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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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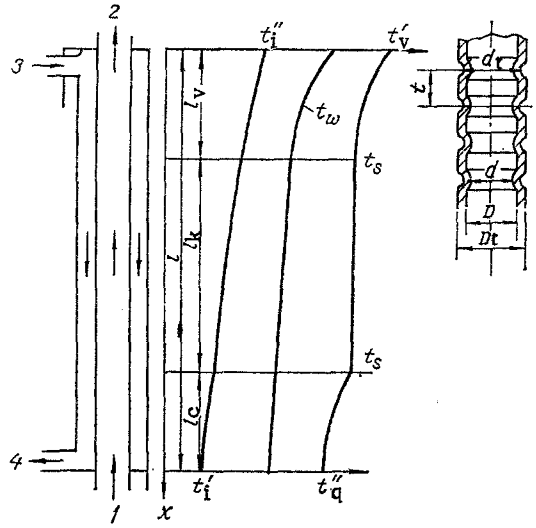

Fig. 1. Diagram of temperature change along test section and of a tube with annular swirl vanes: 1) water inlet; 2) water outlet; 3) vapor inlet; 4) condensate outlet; l_v, l_c, l_k) sections associated with cooling of superheated vapor, condensation, and cooling of condensate.

ture of the condensate at the outlet, temperature of the coolant water at the inlet and outlet, flow rate of vapor, condensate, and coolant water. The saturation temperature was determined from the pressure measured in the condenser. Heat flux was determined from the quantity of condensate formed and from the heating of the coolant water. The discrepancy in the heat balance did not exceed ±5%. The error of the determination of the heat-transfer coefficient, corresponding to a confidence level of 0.997, did not exceed 10%.

The tests on extraction benzine were conducted at a pressure $p = 0.05$-0.07 MPa, saturation temperature $t_s = 50$-$69°C$, vapor temperature at the inlet $t_v' = 70$-$76°C$, condensate temperature at the outlet $t_q'' = 23$-$59°C$, wall temperature $t_w = 26$-$48°C$, temperature heat $\Delta t = t_s - t_m = 4$-$45°C$, heat flux $q_w = (6$-$51)\cdot 10^3$ W/m², and vapor velocity at the inlet to the annular gap $w_v = 2.27$-11.5 m/sec. The tests on water vapor were conducted at $p = 0.07$ MPa, $\Delta t = 4$-$45°C$, and $q_w = (4$-$30)\cdot 10^4$ W/m². For both condensing media, the temperature of the coolant water at the inlet $t_i' = 8$-$12°C$, water velocity $w_i = 0.2$-2.2 m/sec, Reynolds number for the water $Re_i = (2$-$30)\cdot 10^3$, Reynolds number for the film $Re_v = 80$-800, and the dimensionless parameter $Z = (Ga)^{1/3}\lambda_q\cdot\Delta t/r\mu_q = 3\cdot 10^2$-$4\cdot 10^3$.

Figures 2 and 3 compare data obtained on mean heat transfer on the condensation section on the outer surface of the tube with the results for condensation for the smooth tube. The parameter $Re = \alpha\Delta t l_k/r\mu_q$, and the correction for the variability of the physical properties of the condensate $\varepsilon_t = [(\lambda_w/\lambda_s)^3\mu_s/\mu_w]^{1/8}$ [2]. The liquid properties entering into Z and Re are determined from the temperature t_s, while the linear dimension l_k corresponds to the length of the tube section over which condensation occurs. The above-noted measurements and well-known data on heat transfer inside grooved tubes [1] allows us to find l_k and the mean heat-transfer coefficient on the condensation section α. The calculation was performed by the method of successive approximations. In the first approximation, the lengths of the sections associated with cooling of the superheated vapor l_v, condensation l_k, and cooling of the condensate l_c are proportional to the heat fluxes on the corresponding sections. Then for each section we find the mean temperatures of the wall and coolant water and the heat-transfer coefficients for the water. This allows us to refine the values of l_v, l_k, and l_c and determine the mean-logarithmic heat and mean heat-transfer coefficient for the condensation section.

