INFLUENCE OF IRRIGATION DENSITY ON HEAT TRANSFER DURING CONDENSATION IN FINNED TUBE BUNDLES

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The authors presented results of experimental investigations of heat transfer with film condensation of practically motionless vapor in finned tube bundles of different geometry.

This paper is a continuation of systematic investigations of heat transfer with film condensation of vapor on horizontal finned cylinders. The measurements were made using the technique described in [1], with wide variation of the saturation temperature and the specific heat flux. The experiments were conducted on bundles of ten horizontal tubes, located in a vertical bank. Three series of measurements were used, with condensation of cooling agents R11 and R12. In the first series of tests, the only varying geometric parameter of the finning was the distance between the fins. In the second series we varied only the fin thickness. In these two series the shape of the fin was rectangular, and the fin height and the tube diameter were not changed. In the third series we investigated heat transfer in tubes with a trapezium-shaped fin profile.

In all the series the coefficient of efficiency of the fins, calculated according to [3], was not less than 0.94. Some of the results of the measurements have been presented in [1, 2] in tabular form.

Some data of the first series of tests, showing the influence of the liquid properties and the distance between the fins on the variation of heat transfer intensity as a function of the irrigation density, are shown in Fig. 1. Here

$$\mathrm{Nu}^* = \alpha\,(\nu^2/g\,(1-\rho''/\rho'))^{1/3}/\lambda;\quad \mathrm{Re} = qh/(\mu r\,(1+3/8K)\cos\varphi). \tag{1}$$

The Nusselt and Reynolds numbers were computed from the heat flux on the side surface of the fin. However, in the tests we measured the average amount of heat taken from the tube. In calculating the values of $q$ and $\alpha$ in Eq. (1) the heat flux on the fin face and on the trough were found from the data of the authors [4] for horizontal smooth tubes. This way of determining the heat flux does not allow for possible variation of heat transfer on the face and the trough of the fin due to the action of surface tension forces.

Fig. 1 shows data on heat transfer during condensation of the vapors of two liquids in tubes of the same geometrical dimensions. It is clear that the influence of irrigation density on heat transfer intensity depends appreciably on the liquid properties. The data on heat transfer during condensation of a single liquid in tubes with different distances between fins (Fig. 1b) show that there is no single-valued influence of irrigation density on the heat surface. It was observed in [5] that one of the parameters governing heat transfer during condensation on a finned tube is the dimensionless distance between the fins, $\bar{a}$. It can be seen from the data shown in Fig. 1 that this parameter appreciably affects the heat transfer intensity also during condensation in a bundle of tubes.

Variation of the thickness of the fin face in the range $0.4 \leqslant \bar{b} \leqslant 3.0$ showed a weak influence on the variation of heat transfer over the depth of the bundle, and it was not taken into account in subsequent reduction of the experimental data.

Motion and still pictures of the condensation process have shown that liquid falling into the lower-lying tube, under the influence of surface tension forces, does not flow over the entire vertical part of the fin. The liquid flow scheme in a finned tube located in a bundle is shown in Fig. 2a. The film thickness over the major part of the side surface of

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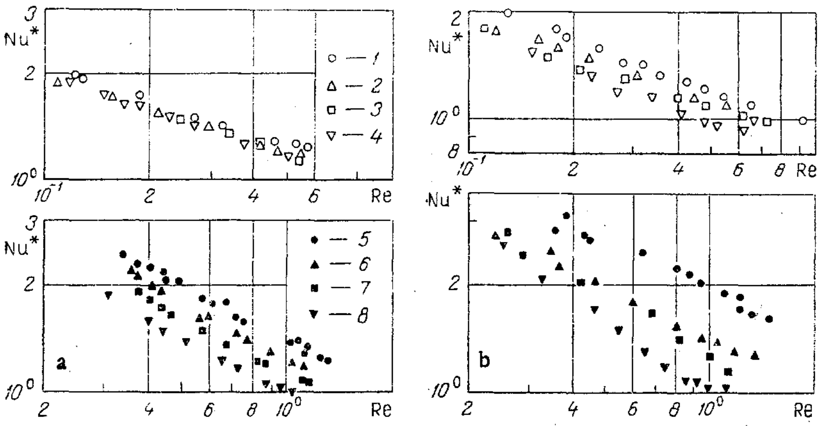


