ANALYSIS OF THE RELATIONSHIPS FOR THE CALCULATION OF HEAT TRANSFER COEFFICIENTS FOR THE FLOW IN VAPOR-GENERATING CHANNELS OF A LIQUID HEATED TO THE SATURATION TEMPERATURE

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Regularities of heat transfer in the motion of a two-phase flow in various-shaped channels has been studied intensively over a period of many years. Numerous calculational procedures appear in the literature based, not on an analytical approach, but on the treatment of experimental results.

The great majority of the calculational relationships can only be used in a restricted range of the parameter regimes. However, many authors treat this situation with insufficient clarity or else they introduce restrictions which, in the light of the presently available experimental data, cannot be regarded as exhaustive.

In this connection the question concerning the proper use in a calculational procedure of one or another of these relationships takes on great practical significance. This is especially important owing to the fact that in carrying out thermal calculations there is a tendency to use obsolete procedures. An approach of this kind usually leads to the creation of unnecessary allowances with respect to vapor-generating surfaces, to an incorrect estimate of the wall temperature regime of a vapor-generating channel, and to a substantial complication in the optimization of a facility.

We attempt here to give a survey and a critical estimate of the known assumptions in the literature for the calculation of heat transfer in the flow of stream in tubes and ducts.

Contemporary stream-generating installations are constructed in such a way that the steam is formed either on the outer surfaces of tubes submerged in a liquid, with allowance made for its free circulation, or on the inner surfaces of steam-generating tubes (ducts) in which a systematic motion of a two-phase flow is effected.

Various flow regimes may be observed in the motion of a two-phase flow in ducts depending on the output of the heat conductor, the fractions of the duct cross-section occupied by the liquid and the gaseous (vapor) phases, the ratios of their velocities, the position of the duct, and the flow directions. The most accurate calculational relationships can, to all outward appearances, be obtained in considering a specific flow regime, but with the regime boundaries sufficiently undetermined, changing with a change in the pressure, the duct length, the entrance temperature, and a whole series of other factors. In this connection, the great majority of the relationships for the calculation of heat transfer intensity in two-phase flows is not associated with a specific flow regime. This is due to the fact that, in spite of the whole variety of regimes, there is a limited number of basic defining parameters which characterize the heat transfer intensity during boiling under two-phase flow conditions in tubes and ducts. As basic parameters of this kind we cite the physical properties of the liquid and gaseous (vapor) phases, the heat conductor output, the specific heat flow q, and the pressure p.

From the modern point of view the coefficient of heat transfer in the motion of a two-phase flow in ducts is a function of a) the intensity of the heat transfer due to turbulization of the wall boundary layer by

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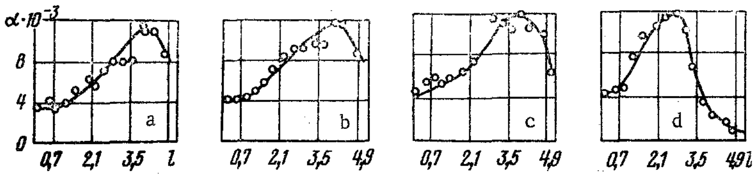

Fig. 1. Variation of the heat transfer coefficient along the tube length for various circulation rates [11]: a) q_{av} = 61,000 kcal/m^2·h; w_0 = 0.5 m/sec; b) corresponding values here are 64,500 and 0.35; c) 72,800 and 0.24; d) 60,000 and 0.15. Units of α and l are, respectively, kcal/m^2·h·°C and m.

vapor bubbles formed during boiling, and b) the intensity of the heat transfer occasioned by the turbulent exchange arising in the forced motion of the two-phase flow.

The influence of the first factor on the heat transfer can be ascertained through a specific thermal loading. The amount of influence of the second factor depends both on the total weight output of the two-phase flow (the circulation rate) and on the velocity of the vapor core (for the dispersed-ring flow regime).

Such a description of the mechanism in the heat transfer process during the boiling of a liquid in tubes was built up as the result of many years of investigation, both in the USSR and elsewhere.

For convenience in their consideration we can conditionally divide the calculational relationships defining heat transfer intensity into four groups, beginning with the list of defining parameters included in the relationship considered.

