FILM CONDENSATION OF A STATIONARY VAPOR ON A FINNED SURFACE

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Experimental data are presented on heat exchange and capillary retention of liquid during vapor condensation on horizontal finned cylinders in a wide range of variation of the geometrical and hydrodynamic characteristics.

Film-type vapor condensers are an inseparable part of thermoelectric stations, refrigerating equipment, chemical engineering apparatus, and water-distilling devices. The most widespread means of intensification of heat exchange during film condensation of liquids is finning of the heat-exchange surface.

At present there are a considerable number of papers devoted to the study of heat exchange during film condensation of vapor onto single horizontal finned cylinders [1-7]. However, they were carried out in a rather narrow range of the parameters determining heat exchange. Most of the experiments described in the literature were made on semiindustrial apparatus, with a low experimental accuracy, and without a control on the purity of the working substance. In a number of papers the heat exchange was determined through the heat-transfer coefficient. There are practically no theoretical papers devoted to the investigation of heat exchange during film condensation on a horizontal finned cylinder.

The task of the investigation was to clarify the main laws of heat exchange during film condensation as a function of the hydrodynamics of liquid flow and the geometrical parameters of the finning. A description of the test bench and the procedure for running the tests is presented in [8, 9]. We used Freon-12 and -21 as the working liquids. In the tests we determined the dependence of the average heat flux on the temperature difference between the saturated vapor and the average temperature of the wall. The heat flux was calculated from the change in the enthalpy of the cooling water. The wall temperature of the experimental sections was measured with thermocouples. The measurements of the main quantities on which heat exchange depends were doubled. The electrical signals from the outputs of all the sensors were sent to a measurement-calculation complex based on an Élektronika-60 microcomputer. The process of data collection and processing was fully automated. At values of the vapor—wall thermal head of more than 2°C the experimental error did not exceed 10%. In the process of running the tests on heat exchange it could be observed visually that capillary retention of liquid in the troughs between fins occurs for individual finning parameters. This phenomenon was first noted in [1]. The authors of [1] concluded that the liquid retained between fins has no appreciable influence on heat exchange. Experiments on the capillary retention of liquid were subsequently made in [10]. At present there are no data in the literature on the height of capillary retention of liquid which were obtained directly during vapor condensation.

In our experiments the height of the liquid layer retained on a finned cylinder was measured simultaneously with the measurement of heat exchange during vapor condensation. The height of capillary retention of the liquid was also measured on a specially built test bed simulating the process of liquid flow over a finned surface. The tests were run at atmospheric pressure under isothermal conditions. Distilled water or an alcohol—water mixture was used as the working liquid. The height of capillary retention of the liquid was measured with a cathetometer. In Fig. 1a, b we present photographs of capillary retention of liquid in the troughs on finned cylinders with different distances between the fins. In Fig. 1c the data obtained on finned cylinders with a rectangular fin profile are treated in the form of the dependence of the average dimensionless immersion height on the inverse dimensionless distance between fins. The value of the capillary constant of the liquid was used as the linear scale. It is seen from the figure that the experimental data obtained in the condensation

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Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 49, No. 5, pp. 709-717, November, 1985. Original article submitted October 17, 1984.

