\frac{d\eta}{d\varphi} = \frac{BC(1-\eta)^{0.5}}{\eta^{1.5}\sin\varphi^{0.75}} - \frac{3}{8}\eta\cot\varphi. \tag{3}$$

The numerical solution of this equation can be approximated by the expression

$$\eta = 1{,}376(BCH)^{0.4}. \tag{4}$$

In (3) and (4) $\eta = l_2/l$; $B = \frac{28.5}{\rho D^2}\left(\frac{\lambda\Delta T}{rg}\right)^{0.75}(\sigma\nu)^{0.25}$, $C = D^3/dl^2$, $H = \varphi/\sin^{0.75}\varphi$.

In order to obtain the relation between the stream width l_2 and the condensate flow rate G_c, it is necessary to solve the equation of motion of the condensate film on the interval l_2 under the action of gravity, written in Cartesian coordinates for various angles of inclination of the plane surface:

$$\mu\left(\frac{\partial^2 w_\varphi}{\partial x^2} + \frac{\partial^2 w_\varphi}{\partial y^2}\right) = -\rho g \sin\varphi. \tag{5}$$

On the interval l_2 the condensate film has a constant curvature with radius R. The boundary conditions for (5) are: $w_\varphi = 0$ at $y = 0$; $\partial w_\varphi/\partial y = 0$ at $y = \delta$. The numerical solution of (5), the results of which were compared with the experimental data in [4], is approximated by the expression

$$G_c = 0.01\frac{\rho l_2^2 d^{1.25}(g\sin\varphi)^{0.75}}{\nu^{0.5}}. \tag{6}$$

In [2] the differential equation for the flooding by the condensate of the interfin channel was calculated up to $\varphi = \pi - \varphi_{f.sm}$ where $\varphi_{f.sm}$ is the angle embracing the bottom layer of the smooth tube. As our investigations showed [1], for profiled tubes φ_f may considerably exceed $\varphi_{f.sm}$ In [1] an experimentally verified method of calculating φ_f for profiled tubes was proposed, based on the analogy between flooding and the capillary rise of a liquid in vertical grooves [5]. In [1] a dependence is given for determining the rise Z_1 corresponding to total flooding of the groove or interwire space by condensate, on the assumption that the volume of liquid raised above Z_1 is sufficient for total flooding of the groove to height Z_1. The angle $\varphi_{f.1}$ corresponding to Z_1 is equal, according to [1], to

$$\varphi_{f.1} = \arccos\left[1 - \frac{8\pi\sigma}{\rho g D(8l - \pi d)}\right]. \tag{7}$$

Then the angle φ_3 (Fig. 1) corresponding to the maximum rise of the condensate over the entire width S can, assuming a linear distribution of liquid with respect to φ from $\varphi_{f.sm}$, be estimated as

$$\varphi_f = 2\varphi_{f.1} - \varphi_{f.sm}. \tag{8}$$

Then the mean heat transfer is given by

$$\bar{\sigma}_p = 2G_c r/\pi D l \Delta T. \tag{9}$$

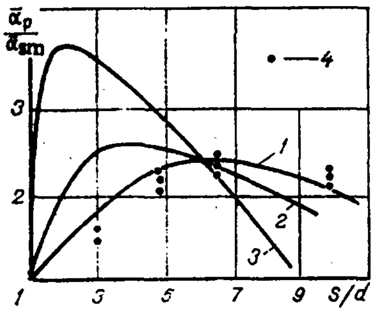

Fig. 2.

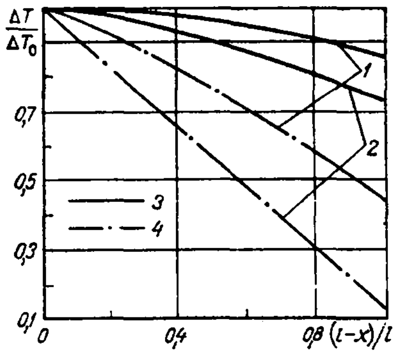

Fig. 3.

Fig. 2. Intensification of vapor condensation due to wire profiling: 1-3) curves calculated from (11) (1 – water vapor at T_i = 373°K; 2 – ammonia, T_i = 300°K; 3 – Freon-12, T_i = 300°K); 4) experimental data for water vapor.

Fig. 3. Variation of temperature head on the interval l_1 of condensate flow towards the base of the fins for the same wall thickness δ_w at the base: 1) Bi = 0.3; 2) 1.4; 3) wire fins; 4) rectangular fins, height h = 1 mm.