I. Calculational Procedures in which the Heat Transfer Intensity Depends on the Magnitude of the Specific Thermal Loading and the Pressure

For boiling it is well known [1] that the heat transfer intensity on heat emitting surfaces submerged in a volume of liquid depends only on the specific thermal loading q and the pressure p, and also on the physical properties of the liquid and gaseous phases. The dependence of the heat transfer coefficient on q and p for the boiling of water under free convection conditions was established by V. M. Borishanskii [2, 3]. He obtained the expression

$$\alpha_{\text{l.v.}} = 3\,(p^{0.14} + 1.83\cdot 10^{-4} p^2)\, q^{0.7} \tag{1}$$

verified by experimental data for a wide range of variation of the defining parameters.

Heat transfer for boiling in ducts depends, in the general case, not only on the specific thermal loading and the pressure but also on the rate of systematic movement of the two-phase flow and the magnitude of the local vapor content.

However, a number of the relationships, recommended for conditions of boiling in ducts, were formulated on the basis of experiments in which there was no manifestation of the influence either of the rate of flow or the vapor content, and the heat transfer intensity was determined, as for boiling in a large volume, by the pressure p, the specific thermal loading q, and the physical properties of the phases.

A similar type of dependence was obtained for the first time for ethyl alcohol by S. M. Lukomskii and S. M. Madorskaya back in the forties. According to [4], the heat transfer coefficient for boiling in tubes depends only on the specific thermal loading and the pressure, and can be calculated from the expression

$$\alpha = A(p)\, q^n, \tag{2}$$

where n = 0.73-0.01p; A(p) is a function of the pressure p (kg/cm^2).

Experiments conducted much later by other authors showed that actually there exists a region of the parameter regimes inside of which α is a function only of q and p; however, these authors did not confirm the dependence of the exponent n in the relation (2) on the pressure. At the present time it should be regarded as established [3, 5-10] that for the most prevalent combinations of the liquid—solid surface n ≈ 0.7.

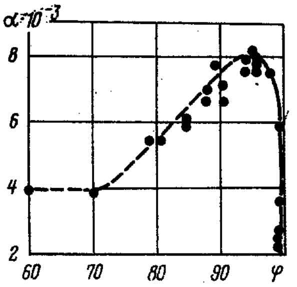

Fig. 2. Local heat transfer coefficients versus flow vapor content [11]. φ, %.

Formulas for calculating the heat transfer coefficient for the boiling of water in tubes, when the heat transfer intensity is determined, in the judgement of the authors, only by the quantities p and q, were proposed by Levy [7]:

$$\alpha = \frac{c'\lambda'\rho'^2}{\sigma T_s(\rho'-\rho'')B_L} q^{\frac{2}{3}}, \qquad (3)$$

where the coefficient B_L is determined empirically and is a function of the product $\rho''r$; by MacNelly [8]:

$$\frac{\alpha d}{\lambda} = 0.255\left[\frac{\nu}{a}\right]^{0.69}\left[\frac{qd}{3600rg\mu}\right]^{0.69}\left[\frac{pd}{\sigma}\right]^{0.31}\left[\frac{\gamma'}{\gamma''}-1\right]^{0.33}; \qquad (4)$$

by Z. L. Miropol'skii and M. E. Shitsman [9]:

$$\alpha = 1.1p^{0.43}q^{0.7}; \qquad (5)$$

by G. V. Alekseev, B. A. Zenkevich, and V. I. Subbotin [10]:

$$\alpha = 0.1K_p^{0.314}q^{0.7}, \qquad (6)$$

where $K_p = p \cdot 10^4/\sqrt{\sigma(\gamma'-\gamma'')}$, and also by M. A. Kichigin and N. Yu. Tobilevich [11]:

$$\frac{\alpha}{\lambda}\left(\frac{\sigma}{\gamma}\right)^{\frac{1}{2}} = 3.25\cdot 10^{-4}\left[\frac{q}{r\gamma'a}\left(\frac{\sigma}{\gamma}\right)^{\frac{1}{2}}\right]^{0.6}\left[\frac{g}{\nu^2}\left(\frac{\sigma}{\gamma}\right)\right]^{0.125}\left[\frac{p}{(\sigma\gamma)^{1/2}}\right]^{0.7}. \qquad (7)$$

The heat transfer coefficient values, obtained in [9, 10] for the condition of boiling in tubes, agree fairly well with the values of α calculated from the formulas for boiling in a large volume. This can be used as a confirmation of the fact that under specified conditions the heat transfer process for boiling in ducts in the case of systematic movement is subject to the same regularities as the process of boiling in a large volume.