0022-0841/85/4905-1247$09.50

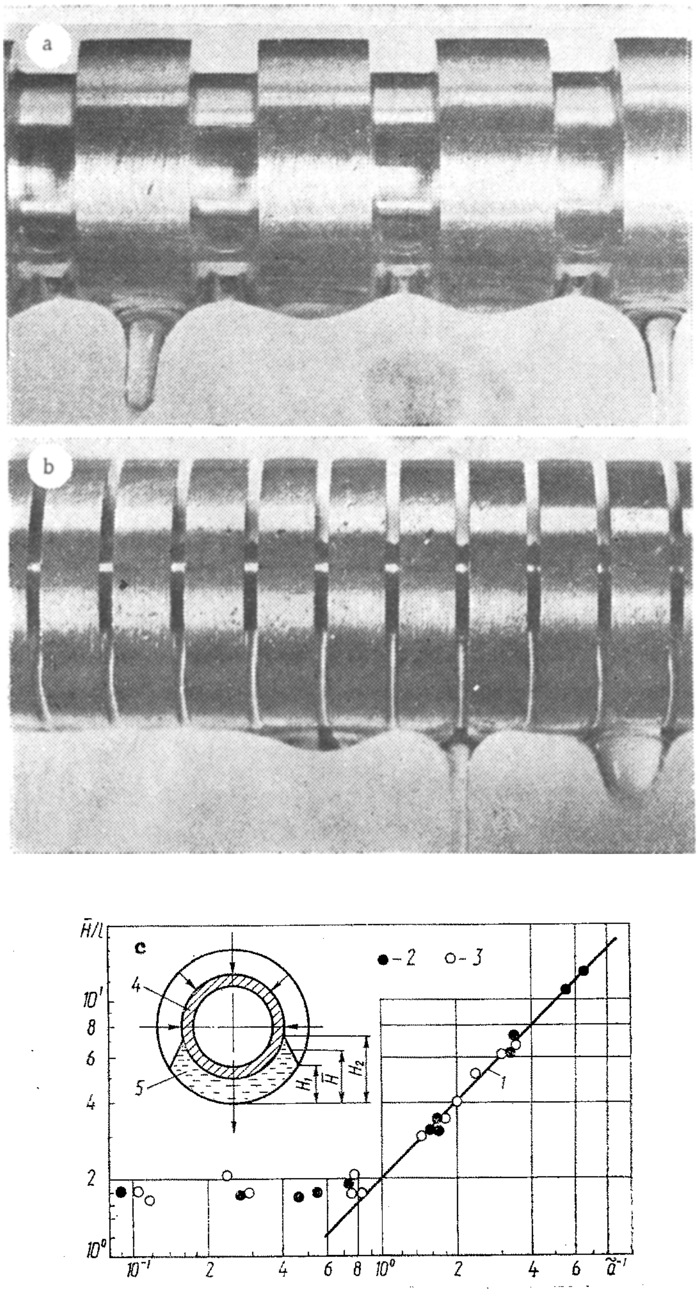

Fig. 1. Capillary retention of liquid on a horizontal finned cylinder: a) water, T = 25°C, $\tilde{a}$ = 2.21, Re_0 = 50; b) 14% solution (by weight) of ethyl alcohol in water, T = 25°C, $\tilde{a}$ = 0.65, Re_0 = 58; c) 1) calculation from (1); 2) water, T = 25 °C; 3) R12, T'' = 20, 40, 70°C; 4) cylinder; 5) liquid layer retained between fins.

of Freon-12 vapor and in the simulation of the condensation process by sprinkling are in satisfactory agreement with each other. For $\tilde{a} \leq 1$ the height of the liquid layer retained in the cylinder matches well with the height of the capillary rise of liquid in an infinite vertical slot with a width equal to the distance between fins:

$$H_0 = 2\sigma / (\rho' - \rho'')\, ga. \qquad (1)$$

For $\tilde{a} > 1$, the height of the liquid layer retained on the cylinder does not depend on the distance between fins, and remains constant. It is seen from the figure that there is a clear transition from one region of the dependence to the other.

A similar dependence was obtained for the height of capillary retention of liquid on cylinders with a trapezoidal fin profile. As the characteristic linear size in a trough we used the equivalent distance a_* between fins, introduced in [11] for the generalization of experimental data on the height of capillary retention in vertical channels:

$$a_* = (a\cos\varphi + h\sin\varphi)/(1 - \sin\varphi).$$

It should be mentioned that such data, obtained on finned cylinders with rectangular and trapezoidal fin profiles, are absent from the literature.

In the experiments on heat exchange we used cylinders with isothermal fins. To simplify the problem, most of the tests were run on cylinders with a rectangular fin profile.

If we assign the condition of isothermicity for the fins, then the main unknown parameters of the finning are the width of a fin and the distance between fins.