It can be seen from Figs. 2 and 3 that the data obtained for condensation of vapors of water and extraction benzine on the smooth tube agree well with the relation in [2] for film condensation with laminar flow of the film

$$Re_{sm} = 0.943\,Z^{0.75}. \qquad (1)$$

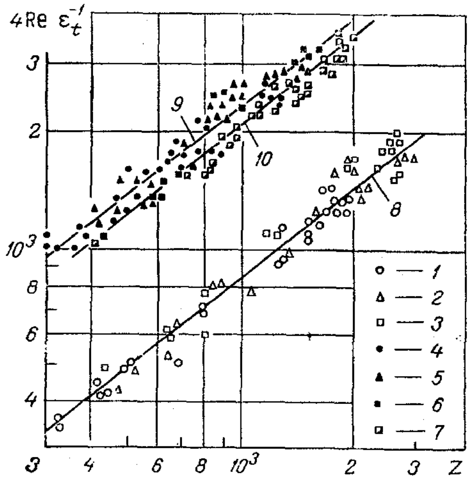

Fig. 2. Heat transfer in the film condensation of benzine on a vertical tube: 1, 2, 3) smooth tube for p = 0.05, 0.06, and 0.07 MPa; 4, 5, 6) tube No. 1 with d_t/D_t = 0.876, t/D_t = 0.248 for the same pressures; 7) tube No. 2 with d_t/D_t = 0.938, t/D_t = 0.248 for p = 0.07 MPa; 8) relation (1); 9, 10) relation (2) for tubes Nos. 1 and 2.

The intensification of heat transfer on the outer surface of the tubes does not depend on the saturation pressure or the parameter Z. It is greater, the greater the depth of the annular grooves, and it decreases slightly with an increase in the spacing of the grooves. The heat-transfer coefficient during condensation of benzine vapors increases by a factor of 2.4-2.8 compared to the smooth tube, while the increase for the water vapor condensation is 1.7-1.9. This difference can be explained as follows. Since the heat of vaporization is 7-8 times greater and the thermal conductivity of the film 3-4 times greater for water than for benzine, the heat flux on the wall is about one order greater in the condensation of water. The Kh18N10 tubes used in the tests has a relatively low thermal conductivity, so the mean heat-transfer coefficient on these tubes decreases with an increase in heat flux due to an increase in the temperature nonuniformity on the tube projections and depressions. Corresponding estimates are given in [3]. This effect should not have any influence for tubes with a high thermal conductivity (such as brass tubes), and the results for water in this case are close to those for benzine.

Since we determined the mean heat-transfer coefficient for the condensation section in our tests as the ratio of the mean flux on this section to the mean temperature head along the length, then their ratio for the smooth and grooved tubes corresponds to the increase in heat flux during condensation.

The data obtained on heat-transfer intensification is generalized by the following relations (d_t/D_t = 0.876-1):

1) for condensation of benzine vapors (t/D_t = 0.248)

$$\frac{\alpha}{\alpha_{sm}} = 1 + 1{,}961\,\{1 - \exp[-20.189\,(1 - d_t/D_t)]\}, \tag{2}$$

2) for condensation of water vapors (t/D_t = 0.248-0.437)

$$\frac{\alpha}{\alpha_{sm}} = 1 + \left(1{,}4 - 1{,}601\,\frac{t}{D_t}\right)\left\{1 - \exp\left[-18{,}203\left(1 - \frac{d_t}{D_t}\right)\right]\right\}. \tag{3}$$

We were able to generalize the test data on condensation of benzine and water vapors with the following relation by introducing the dimensionless complex $\lambda_w\delta_w/\lambda_{fi}d_e$, where λ_w, λ_{fi} are the thermal conductivities of the material of the wall and the smooth film; δ_w is the wall thickness; d_e is the equivalent diameter of the annular channel

$$\frac{\alpha}{\alpha_{sm}} = 1 + 0.551\left(\frac{\lambda_w\delta_w}{\lambda_{fi}d_e}\right)^{0.629}\left(1.196 - 0.79\frac{t}{D_t}\right) \times \{1 - \exp[-19.196\,(1 - d_t/D_t)]\}. \tag{4}$$

