average temperature of the liquid over the transverse cross section of the film; H, density of the heat flow on the wall; $\xi = (x/\delta)Pe$, $\eta = y/\delta$, dimensionless coordinates; δ, thickness of the film; $Pe = u_0\delta/a$, Peclet criterion; u_0, velocity on the free surface; λ, a, thermal conductivity and thermal diffusivity of the liquid; $q(\xi)$, thickness of the thermal boundary layer; n_1, n_2, parameters of the temperature profile in the first and second stages of the process; $Nu = \alpha\delta/\lambda$, $\langle Nu \rangle = \langle \alpha \rangle \delta/\lambda$, local and average Nusselt numbers; α, $\langle \alpha \rangle$, local and average heat-transfer coefficients.

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REFINED METHOD OF CALCULATING HEAT EXCHANGE IN THE CONDENSATION OF STATIONARY STEAM ON FINNED HORIZONTAL TUBES

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A relation is obtained for calculating the heat-transfer rate in the condensation of pure vapors on finned tubes with allowance for fin efficiency.

The study of the heat-exchange laws in the condensation of vapors of liquids is directed toward solving important practical problems of finding surfaces on which heat- and mass-transfer processes can occur efficiently. A certain amount of experimental material has already been accumulated in this direction, including data on heat exchange in the condensation of vapors of Freons 11, 12, 22, and 113, and water on single horizontal tubes with transverse finning. These tubes have been made of different materials and have had fins of various geometries. The main data on the test conditions and geometrical characteristics of these tubes, including the findings in [1, 2], are presented in Table 1. These studies have proposed relations in the form of criterial equations, including equations of the Nusselt type for smooth tubes, with the introduction of constant coefficients to account for the specifics of heat exchange on the fins. Analysis of these relations shows that they do not allow for generalization of the data of various authors which is in the literature. This is due to the complexity of the process of condensation on finned tubes. In contrast to the same process on smooth tubes, here condensation is determined by several new factors: the geometry of the surface, the heat conductivity of the wall material, and, as noted in certain investigations, the effect of surface tension.

Evidently, the only way out of this dilemma is to obtain theoretical solutions and refine these solutions on the basis of experimental data.

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TABLE 1. Geometric Characteristics of Finned Tubes