To analyze the effect of the profiling parameters on the rate of vapor condensation on a horizontal tube we find the ratio of α_p to the heat transfer for a smooth tube α_{sm}, which can be calculated from the Nusselt relation for a single tube:

$$\bar{\alpha}_{sm} = 0.728 \left(\frac{\lambda^3 r \rho g}{\nu \Delta T D} \right)^{0.25}. \tag{10}$$

In dimensionless form, using (4), (6), (9) and (10) to calculate $\bar{\alpha}_p$, the ratio $\bar{\alpha}_p/\bar{\alpha}_{sm}$ can be written

$$\frac{\bar{\alpha}_p}{\bar{\alpha}_{sm}} = 0.11\,(\mathrm{Pr}\,K)^{0.15}\,\mathrm{Ga}^{0.10}\,\mathrm{We}_l^{0.2} \left(\frac{d}{D} \right)^{0.25} \sin^{0.75}\varphi\, H^{0.8}. \tag{11}$$

Here

$$\mathrm{We}_l = \sigma d / \rho g l^3, \tag{12}$$

and $\varphi = \pi - \varphi_f$ where φ_f is calculated from (7) and (8).

Relation (11) makes it possible to determine the heat transfer enhancement due solely to the constricting effect of the wires without allowance for vapor condensation on the wire itself, which may occur if there is good contact between the wire and the surface. In this connection it should be noted that (11) is valid only when the flow of the condensate film to the wires under the action of surface forces is predominant, i.e.. when We = $2\sigma/\rho g S R > 1$. In this case the radius of curvature must be calculated from (1), with l_2 being found from (4).

According to (11), the tube diameter affects the ratio $\bar{\alpha}_p/\bar{\alpha}_{sm}$ only through φ_f (relation (7)). For small-diameter tubes and vapor whose condensate has a large value of σ, the effect of profiling will be only slight. Figure 2 presents the results of calculating $\bar{\alpha}_p/\bar{\alpha}_{sm}$ from (11) as a function of S/d for the condensation of H_2O at $T_i \approx 373$°K, NH_3 and F- 12 at $T_i \approx$ 300°K, ΔT = 1°K and D = $16 \cdot 10^{-3}$ m. By virtue of (11) the variation of T_i over a broad interval has little effect on $\bar{\alpha}_p/\bar{\alpha}_{sm}$. As ΔT increases, the ratio $\bar{\alpha}_p/\bar{\alpha}_{sm}$ falls, to a greater degree than for a smooth tube.

For liquids with different surface tensions, as a result of the different values of φ_f, at the same S and d the maximum of $\bar{\alpha}_p/\bar{\alpha}_{sm}$ is reached at different values of S/d. The smaller σ, the smaller the value of S/d at which the maximum enhancement is reached. Figure 2 also gives the experimental data on the condensation of almost stationary water vapor on a single brass tube with various finning pitches at d = 1.5 mm and $T_i \approx 373$°K. In the experiments the mean wall temperature was measured by means of a copper resistance thermometer laid in the wall along the spiral. Clearly, the agreement between the experimental and the theoretical data is quite satisfactory.

It is legitimate to calculate $\bar{\alpha}_p$ from these relations when T_w = const for all φ and the tube generator (with respect to x).

The variation of ΔT with respect to x is found by solving the heat conduction equation for the wall:

$$\frac{\partial^2 T_w}{\partial y^2}+\frac{\partial^2 T_w}{\partial x^2}=0, \tag{13}$$

where y is reckoned from the inner surface of the tube.

The boundary conditions are

$$x=0,\ x=l,\ \frac{\partial T_w}{\partial x}=0; \tag{14}$$

$$y=0,\ T_w=T_{w.0};\ y=\delta_w\ \lambda_w\left(\frac{\partial T_w}{\partial y}\right)=-\frac{\lambda}{\delta_x}\Delta T. \tag{15}$$

The thickness of the condensate film on the interval l_1, where the condensate flows toward the wire, is determined by analogy with [2] from the relation

$$\delta_x=\left[\frac{4\mu\lambda\Delta Tx}{\rho r\,|dP/dx|}\right]^{0.25}. \tag{16}$$

As a result of the numerical solution of (13) we obtained the temperature distribution with respect to x:

$$\frac{\Delta T}{\Delta T_0}=\left(\frac{x}{l}\right)^{0.067\mathrm{Bi}}. \tag{17}$$

Here $\Delta T = T_i - T_0$ is the temperature drop at $x = l$, which is given beforehand in the solution. From (17) we find the mean ΔT with respect to l:

$$\Delta\bar{T}=\frac{\Delta T_0}{1+0.067\,\mathrm{Bi}}. \tag{18}$$

The value of $\overline{\Delta T}$ is substituted in expression (4).