Fig. 1. Heat transfer during condensation of a motionless vapor in bundles of finned tubes, T" = 40°C: a) open points — R11, $\bar{a}$ = 0.89, $\bar{\Gamma}$ = 1 (1); 3.2 (2); 4.7 (3); 5.4 (4); closed points — R12, $\bar{a}$ = 1.26, $\bar{\Gamma}$ = 1 (5); 1.9 (6); 3.7 (7); 5.6 (8); b) open points — R12, $\bar{a}$ = 0.55, $\bar{\Gamma}$ = 1 (1); 2.1 (2); 3.6 (3); 7.4 (4); closed points — $\bar{a}$ = 29, $\bar{\Gamma}$ = 1 (5); 2.2 (6); 4.3 (7); 7.2 (8).

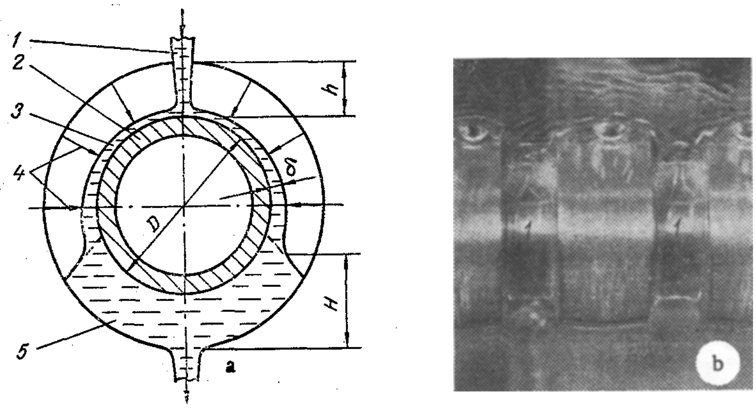


Fig. 2. Flow of liquid during film condensation of vapor in a vertical bank of horizontal finned tubes: a) scheme of the process: 1) inflow of condensate; 2) finned cylinder: 3) flux of liquid flowing along the fin; 4) direction of the liquid flow over the side surface of the fin; 5) layer of liquid maintained constantly between fins; b) flow of 14% solution (by mass) of ethyl alcohol in water, T = 25°C, $\bar{a}$ = 2.8, $Re_*$ = 100; 1) region with a thin liquid film.

the fins does not depend on the irrigation density. The condensate flows only over the troughs between fins and over the fin faces. In the lower part of the finned cylinders the layer of liquid is maintained constantly between the fins, and its average height H increases on decrease of the parameter $\bar{a}$ [5], if $\bar{a} < 1$-0.

One can identify the following mechanisms for irrigation density affecting heat transfer.

1. An increase of irrigation density leads to a thickening of the film in the trough and a decrease of the effective fin height $h - \delta$. The change of film thickness in the trough with variation of irrigation density can be described by the Reynolds number, based on the condensate mass flow rate:

$$Re_* = \Gamma/2\mu. \tag{2}$$

2. It was established by special measurements in [2] that the height of capillary maintenance of liquid in a tube in a bundle is determined not only by the parameter $\bar{a}$, but it also depends on Re*. For $\bar{a}$ < 0.3 the height of the liquid layer maintaned between fins