A very common material disadvantage, inherent in the formulas (3)-(7) considered above, which makes their possible use in a practical computation difficult, is that the authors do not assign specific boundaries of their applicability beyond which the influence of the vapor content and the rate of movement of the two-phase flow on the heat content should be taken into account.

Thus, for example, from [1] one can make an invalid conclusion concerning the fact that the heat transfer intensity does not depend on the vapor content over a wide range of variation of the latter ($0.05 \le x \le 0.9$).

Apparently, in [10] high vapor contents were obtained for small outflows, but sufficiently high p and q, when the turbulization of the fluid layer at the wall, due to the formation of bubbles, was high, while the speed of the two-phase flow remained low and proved to have no noticeable intensifying influence on the heat transfer. The lack of tabular experimental data in [10] did not allow, unfortunately, verification of the stated supposition by means of an appropriate calculation.

The formula (7), obtained in [11], is an example of a generalization from experimental data, which is only partially successful. The experimental data given there (see Fig. 1) testifies to the influence on the heat transfer intensity during boiling in tubes not only of q and p, but also of the weight vapor content x. For pressures close to atmospheric the influence of the vapor content on the heat transfer intensity begins to manifest itself already for φ = 70 to 75%, i.e., for weight vapor contents on the order of several percent (Fig. 2). However, having deduced the effect of w_0 on the heat transfer intensity and having shown experimentally the vapor content influence, the authors of [11] generalized the experimental data by using heat transfer coefficients α, not local but averaged over the tube length, and also criteria valid for boiling in a large volume. The result of doing this was that the dependence of local heat transfer coefficients on the vapor content did not show up in the formula (7) recommended by the authors of [11].

At the present time relations employing average values of heat transfer coefficients have fouünd fairly wide use, particularly in papers of authors studying heat transfer during boiling in various industrial processes. The calculation of heat transfer from relations of this kind, of a supposedly generalized nature, can only be derived with confidence for conditions close to those for which the experimental data, used in formulating the computational relationship, were obtained. But if the given relation is used for the calculation of heat transfer intensity under other conditions, substantial errors may result.

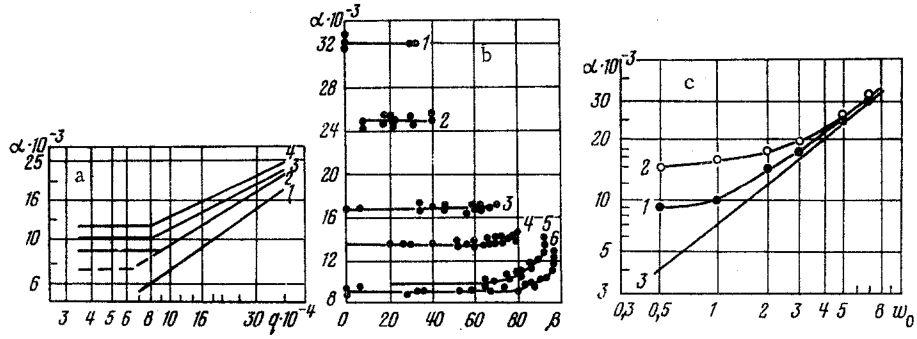

Fig. 3. Dependence of the heat transfer coefficient on a) the number of revolutions of the mixer [12] (Curves 1, 2, 3, and 4 are for n = 0, 165, 295, and 520 rev/min, respectively); b) the circulation rate for $q = 200 \cdot 10^3$ kcal/m$^2 \cdot$h [12] (Curves 1, 2, 3, 4, 5, and 6 are for w_0 = 6.67, 5, 3, 2, 1, and 0.5, respectively); c) the circulation rate and the specific thermal flow [12] (Curves 1 and 2 are for $q = 200 \cdot 10^3$ and $400 \cdot 10^3$ kcal/m$^2 \cdot$h, respectively; Curve 3 is for $Nu = 0.23\, Re^{0.8} Pr^{0.4}$). The units of q, β, and w_0 are, respectively, kcal/m$^2 \cdot$h, %, and m/sec.