Serial No.	Tube diam. along base of fin, mm	Fin dimensions, mm: spacing	Fin dimensions, mm: height	Fin dimensions, mm: thickness, base	Fin dimensions, mm: thickness, at end	Fin opening angle, deg	Degree of finning	Material of tube with fin	Working substance	Weber number	Source in literature
1	16,0	2,06	2,2	1,81	0,47	16,9	3,0	Copper	Freon-12	1,1	
2	12,0	5,16	2,2	0,3	0,3	—	4,5	Brass	«	0,8	
3	20,0	4,0	14,75	0,5	0,5	—	14,0	«	«	0,2	[6]
4	17,7	2,03	1,9	1,6	0,38	—	2,76	Copper	Freon-22	—	[7]
5	23,0	2,0	1,5	1,6	0,5	—	2,29	Steel	«	—	
6	9,46	1,58	1,42	0,533	0,305	4,5	3,0	Copper	Freon-12	2,9	
7	19,1	1,58	1,32	0,38	0,38	—	2,8	«	«	2,7	
8	25,4	1,6	1,35	0,431	0,229	4,3	2,6	«	«	4,1	[9, 10]
9	19,7	3,7	8,3	0,814	0,431	1,3	9,6	«	«	0,4	
10	19,7	4,35	8,25	0,99	0,56	1,5	6,4	«	«	0,4	
11	19,6	6,62	8,3	1,14	0,458	2,4	5,51	Copper	Freon-12	0,3	
12	15,9	1,64	1,47	0,585	0,33	4,9	2,9	Copper	Freon-22	2,5	
13	15,8	1,76	1,6	0,33	0,33	—	3,0	«	«	2,5	[11]
14	19,5	3,67	8,66	0,94	0,36	2,0	7,7	«	«	0,5	
15	19,5	3,71	3,46	0,94	0,736	1,7	3,2	«	«	0,5	
16	19,2	3,89	7,4	1,04	0,406	2,5	6,2	«	«	0,4	
17	19,0	1,0	1,38	0,4	0,45	—	3,8	Copper	Freon-12	2,2	
18	20,05	1,5	2,0	0,5	0,5	—	4,0	«	«	1,4	[12]
19	19,8	1,7	3,63	0,5	0,5	—	6,2	Copper	Freon-12	0,70	
20	20,0	1,65	4,5	0,5	0,5	—	7,7	«	«	0,6	
21	16,0	2,0	2,0	1,8	0,5	18	2,8	Copper	Freon-12,22	1,2	
22	16,0	2,0	2,2	1,15	1,14	—	3,6	«	«	0,5	
23	18,6	2,03	1,2	0,43	0,78	15,2	2,0	«	«	1,1	[8]
24	16,0	1,07	2,2	0,4	0,4	—	5,8	«	«	1,5	
25	17,0	2,0	2,2	1,67	0,43	15,7	3,0	Copper	Freon-21	2,0	
26	13,0	1,25	1,5	0,75	0,33	8,0	3,4	«	«	4,3	
27	15,5	0,75	1,4	0,55	0,22	6,7	4,6	«	«	7,0	[4]
28	18,1	0,81	0,923	0,67	0,14	16,0	2,8	Brass	Freon-113 water	70/14 60/12	
29	18,1	1,28	0,923	1,14	0,14	28,5		«	«	42,5/7,5	
30	18,1	0,8	0,923	0,52	0,28	7,4	3,2	«	«	7,0/1,5	[1,3]
31	17,0	2,0	2,0	1,8	0,64	16,5	2,8	Copper	«	50/9	
32	17,4	0,81	1,32	0,67	0,14	11,4	3,9	Brass	«	54,5/9,5	

In [1], empirical data on the heat transfer of Freon-113 vapors on small-finned tubes was generalized in the traditional form, with the introduction of a Weber number into the equation to account for the effect of surface tension. Later, when experimental data was obtained in [3] for the same type of specimens for heat transfer in the condensation of steam, it turned out that the results obtained for water and Freon-113 could not be generalized together by any of the known relations. In connection with this, a partial theoretical solution was obtained in [3] for the heat-exchange problem in the condensation of stationary stream on a horizontal, small-finned tube under conditions of the predominating effect of surface tension. This partial solution was then used as the basis for a new form of generalization

$$z = f(H,\ \mathrm{m}), \tag{1}$$

where $z = \Delta/h$ is the relative filling of the interfin grooves;

$$H = \frac{\sigma^{0.25}\mu^{0.75}\lambda^{0.75} D_0 \overline{\vartheta}^{0.75}}{\rho^{1.75} r^{0.75} b^{0.25} h^{3.5} \sin^3\varphi (1 + \operatorname{tg}\varphi)^{0.25}},\quad m = \frac{a}{h \operatorname{tg}\varphi}.$$

The proposed method makes it possible to generalize the data for water and Freon-113 on all investigated models of small-finned tubes. Here, the need was emphasized to consider the nonisothermal character of the fin over its height.

However, analysis of test data from [4-12] (see Table 1) shows that the above method has only a limited application. As a detailed analysis showed, this is the result of two basic factors.

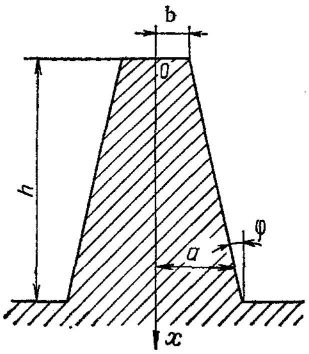

Fig. 1

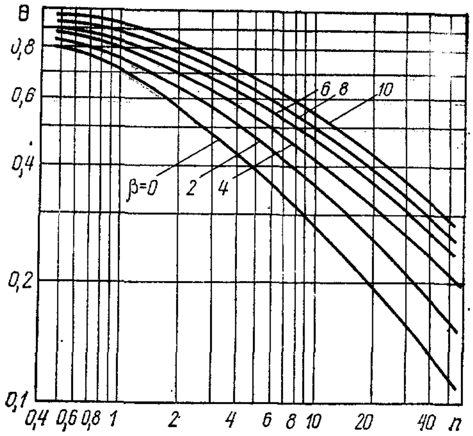

Fig. 2

Fig. 1. Model of condensate film flow over a trapezoidal fin.