The experiments on heat exchange were made on 15 finned cylinders at different saturation temperatures. The heat flux was varied by supplying cooling water having a variable temperature. The parameters of the finning are presented in Table 1. Three series of investigations were made. In the first, the only geometrical dimension varied was the distance between fins. The parameter $\tilde{a}$ was varied in the limits of from 0.3 to 9.2. In the second series of tests the fin width was varied by an order of magnitude. Heat exchange on four cylinders having a trapezoidal fin profile was investigated in the third series. The results of these experiments were partially published in tabular form in [12].

A horizontal finned pipe can be treated as consisting of three different heat-exchange surfaces: the vertical surface and two horizontal ones, the face and the trough. The heat fluxes at each of these surfaces can differ considerably from each other. For generalization of the experimental data, therefore, it is incorrect to use the averaged heat flux calculated over the total surface of a finned pipe. The total amount of heat is measured in the experiment, however.

An important property of finned cylinders, which has been used up to now by various investigators to study heat exchange and is now being used in industry, is the fact that the lateral surfaces of the fins on them comprise, as a rule, 70% of the total outside surface of the pipe. The height of a fin is usually an order of magnitude less than the pipe diameter. Therefore, there is reason to assume that the overwhelming share of the heat is removed from the lateral surfaces of the fins. Our experiments were analyzed under the following assumptions. Heat exchange on the horizontal sections of a finned pipe is determined through the same functions as for condensation on a smooth horizontal pipe at the same Reynolds number for the film. It was assumed that all the rest of the heat is transferred through the lateral fin surface. From the heat flux thus defined we calculated the numbers Nu* and Re for the film on the lateral fin surface. Here

$$\mathrm{Nu}^* = \frac{\alpha}{\lambda}\left(\frac{\nu^2}{g(1-\rho''/\rho')}\right)^{1/3}, \tag{2}$$

$$\mathrm{Re} = qh/(\mu r(1 + 3/8\,\mathrm{K})\cos\varphi. \tag{3}$$

TABLE 1. Geometrical Characteristics of Finned Pipes

No.	No. of test series	D, mm	h, mm	a, mm	δ, mm	φ, deg	ε	F_1, %	F_2, %
1	I	19,78	1,03	0,25	0,50	0	3,98	73	8
2		19,90	1,01	0,41	0,52	0	3,36	68	13
3		19,86	1,09	1,00	0,50	0	2,58	60	26
4		19,75	1,01	2,14	0,46	0	1,84	45	45
5		19,87	1,05	3,04	0,47	0	1,65	38	53
6		19,89	1,08	6,94	0,56	0	1,31	23	70
7	II	19,90	1,04	0,95	0,30	0	2,79	63	27
8		19,80	1,00	0,95	0,40	0	2,6	60	27
9		19,89	1,06	1,00	0,75	0	2,33	55	24
10		19,87	1,05	1,02	1,18	0	2,07	49	22
11		19,90	1,04	0,97	2,24	0	1,77	39	17
12	III	21,50	2,5	0,37	0,56	12,1	3,49	84	5
13		20,30	1,6	0,51	0,54	15,2	2,49	76	11
14		20,60	1,41	0,46	0,47	20,8	2,13	76	11
15		16,74	1,23	0,42	0,44	10,6	2,77	74	12

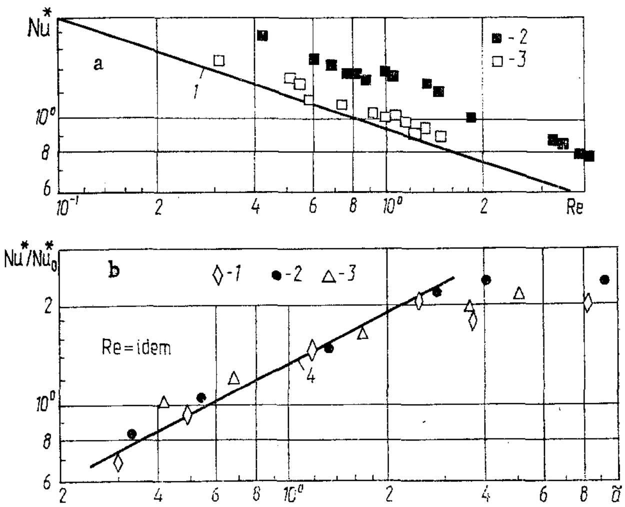

Fig. 2. Dependence of heat exchange on the dimensionless distance between fins: a) pipe 13: 1) calculation from (4); 2) R12, T" = 60°C, $\tilde{a}$ = 0.78; 3) R21, T" = 60°C, $\tilde{a}$ = 0.49; b) pipes 1-6, R12: 1) T" = 20°C; 2) 40°C; 3) 70°C; 4) $Nu^*/Nu_0^* \sim \tilde{a}^{\frac{1}{2}}$.