According to (18), at Bi < 1 it is possible to assume that $\overline{\Delta T} \approx \Delta T_0$, i.e. T_w = const with respect to x.

Comparing (17) with the expression for the variation of ΔT over the height of the fins from [2], we find (Fig. 3) that for the same Bi the nonisothermality on finned surfaces is much greater than on a surface profiled with wire. As shown in [6, 7], for water vapor condensing on finned stainless steel and titanium tubes (Bi $\gg$ 1) there is no heat transfer enhancement, whereas on wire-finned tubes enhancement occurs [8].

Investigations of the condensation of water vapor [4, 8] on a bundle consisting of seven rows of tubes arranged vertically and the testing of a horizontal-tube ammonia condenser [9] consisting of 34 rows of tubes have shown that, as compared with smooth tube bundles, there is a more than 1.5-fold enhancement of heat transfer. However, obtaining a theoretical expression for the mean heat-transfer coefficient over a bundle of profiled tubes will require a more detailed investigation.

NOTATION

λ, ν, a, c_p, r, σ, and ρ, thermal conductivity, kinematic viscosity, thermal diffusivity, heat capacity, heat of vaporization, surface tension and liquid density, respectively; λ_w, thermal conductivity of the wall; S, distance between wires; l_2, width of the condensate stream along the wire; D, tube diameter; R_{in}, inside radius of the tube; d, diameter of the wire; $l = S/2$; x, coordinate along the generator of the tube; y, coordinate normal to x; z, coordinate in the direction of flow of the condensate along the wire; δ_w, tube wall thickness; δ_x, thickness of the condensate film; δ_l, same at $x = l$; T_i, vapor condensation temperature; T_w, wall temperature at the boundary with the condensate film; T_0, wall temperature near the wire; α, heat transfer from the vapor to the wall; α_{is}, same for an isothermal wall; α_0, the heat transfer to the cooling medium; $\Delta T = T_i - T_w$; $\Delta T_0 = T_i - T_0$. $\mathrm{Pr} = \nu/a$; $K = r/c_p\Delta T$; $\mathrm{Ga} = gD^3/\nu^3$; $\mathrm{We} = 2\sigma/\rho g S R$; $\mathrm{We}_l = \sigma d/\rho g l^3$; $\mathrm{Bi} = \lambda\delta_w/\lambda_w\delta_l$; $|dP/dx| \approx 2\sigma d/l_1 l_2^2$.

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HEAT AND MASS TRANSFER IN THE CONDENSATION REGION OF VAPOR FILTERING IN A DISPERSE LAYER

V. L. Ganzha and G. I. Zhuravskii

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A calculation of the process of water-vapor condensation as it filters through a disperse medium is given. Relations are obtained for determining the rate of growth and the magnitude of the heated zone.

Recently, there has been growing interest in the investigation of vapor condensation in the course of filtration in disperse media, in connection with the prospects for the use of vapor–thermal methods of intensifying petroleum and gas extraction. Various empirical and semiempirical relations for the calculation of the magnitude and growth rate of the heated (vapor-treated) zone of a petroleum deposit in the form of a disperse sphere have been widely used in practice.

In the present work, on the basis of physical and mathematical models of the vapor-condensation process in a disperse medium [1, 2], an attempt is made to obtain more general analytical relations for calculating the heating dynamics of vapor-treated petroleum deposits. The following assumptions are made: that filtration is one-dimensional; that the condensation region is of depth ΔH; that grad T = 0 and grad P = 0 over the whole region; hat there are no heat losses; that the vapor flow rate at the inlet to the layer is constant and equal to G_V. Thus, the energy equation of the system [2], written for the condensation zone, takes the form

$$[(1-\varepsilon)\rho_c c_p^c + \varepsilon\sigma_w\rho_w c_p^w + \varepsilon\sigma_v\rho_v c_p^v]\frac{dT}{d\tau} = \xi_p r_p - [\varepsilon\sigma_v\rho_v\psi_v + \varepsilon\sigma_w\rho_w\psi_w]\frac{dP}{d\tau}, \tag{1}$$

where

$$\psi_v = T\left(\frac{\partial S_v}{\partial P}\right)_T; \quad \psi_w = T\left(\frac{\partial S_w}{\partial P}\right)_T.$$

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