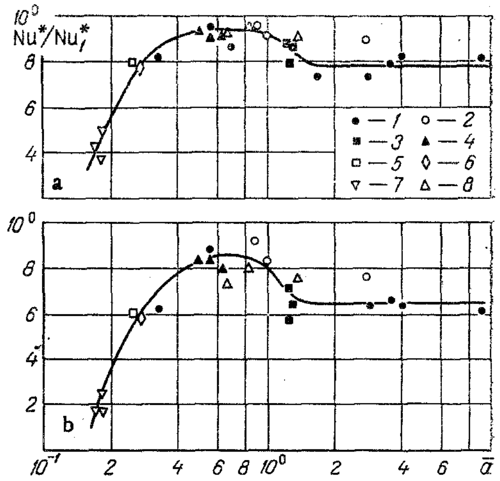


Fig. 3. Influence of irrigation density on heat transfer during condensation in a bundle as a function of the dimensionless distance between the fins: 1-4) data of the authors; 1) series 1, R12; 2) series 1, R11; 3) series 2, R12; 4) series 3, R12; 5) 512, T" = 40°C [9]; 6) R12, T" = 30°C [10], 7) R11, T" = 30-50°C [8]; 8) R12, T" = 42°C, n-butane, T" = 65°C, acetone, T" = 68°C [11]. a) $Re_*/Re_{*1} = 2$; b) 5.

increases appreciably with increase of $Re_*$, up to full saturation of the trough between fins over the entire cylinder perimeter. In tubes with $0.4 \leqslant \bar{a} \leqslant 1.0$ the height of the liquid layer, under the influence of the dynamic head of condensate flowing along the trough, decreases somewhat, compared to the height in an ordinary finned tube of the same geometry.

3. In cylinders with the parameter $\bar{a} > 1.0$ in the interfin trough the action of the capillary pressure gradient forms a region with a thin liquid film (see Fig. 2b). The influence of this pressure on the heat transfer was investigated in [6, 7], where a wire attached to the tube surface was used instead of a fin. The wire was wound on a horizontal tube in the form of a spiral with a constant distance between the coils. The action of surface tension forces caused thinning of the condensate film between the wires, and, as a result, the heat transfer intensified.

The observations that we took have shown that for an increase of $Re_*$ number there is contraction of the region with a thin film of liquid, right down to the point where the film vanishes.

Figure 3 shows a correlation of the results of our tests and those of other investigators [8-11] in the coordinates

$$Nu^*/Nu_1^* = f(\bar{a}) \tag{3}$$

for Re = idem and $Re_*/Re_{*1}$ = const. Here $Nu^*$ and $Nu_1^*$ are the Nusselt number in the i-th and the first tube of the bundle; $Re_*$ and $Re_{*1}$ are the Reynolds numbers in the trough in the i-th and the first tubes of the bundle. The parameter $Re_*/Re_{*1}$ describes the variation of film thickness in the trough over the depth of the bundle. Here we did not account for change of hydrodynamics of the liquid flow in the trough with increase of Reynolds number, i.e., possible variation of how the film thickness depends on the $Re_*$ number. However, for the condition Re = idem the film Reynolds number in the trough varies altogether by a factor of several, which allows us not to account for the above circumstance.

It can be seen that the relation obtained has a maximum. For $\bar{a} < 0.3$ the sharp drop in heat transfer is associated with flooding the interfin through with liquid. It should be noted that the data of experiments of different series show satisfactory agreement.

Analysis of the experimental data has shown that heat transfer during condensation of

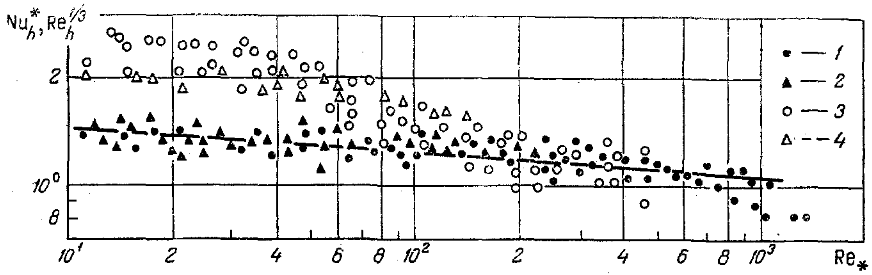


Fig. 4. Processing of the test data on heat transfer with condensation of vapor in bundles of finned tubes: the closed points: $0.4 \leqslant \bar{a} \leqslant 1.0$; 1) R12, T" = 40, 70°C; 2) R11, T" = 40, 70°C; open points: $2 \leqslant \bar{a} \leqslant 10$; 3) R12, T" = 40, 70°C; 4) R11, T" = 40°C.