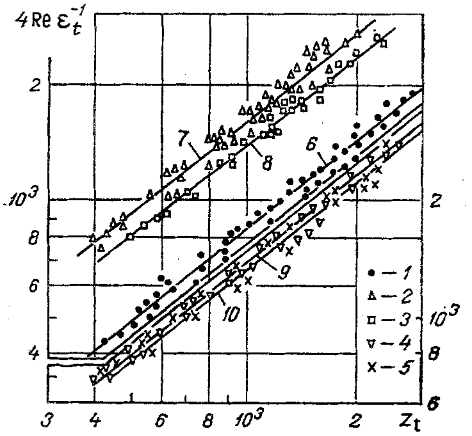

Fig. 3

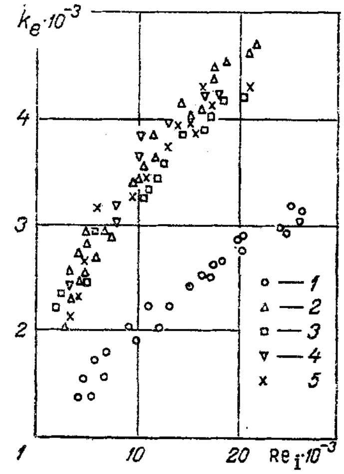

Fig. 4

Fig. 3. Heat transfer in the film condensation of water vapor on a vertical tube: 1) smooth tube; 2, 3, 4, 5) tubes Nos. 1, 2, 3, 4; 6) Eq. (1): 7, 8, 9, 10) Eq. (3) for tubes Nos. 1, 2, 3, and 4, respectively.

Fig. 4. Heat-transfer coefficients for different tubes: 1-5) same as in Fig. 3. $k_e \cdot 10^{-3}$, W/m•K.

Equation (4) is valid for $\lambda_w \delta_w / \lambda_{fi} d_e$ = 2.576-8.514, t/D_t = 0.25-0.44, d_t/D_t = 0.876-0.938.

Since the use of tubes with rolled-in grooves simultaneously appreciable intensifies heat exchange inside the tube at the inlet (by a factor of up to 2-2.5 in the tests in question), on the whole the grooves make it possible to reduce the volume of the condensers 1.5-2 times. This is a significantly better improvement than can be made by other methods (Fig. 4). For example, the use of longitudinally rolled tubes or tubes with longitudinal wire fins increases the heat-transfer coefficient by 1.4-1.6 times at most [4].

It should be noted that we compared heat exchangers of the same length and that the one with grooved tubes did not function in the optimum regime, since condensation took place on only part of its length (the film of condensate was supercooled on the lower part of the tube). If the heat exchanger with grooved tubes had been operated under optimum conditions, it would have been even more efficient than the smooth-tube unit and the heat-transfer coefficient would have increased by a factor of 1.9-2.2.

Thus, the test data showed that when the tubes are positioned vertically, the chosen tube profile helps organize condensate discharge and reduces the thickness of the condensate film at the tops of the annular projections. Accordingly, it increases the rate of heat transfer during vapor condensation.

NOTATION

D_t, outside diameter of tube; d_t, diameter of annular grooves; t, spacing of grooves; g = 9.8 m/sec²; r, heat of vaporization; α, coefficient of heat transfer; ρ, density; λ, ν, μ, thermal conductivity and kinematic and absolute viscosities; $Ga = gl^3{}_k/\nu^2{}_k$, Galileo number; Re, Reynolds number of film; ε_t, correction for variability of physical properties of condensate. Indices: w, wall; s, saturation; q, liquid; v, vapor; i, cold liquid; fi, film; sm, smooth tubes; k, condensate.

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