This way of writing the Reynolds number assumes that the fin height is comparable with the capillary constant of the liquid, and the condensate flows into a trough from the lateral surface under the action of capillary forces (see Fig. 1c).

Estimates show that the assumptions adopted are fully admissible, since for the above-indicated pipes the calculated contribution of the faces and troughs to the total amount of heat removed from the pipe surface does not exceed 15%.

From an analysis of the experimental results in the coordinates (2) and (3) it follows that the line generalizing the experimental data has a slope close to the theoretical Nusselt function for the case of condensation on a vertical plate [13],

$$Nu_0^* = 0{,}925\,Re^{-1/3}. \tag{4}$$

For any of the cylinders for which $\tilde{a} < 1$ the experimental data are not described by the single function (4). As was shown in [9], besides the Reynolds number, the heat-exchange intensity depends on the saturation temperature. The experimental data on heat exchange during the condensation of the two refrigerants onto the same cylinder also are not generalized by the function (4). They are shown in Fig. 2a. The results corresponding to smaller values of the dimensionless distance between fins, i.e., to greater immersion of the troughs between fins in liquid, lie lower. For finned cylinders with $\tilde{a} > 1$ there is no clear layering of the experimental data with respect to the saturation temperature.

As indicated earlier, the height of capillary retention of liquid on a finned cylinder with $\tilde{a} > 1$ remains practically constant, i.e., a qualitative connection between the height of capillary retention and the heat-exchange intensity is clearly missed.

In Fig. 2b the experimental data obtained in the condensation of refrigerant 12 onto cylinders with different distances between fins are represented in the form of the dependence of the relative Nusselt number on the parameter $\tilde{a}$ for the same value of Re of the film on the lateral fin surface. It is seen from the figure that in the region of $0.3 < \tilde{a} < 2$ the relative Nusselt number grows with an increase in $\tilde{a}$. At $\tilde{a} > 2$, the relative Nusselt number remains practically constant. For values of the dimensionless distance between fins of less than 0.5, at which capillary retention of liquid in the troughs becomes significant, the relative Nusselt number is less than one.

If we assume that heat exchange is absent in the immersion zone, and we construct the relative variation of the Nusselt number with the parameter $\tilde{a}$, then in the region of $0.3 < \tilde{a} < 3$ the function has the form $Nu^*/Nu_0^* \sim \tilde{a}^{\frac{1}{4}}$ [9]. In this region of values of the parameter $\tilde{a}$ the intensification of heat exchange on a finned cylinder is evidently connected with a decrease in the thickness of the liquid film in the troughs between fins with an increase in the dimensionless distance between fins.