Fig. 2. Laws of change in the mean dimensionless temperature head on a trapezoidal fin in the form $\bar{\Theta} = f(n, \beta)$.

First, solution of the heat-exchange problem at We ≥10 corresponds to the case — investigated only in our experiments [1, 3] — of tubes with small interfin spaces (grooves). In this case, a large portion of these grooves is inundated with condensate, and the main resistance offered to the movement of the condensate about the circumference of the tube is associated with friction against the side walls of the fin. Second, the method used to calculate the so-called efficiency of the fin (i.e., to account for its nonisothermal character) is based on the solution of the heat-conductivity equation of a fin of constant thickness. As a result, this method (by which the results of the experimental studies [1, 3] were generalized) proves to be unsuitable for generalizing most other investigations also conducted at least in part for conditions of a certain (We < 10) effect of surface tension but, in addition, generally involving finning with fairly broad interfin grooves and trapezoidal fins.

In the present work, we again develop a theoretical solution to the problem under conditions whereby surface tension (We ≥10) affects condensation. However, we now take the above considerations into account, so that we come up with a refined method which generalizes most available empirical data and is close in form to the Nusselt equation. Here, the general principle underlying the solution to the problem remains the same, as does the mechanism by which the condensate film is distributed over the horizontal finned tube at We ≥10. As in [2], the condensate is drawn from a fin of thickness

$$\delta = \left[\frac{4\mu\lambda b x \bar{\vartheta}(1 + \operatorname{tg}\varphi)(h - \Delta)}{\rho r \sigma \cos\varphi} \right] \tag{2}$$

at a velocity

$$\bar{W}_x = \frac{\sigma\delta^2 \cos\varphi}{3b\mu(h - \Delta)(1 + \operatorname{tg}\varphi)} \tag{3}$$

by surface tension into the interfin groove about the entire circumference of the horizontal tube, except for the bottom zone, taken equal to $\bar{\psi} = \pm 150°$ (Fig. 1).

As in the case of a flat wall, in deriving a differential equation of laminar flow of the condensate in the interfin groove under the influence of the force of gravity in the case of negligible flooding of the groove (i.e., in the case of a groove of sufficient width), we may restrict ourselves to examining a semihyperbolic law of velocity distribution according to film thickness Δ, and friction of the film against the side walls of the fin may be disregarded. Here the mean rate of flow of the condensate along the groove at a certain point ψ is determined by the newer (compared to the equation in [3]) expression

$$\bar{W}_\psi = \frac{h^2 \rho \sin\psi}{3\mu} z^2. \tag{4}$$

Now, having determined the flow rate of the condensate along the groove (on 1/2 its width):

$$G_\psi = \rho \overline{W}_\psi F_q = \frac{a\rho^2 h^3 \sin\psi}{3\mu\cos\varphi} z^3, \tag{5}$$

where

$$F_q = \frac{a\Delta}{\cos\varphi} = \frac{ahz}{\cos\varphi},$$

and, as in [3], connecting its change only with the flow of condensate from the fin surface (condensation of vapor in the groove itself is not considered here), we obtain the differential equation

$$\frac{dz}{d\psi} = 0.47\,\mathrm{Fi}\,\frac{\sqrt{1-z}}{z^2\sin\psi} - \frac{z}{3}\,\mathrm{ctg}\,\psi, \tag{6}$$

where

$$\mathrm{Fi} = \frac{\sigma^{0.25}\lambda^{0.75}\mu^{0.75}D_0\bar{\vartheta}^{0.75}\bar{\Theta}^{0.75}\cos^{0.75}\varphi}{ab^{0.25}r^{0.75}h^{2.5}\rho^{1.75}(1+\mathrm{tg}\,\varphi)^{0.25}},$$

which describes the rate of flow relative to the flooding of the interfin groove with condensate about the tube circumference.