vapor in a bundle depends on several basic parameters:

$$Nu^* = f(Re;\ Re_*;\ \bar{a}). \tag{4}$$

The functional relation (4) is valid for the condition $h \gg \delta$, which always holds in the experiments. It was noted above that for $\bar{a} < 0.3$ the fin area not flooded with liquid depends appreciably also on the irrigation density and on the dimensionless distance between fins. At present we do not know the laws for determining this area, which makes it difficult to correlate the data for $\bar{a} < 0.3$.

The correlation of the experimental data for the value $\bar{a} \geqslant 0.4$ of the parameter is shown in Fig. 4. It was postulated that the height of the liquid layer maintained between the fins in the lower part of the cylinder remained unchanged over the depth of the bundle. According to measurements taken with the help of a cathetometer, the maximum variation of the unflooded area of the fin resulting from a decrease of height H (see Fig. 2a) did not exceed 10%. It was also assumed that there was no heat transfer in the flooded zone.

The Nusselt and Reynolds numbers shown in Fig. 4, defined according to Eq. (1), were computed from the heat flux to the nonflooded fin area. Thus, Eq. (4) can be simplified:

$$Nu_h^* = f(Re_h;\ Re_*). \tag{5}$$

As was stressed above, the main part of the vertical surface of the fin of any tube of the bundle is practically not flooded with liquid flowing from tubes located above it. Because of the small linear size we can take the view, with sufficient accuracy, that in the vertical part of the fin the heat transfer is determined from a relation similar to the Nusselt relation, i.e.,

$$Nu_h^* \sim Re_h^{-1/3}. \tag{6}$$

As can be seen from Fig. 1, for the major part of our tests Eq. (6) holds quite satisfactorily.

Figure 4 shows the correlation of the experimental results in the coordinates of Eq. (5). It can be seen that many data coincide satisfactorily over the whole range of the governing parameters investigated, and can be described by the single empirical relation

$$Nu_h^* = 1{,}7 Re_h^{-1/3}\, Re_*^{0,07}. \tag{7}$$

The data of the tests of tubes with $\bar{a} > 1.0$ and $Re_* < 150$ are not described by the relation. As was noted earlier, in tubes with these finning parameters a region is formed with a thin film in the trough between fins, which leads to intensification of the heat transfer there, not accounted for in Eq. (7). For $Re_* > 150$ the results of all the experiments show interagreement.

With increase of irrigation density there is a change in the relation between the inertia forces of the condensate flow and the surface tension forces. The role of the latter decreases, as evidenced by the photographs of the liquid flow in the trough taken in [2]: it can be seen that the area with a thin liquid film is reduced. The correlation presented in Fig. 4 allows us to assert that for most irrigation densities it is inefficient to intensify

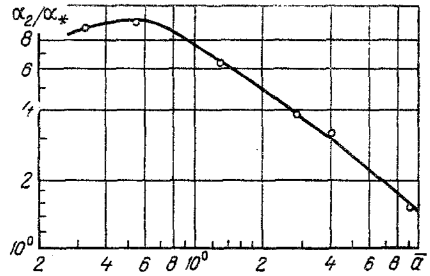


Fig. 5. Comparison of heat-transfer intensity during condensation of vapor in bundles of smooth and finned tubes, R12, $T'' = 40°C$, $Re_0 = 400$, $q_2$ = idem, $Re_0$ = idem.

heat transfer by means of wire finning [6, 7].