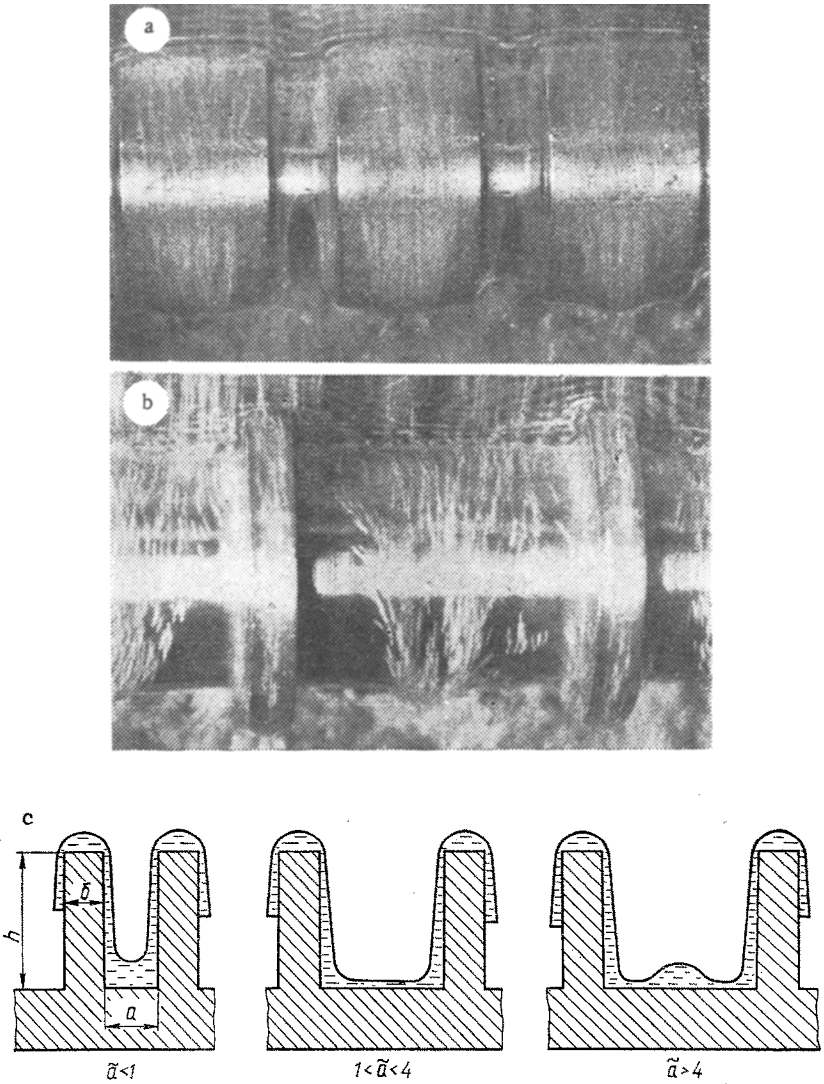

Fig. 3. Principal regimes of liquid flow in a trough between fins [14% solution (by weight) of ethyl alcohol in water, T = 25°C]: a) $\tilde{a}$ = 2.3, Re_0 = 350; b) $\tilde{a}$ = 9, Re_0 = 353; c) schematic depiction.

A basis for the foregoing may be provided by photographs obtained of the process of liquid flow in the troughs between fins, where we were able to make visible the streamlines of the draining liquid (Fig. 3a, b). The finned cylinders are sprinkled with a continuous liquid film. On the basis of an analysis of the photographs we can distinguish three characteristic regimes of liquid flow in a trough as a function of the distance between fins and the Reynolds number of the film.

Profiles of the liquid film in a trough between fins are shown schematically in Fig. 3c. In the troughs on finned cylinders with $\tilde{a} < 1$ the liquid profiles have the shape of a semicircle, while the velocity profile over the channel width has the form of a parabola. For experimental sections with $1 < \tilde{a} < 4$ and $Re_0 < 400$, the liquid, after traveling a certain way along the trough, separates into two streams under the action of capillary forces and flows down along the fins. As a result, a region with a thin liquid film is formed at the center of the trough. The position of the point of separation of the film in the channel is a function of the parameters $\tilde{a}$ and Re_0. For finned cylinders with $\tilde{a} > 4$ and $Re_0 > 80$ it is clearly seen that the liquid has the highest flow velocity in the middle part of a trough. The film becomes thicker at this point. A region with a thin liquid film is formed between the central flow in the trough and a fin, along which a liquid stream also flows down. Similar patterns of liquid flow were recorded in [14], where the liquid flowed down in a rectangular vertical channel, the width of which was varied.