In accordance with the form of the derived differential equation (6), test data for horizontal finned tubes at We ≥10 may be generalized by the simpler (compared to [3]) relation

$$z = f(\mathrm{Fi}). \tag{7}$$

By analogy with [3], we may take for the value of the relative flooding of the interfin groove in Eq. (7) the value at the boundary with the bottom layer ($\bar{\psi} = 150°$). Here, the total flow rate along half of the groove is determined from Eq. (5) by the formula

$$G_g = \frac{a\rho^2 h^3}{6\mu\cos\varphi} z^3. \tag{8}$$

As in the previous analysis, test data should be generalized according to Eq. (8) together with estimates of fin efficiency $\bar{\Theta}$, which enters directly into the expression for Fi.

Since the calculational methods in [3] pertain only to fins of constant thickness, we examined existing methods [2, 13] of determining the efficiency of fins of variable cross section (Fig. 1). The method in [2] pertaining to the process of heat transfer on finned surfaces is most acceptable, in our opinion. The results of these studies are shown in Fig. 2 in the coordinates $\bar{\Theta} = f(n, \beta)$, where

$$n = \frac{\rho r\sigma\lambda^3 h^6}{4\mu b^5 \lambda_p^4 \vartheta_0 (1+\mathrm{tg}\,\varphi)}, \quad \beta = \frac{h\,\mathrm{tg}\,\varphi}{b}. \tag{9}$$

Using Eq. (7) and the theoretical relations in Fig. 2 on fin efficiency, we analyzed all of the test data in Table 1. This analysis showed that, in this form, a large group of data may be generalized with respect to the degree of the effect of surface tension, embracing a region We ≥1. This is significantly broader than the theoretically substantiated region We ≥ 10.

The results of experimental studies of heat exchange during the condensation of vapors of different Freons [4-12] and water [3] on single horizontal tubes of different geometries and fin materials may be generalized with an error of ±15% for the case We ≥ 1 by means of the relation

$$z = 2.0\,\mathrm{Fi}^{1/3}. \tag{10}$$

Analysis of the empirical scatter shows that certain results — those obtained on tubes with a fairly thick fin (b/h > 0.1) [6-8] — exhibit the greatest deviation. This is explained by the fact that the theoretical solution (9) does not consider vapor condensation on the end. However, this factor may be accounted for in Eq. (10) by conditionally increasing the height of the fin (entering into the expression for Fi) by half the thickness of the fin on the end (i.e., $h \to h + b$). This procedure reduces the scatter to ±10% of the test data in the generalization (10). As calculations show, Eq. (10), which in physical terms characterizes the degree of flooding of the interfin grooves with condensate in relation to the aggregate conditions of heat exchange reflected by the expression for Fi, may easily be transformed into a relation to calculate the heat-transfer coefficient. Thus, if the mean heat-transfer coefficient for a finned tube is related to the surface of a smooth tube of diameter D_0 (with respect to the base of the fin) and to the temperature head ϑ_0 at the base of the fin, it is expressed by the following ratio:

$$\bar{\alpha}_{\text{fin}} = 1.7 \frac{\rho^2 r a h^3}{\mu \vartheta_0 D_0 s \cos\varphi} \text{Fi}, \tag{11}$$

where $s = 2(a + b + h \tan \varphi)$ is the spacing of the fins.