Figure 5 shows potential ways of increasing the heat-transfer coefficient in finned tube bundles. For an appropriate choice of finning geometry, the conventional heat-transfer coefficient in a finned tube bundle can be an order of magnitude greater than in bundles of smooth tubes. This is due to the following causes:

a) an increase of the heat-transfer surface due to finning;

b) a substantial increase of the characteristic linear size in part of the surface (in describing heat transfer in the vertical part of a finned tube this is the fin height, and for a smooth tube it is the diameter; the fin height is usually an order less than the tube diameter);

c) the influence of capillary forces acting on the film in such a way that the vertical surface of the fin (the basic heat-transfer surface) remains practically not flooded with condensate flowing from the tubes of the bundle located above. The weak influence of irrigation density is observed most clearly for bundles with optimal distance between fins.

From this investigation we can estimate the heat transfer during condensation of water vapor in a bundle of finned tubes. The data of the calculations show that it is favorable to use finning, since it leads to an appreciable intensification of heat transfer, especially for equipment with high heat transfer. Up till now we have known experimental studies [12-14] in which finning of the surface produced practically no intensification of the process of heat transfer with condensation of water vapor. The common defect of these investigations is the use of tubes with low value of the parameter $\bar{a}$ ($0.08 < \bar{a} < 0.15$), which caused appreciable flooding of the interfin through.

## NOTATION

$\lambda$, thermal conductivity of the liquid; $\nu$, $\mu$, coefficients of kinematic and dynamic viscosity; c, specific heat; r, heat of vaporization; $\sigma$, surface tension; $\rho'$, $\rho''$, densities of the liquid and the vapor; h, fin height; a, distance between the fins; b, fin thickness; $\varphi$, angle of inclination of the side surface of the fin; D, diameter of the tube to the troughs; $T''$, saturation temperature; q, $\alpha$, heat flux and heat-transfer coefficient on the fin side surface; $q_2$, $\alpha_2$, heat flux as calculated on the surface of a smooth tube on the fin base and the conventional heat-transfer coefficient based on this heat flux; $\alpha_*$, heat-transfer coefficient on a smooth tube of diameter D; H, average depth of flooding of the interfin trough; $\Gamma$, irrigation density of liquid in the trough; $\bar{\Gamma} = \Gamma_i/\Gamma_1$, ratio of the irrigation density in the i-th tube of length 1 m to the amount of liquid formed on the first tube per second at a constant conventional heat flux; $K = r/c\Delta T$, Kutateladze number; $\Delta T$, difference of temperatures of saturated vapor and cylinder wall; $\delta$, thickness of the liquid film in the interfil trough; $\bar{a} = a/\sqrt{\sigma/(\rho'-\rho'')g}$; $\bar{b} = b/\sqrt{\sigma/(\rho'-\rho'')g}$; $Re_0 = \Gamma_i/2\mu$, Reynolds number as calculated for a smooth tube and the fin base.

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# HEAT EXCHANGE AT THE BOTTOM OF A DEAD END BATHED BY A LAMINAR IMPINGING JET

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The local and average heat exchange of a round laminar jet flowing perpendicularly onto the center of the bottom of a dead-end chamber are investigated numerically.

The cathodes of plasmatrons and the elements of powerful electronic instruments are often cooled by a liquid which flows, in the form of a round laminar jet, coaxially into a cylindrical chamber and impinges perpendicularly on the flat bottom located opposite to the nozzle.

Certain aspects of the hydrodynamics and heat exchange of a round laminar jet in a dead end are considered in [1, 2]. The investigations were made by integrating the complete Navier–Stokes and energy equations for constant physical properties and incompressibility of the medium. In [1] only initial data are given on local heat exchange at the bottom of a dead end bathed by a jet with a rectangular initial velocity profile. From them one can judge that the degree of constriction b of the stream vitally affects both the intensity of heat exchange and the character of its distribution. For relatively small b, in particular, there are regions with enhanced heat exchange in the corners of the dead end, which does not occur when a jet impinges on a barrier. The influence of the determining parameters on the dispersion of the

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