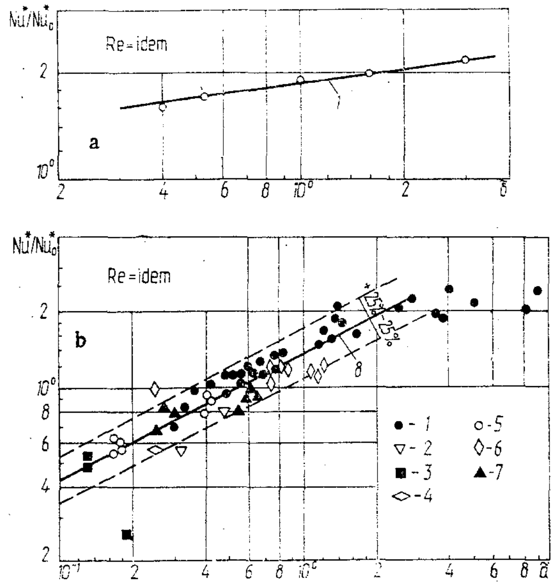

Fig. 4. Film condensation of stationary vapor onto horizontal finned cylinders: a) influence of the width of the fin face on heat exchange, R12, T" = 40°C; 1) $Nu^*/Nu_0^* \sim \tilde{\delta}^{1/8}$; b) generalization of experimental data on heat exchange: 1) R12 and R21, T" = 20, 30, 40, 60, 70°C, authors' data; 2) data of [3]; 3) [2]; 4) [5]; 5) [4]; 6) [6]; 7) [7]; 8) $Nu^*/Nu_0^* \sim \tilde{a}^{\frac{1}{2}}$.

In regimes characterized by considerable nonuniformity of the film thickness in a trough between fins, the heat exchange during condensation can be intensified in comparison with that for a horizontal smooth pipe. The experiments made in [15] can serve as confirmation of this. In them the intensification of heat exchange was observed in the condensation of steam on cylinders with fins in the form of wires not touching the surface of the pipe. In these experiments the parameter $\tilde{a}$ was varied in the range $1 < \tilde{a} < 5$.

It must be noted that the possible intensification of heat exchange in a trough is not taken into account in the calculation of the numbers Nu* and Re by the method proposed in the present article. The treatment of our data in these coordinates is evidently most reliable up to values of $\tilde{a} \leqq 2$, where the vertical surface of the fins comprises more than 60% of the outside surface of a finned pipe. Representing the experimental data in the form of the dependence of the relative Nusselt number on $\tilde{a}$ enables us to determine the value of the dimensionless distance between fins at which the intensification of heat exchange with its decrease ceases. According to our experimental data, this distance is approximately $\tilde{a} \cong 2$.

In Fig. 4a we present experimental data on finned cylinders for which a single geometrical parameter was varied — the fin width.

The comparison was made for a constant Reynolds number of the film on the lateral fin surface. The influence of the width of the fin face on the heat exchange proved to be rather weak. With variation of $\tilde{\delta}$ by almost an order of magnitude the variation of the heat exchange does not exceed 30%. Smaller values of $\tilde{\delta}$ correspond to lower values of the Nusselt number for the same value of Re of the film on the lateral fin surface. In the investigated region the relative Nusselt number is approximately proportional to $\tilde{\delta}^{1/8}$.

The experimental data obtained in the present work in the condensation of Freon-12 and -21 on cylinders with rectangular and trapezoidal fin profiles are generalized in Fig. 4b in the form of the dependence of the relative Nusselt number on the dimensionless distance between fins for the same Reynolds number of the film on the lateral fin surface. Here we also present the known experimental data from the literature on heat exchange during the condensation of five different Freons at different temperatures on 15 cylinders of different geometries. It can be noted that practically all the experiments of other investigators were made for $\tilde{a} < 1$, i.e., in the region where capillary forces considerably lower the heat-exchange intensity. The experimental data in the region of $0.1 < \tilde{a} < 3$ can be described with an accuracy of ±25% by the empirical function

$$Nu^*/Nu_0^* = 1{,}35\tilde{a}^{1/2}. \quad (5)$$

In the treatment of the data presented in Fig. 4b the influence of the dimensionless parameter $\tilde{\delta}$ on the heat exchange was not taken into account, although its variation reached a factor of 30. The satisfactory generalization of the results of different authors confirms that the thickness of the fin face has a weak influence on heat exchange. The dimensionless fin height $\tilde{h}$ was varied in the range $0.9 < \tilde{h} < 3.5$. One of the parameters determining heat exchange during condensation on finned cylinder is the dimensionless distance between fins.