II. Calculational Procedures in which the Heat Transfer Intensity Depends on the Value of the Specific Thermal Loading, the Pressure and the Circulation Rate

Experiments conducted by a number of investigators show that there exists a domain for the parameters (w_0, x, q) inside which the heat transfer intensity during boiling under constant pressure conditions depends on only two factors, the specific thermal loading q and the circulation rate w_0.

This question was studied in detail for the first time in [12], wherein experimental data were presented both for the case of forced motion during boiling in a large volume (mixer) and also for the forced motion in tubes. It was shown that for small q the heat transfer intensity during boiling is mainly determined by the speed of the liquid (Fig. 3a). At much higher thermal flows the heat transfer depends on both q and also the circulation rate. In a region of high q and small w_0 the thermal loading is found to have a decisive effect on the heat transfer intensity.

The experimental data [12] for boiling in tubes, shown in Fig. 3b, testifies to the fact that the influence of the circulation rate w_0 on the heat transfer intensity, even for comparatively high thermal flows ($q = 2 \cdot 10^5$ kcal/m$^2 \cdot$h), is fairly large. Thus, in varying the circulation rate from 0.5 to 6.67 m/sec, the value of the heat transfer coefficient increases by a factor of more than 3.5. Experimental curves are presented in Fig. 3c, which show how α varies as a function of the circulation rate for various values of the thermal flow q. If for low w_0 the heat transfer intensity is determined by the bubble-boiling process, then as w_0 increases the circulation rate is found to have a very large influence on the heat transfer, and the values of the heat transfer coefficients α approach values which are typical of convective heat transfer.

It was shown in [12] that the mutual influence of q and w_0 on the heat transfer intensity depends on a function of the number $K_W = q/w_0 r \gamma''$. It should be stated that although the dependence

$$\frac{\alpha}{\alpha_{\mathrm{l.v.}}} = (1 + K_w)^{0.18} \tag{8}$$

assumed in [12], has not been widely adopted, nevertheless, the number K_W was used later on by a number of authors in constructing various computational relationships.

In [13] L. S. Sterman presents a set of numbers characterizing the heat transfer process for boiling in ducts:

$$Nu = f\left(Re,\ Pr,\ Fr,\ \frac{q/r\gamma''}{w_0};\ \frac{r}{c_p T_s};\ \frac{w_0''}{w_0'},\ \frac{\gamma''}{\gamma'}\right).$$

In this general set of numbers there appears the ratio of the reduced velocities of the vapor and liquid phases characterizing the turbulizing influence on the heat transfer of the vapor core motion (vapor content). However, for the derivation of the calculational relationship

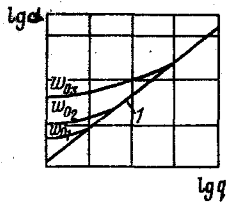

Fig. 4

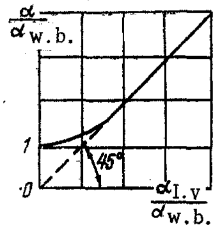

Fig. 5

Fig. 4. Heat transfer to a boiling liquid versus the thermal flow density q and the circulation rate w_0 (see [22]). Curve 1 represents free convection boiling, $w_{01} < w_{02} < w_{03}$.

Fig. 5. The relationship $\alpha/\alpha_{w.b} = f(\alpha'_{l.v}/\alpha_{w.b})$ (see [22]).

$$\frac{Nu_b}{Nu_{w.b.}} = 6150\left[\frac{q/r\gamma''}{w_0}\left(\frac{\gamma''}{\gamma'}\right)^{1.45}\left(\frac{r}{c_pT_s}\right)^{0.33}\right]^{0.7} \tag{9}$$

for

$$\left[\frac{q/r\gamma''}{w_0}\left(\frac{\gamma''}{\gamma'}\right)^{1.45}\left(\frac{r}{c_pT_s}\right)^{0.33}\right] > 0.4\cdot10^{-5}, \qquad \frac{Nu_{w.b.}}{Nu_b} = 1 \tag{10}$$

for

$$\left[\frac{q/r\gamma''}{w_0}\left(\frac{\gamma''}{\gamma'}\right)^{1.45}\left(\frac{r}{c_pT_s}\right)^{0.33}\right] < 0.4\cdot10^{-5}.$$

(Note by translator: in the relations (9) and (10) the subscripts b and w. b. are for "boiling" and "without boiling", respectively.) L. S. Sterman relied only on the experimental data in [4, 14, 15], obtained for conditions in which the vapor core motion did not prove to have a noticeably intensifying influence on the heat transfer. The upshot of this was that the ratio w_0''/w_0' was eliminated from consideration, although the data obtained earlier in [12] (see Fig. 3b) attests to the fact that when $\beta > 70\%$ the heat transfer intensity increases sharply with an increase in the vapor content.