Further, we may change over from the dimensional form (11) to a heat-exchange equation in the criterial form

$$\text{Nu} = 1.2 (\text{Ga}\,\text{Pr}\,\text{K}\,\text{We})^{0.25} \frac{h^{0.75} D_0^{0.25} \bar{\Theta}^{0.75}}{s \cos^{0.75}\varphi}. \tag{12}$$

Thus, as a result of theoretical analysis of the laws of heat exchange, calculation of fin efficiency, and generalization of available empirical data, we have obtained a theoretical relation describing heat exchange in the condensation of vapors of liquids on horizontal, small-finned tubes ($\text{We} \geq 0.1$). As in all of the other investigations, the relation corresponds in form to the Nusselt equation, and the following ratio must be used as the coefficient c for this equation (in place of $c = 0.923$, for smooth horizontal tubes)

$$c = 1.2\,\text{We}^{0.25} \frac{h^{0.75} D_0^{0.25} \bar{\Theta}^{0.75}}{s \cos^{0.75}\varphi}. \tag{13}$$

Also as in all of the other studies examined, the mean value between the vapor saturation temperature and the temperature of the wall at the base of the fin is taken as the determining temperature in Eqs. (10)-(13), while the diameter of the tube at the base of the fin is taken as the determining dimension.

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EFFECT OF UNSTEADY NATURAL CONVECTION ON THE STRUCTURE OF A LIQUID FLOW IN A HORIZONTAL MIXING CHAMBER

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Results are presented of a numerical and experimental investigation of the effect of natural convection on the structure of a liquid flow in a horizontal mixing chamber with changes in the temperature of the liquid at the inlet.

The study of the effect of natural convection on liquid flow structure, particularly in nuclear power reactor elements with large volumes of coolant, is important to both engineering and science. A characteristic feature of unsteady natural convection due to temperature irregularities and transitional modes of operation is the possibility of stratification of the coolant, with the formation of extensive stagnant zones [1, 2].

In the present work, a horizontal chamber was used to conduct an experimental and numerical study of the structure of liquid flow with a sharp change in temperature at the inlet. The experimental unit was made of organic glass in the form of two rectangular chambers (cross sections of 0.05×0.08 m and 0.05×0.105 m; lengths of 0.24 and 0.15 m, respectively). The chambers were joined at a 90° angle by means of a hydraulic grate installed at the outlet of the horizontal (working chamber). Heaters installed in front of the inlet chamber changed the temperature of the working liquid within the range 15-60°C. The temperature of the distilled water was measured over the height of the working section at a distance of 0.12 m from the outlet, as well as at the inlet and outlet of the horizontal chamber, by Chromel—Copel microthermocouples. The liquid was tinted to make it visible in the chamber.

Figure 1a–c gives a qualitative picture of the characteristic structure of the liquid flow for a transitional (transient) regime, with a decrease in the temperature of the liquid at the inlet and Froude numbers $Fr < 1$. As it enters, the cold liquid occupies the bottom portion of the chamber, and the warmer liquid initially forms a stagnant zone in the top region of the chamber. A feature of this flow regime is that liquid with a temperature $\sim t_0$ is drawn from the chamber outlet into the top, heated region (Fig. 1c). The heated region is relatively stable with respect to connective perturbation from the side of the moving flow. The top region becomes smaller with time and the bottom region increases in size. Temperature is equalized throughout the chamber volume as a result of heat conduction and convection.

Figure 1d–f shows a transitional regime with an increasing liquid temperature at the inlet for $Fr < 1$. Stratification of the liquid flow here is characterized by a cold stagnant zone in the lower region of the chamber. The warmer liquid occupies the top region of the chamber and moves toward the outlet.

Figure 2 shows a temperature profile through the height of the working chamber at a distance of 0.12 m from the outlet for different moments of time after the temperature of the liquid at the inlet in the working section was reduced. The liquid is stratified with respect to temperature through the chamber height, the temperature drop between the top and bottom regions reaching a maximum $\sim(t_0 - t_{in})$ and then gradually decreasing. The liquid is isothermal both in the top, stagnant region of the mixing chamber and in the bottom, convective region. With time, a temperature profile characteristic of heat transfer by conduction is established in the stagnant region. Nearly all of the temperature drop between the regions is concentrated in a narrow boundary layer. Convective perturbations from the side of the moving liquid cannot overcome the jump in density at the interface of the regions, so

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