From the data presented we can conclude that the total amount of heat transferred through a finned pipe is determined, on the one hand, by the area of the pipe, which increases with a decrease in the distance between fins and, on the other hand, by the heat-exchange intensity, which decreases for $\tilde{a} < 2$. Therefore, the law of heat transfer through a finned pipe will have a maximum at a certain value of $\tilde{a}$. For a given fin geometry the maximum of this function is determined only by the parameter $\tilde{a}$. The fin geometry is calculated from the classical heat-transfer functions, presented in [16], for example.

NOTATION

λ, ν, μ, c_p, r, σ, ρ', ρ'', coefficients of thermal conductivity of kinematic and dynamic viscosity, specific heat, heat of vaporization, surface tension, and densities of liquid and vapor, respectively; D, outside diameter of cylinder; h, δ, fin height and thickness at the face; a, distance between fins at the base; φ, angle of inclination of the lateral fin surface; ε, finning coefficient with respect to the surface of a smooth pipe with the diameter at the base of the fins; F_1, F_2, parts of the outer area of a finned cylinder belonging to the vertical surface and to the trough surface, respectively; T, temperature; T", saturation temperature; α, q, heat-transfer coefficient and heat flux at the lateral fin surface; Γ, sprinkling density; H_1, H_2, heights of total and maximum immersion of the lateral fin surface; ΔT, temperature difference between the saturated vapor and the cylinder wall; $\ell = \sqrt{\sigma/(\rho' - \rho'')g}$, capillary constant of the liquid; $K = r/c_p\Delta T$, Kutateladze number; $\bar{H} = (H_1 + H_2)/2$; $Re_0 = \Gamma/\mu$; $\tilde{a} = a/\ell$; $\tilde{\delta} = \delta/\ell$; $\tilde{h} = h/\ell$.

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CALCULATION OF HEAT TRANSFER DURING WATER FLOW IN PROFILED TWISTED PIPES

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UDC 536.242

A Semiempirical model of heat exchange during water flow in profiled twisted pipes is formulated on the basis of the modified Prandtl–Taylor analogy.

It is well known [1-7] that a rigorous analytic investigation of hydrodynamics and heat exchange in pipes and channels with artificial roughness is practically impossible, which is determined by the extremely complicated flow structure. This situation also pertains fully to profiled twisted pipes (PTP), where the interaction of axial, swirled, and separation flows is observed. The relations between the intensities of each of these flows, the boundaries of which are practically impossible to predict because of their mutual overlapping, are evidently determined by the geometry of the PTP and by the flow regime of the axial stream. It must be emphasized that near the pipe wall the flow is three-dimensional: The stream has velocity components along the knurling (resulting in friction of the stream against the pipe wall) and perpendicular to the projections (resulting in the loss of mechanical energy in the developing vortices), as well as a radial velocity component. The relations between the velocity components of these flows, and hence the share of friction and local resistances in the overall energy dissipation, can be estimated only by modeling. From the data of a number of papers [3, 4, 6] it has been established that right at the projections (in a zone of artificial roughness) the flow has a cellular character – horseshoe-shaped vortices are formed, the dynamics of which depend essentially on the shape and size of the roughness. Moreover, as noted in [8], the flow region in the zone above the projections is filled with vortices of different scales. Under these conditions, the construction of a calculating model of the flow requires a certain schematization.

On the basis of the concepts presented above, with allowance for the results of investigations of the hydrodynamics of water flow in glass PTP, a semiempirical model of the process of heat exchange during the flow of a one-phase heat-transfer agent in a PTP is based on the modified Prandtl–Taylor analogy. The main idea of this analogy consists in the summing of the thermal resistances of the different regions through which the heat flow occurs [8]. For a two-layer Prandtl–Taylor model the total thermal resistance consists of the sum of the thermal resistances of the regions of turbulent (R_t) and molecular (R_m) transfer:

$$R = R_t + R_m. \quad (1)$$

The following equation was obtained in [8] for moderate Prandtl numbers, using the generalized Reynolds analogy for the thermal resistance in a fully turbulent region:

S.M. Kirov Ural Polytechnic Institute, Sverdlovsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 49, No. 5, pp. 718-723, November, 1985. Original article submitted September 24, 1984.

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