A definite disadvantage of the relation (9) also is that, according to it, the heat transfer coefficient for boiling arbitrary conditions may be expressed in terms of $Nu_{w.b} = 0.023\,Re^{0.8}Pr^{0.4}$, and, consequently, the dependence on d_{equiv} and on the circulation rate w_0 is maintained (the dependence, it is true, is not too strong):

$$\alpha_b \sim d_{equiv}^{-0.2}, \quad \alpha_b \sim w_0^{0.1}.$$

Apparently, just in connection with this case, the relationship (9) turned out to be ill-suited for generalizing the experimental data in [16] for boiling water heat transfer in narrow apertures, however, it was completely satisfactory for generalizing the experimental points in [17] and [18] for boiling in longitudinally-streamlined bundles and apertures with d_{equiv} close to that in [13].

A fundamentally different approach to the problem of taking into account the joint effect of bubble boiling and the forced motion of a liquid was proposed in some papers by authors in the Soviet Union and abroad.

Thus, F. F. Bogdanov [19] proposes to calculate the heat transfer coefficient by a formula which adds together the effects influencing heat transfer intensification for the boiling of a liquid in ducts:

$$\alpha = cp^nq^m + c_1w_0, \tag{11}$$

where c, c_1, n, and m are experimentally determined coefficients.

It was confirmed in [20] that the resulting thermal flow density for boiling under forced motion conditions can be calculated by the direct addition of the thermal flows calculated from the formulas for boiling in a volume and for the forced motion of the liquid without boiling:

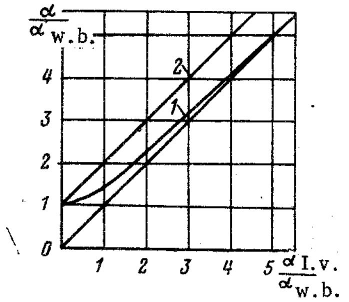

Fig. 6. Comparison of two relationships: Curve 1 is for S. S. Kutateladze's equation (see [22]) and Curve 2 is for W. M. Rohsenow's equation (see [20]).

$$q = q_{I.v} + q_{w.b}. \tag{12}$$

The author argues that equation (12) is applicable in the region of reduced vapor contents, i.e., in a region where the heat transfer for a given pressure is completely determined by the quantities w_0 and q. For the conditions of boiling in a duct of a liquid heated to saturation temperature, the equation (12) can be written in the form

$$\alpha = \alpha_{I.v} + \alpha_{w.b} \tag{13}$$

In his papers [21], [22] S. S. Kutateladze remarks that the presence of a systematic motion (forced or natural circulation) leads to an intensification of the heat transfer process for bubble-boiling. The extent to which this motion influences the heat transfer depends on the relationships of the magnitudes of the turbulent perturbations, arising at the expense of the systematic motion, and the vapor formation process.* For a given circulation rate w_0, depending on the size of the thermal flow increase, the heat transfer coefficient changes vary little at first, after which the influence of q becomes all the more noticeable, until it becomes decisive. As a result, we have, as the envelope of the curve $\alpha(q, w_0)$, a curve $\alpha(q)$ (Fig. 4), which in nature is close to the corresponding relationship for free convection boiling. Upon examining Fig. 4, we can identify three typical zones:

Zone I: Heat transfer here is mainly determined by forced motion (zone of small q values);

Zone II: Heat transfer is determined by the process of vapor formation (zone of large q values):

Zone III: Characterized by the joint influence of forced motion and the vapor formation process (zone of average q values).

D. A. Labuntsov [25] prescribes a separate relationship for each zone:

$$\text{Zone I} \quad 0 < \frac{\alpha_{I.v.}}{\alpha_{w.b.}} < 0.5, \quad \alpha = \alpha_{w.b.} \tag{14a}$$

$$\text{Zone II} \quad \frac{\alpha_{I.v}}{\alpha_{w.b.}} > 2, \quad \alpha = \alpha_{I.v}; \tag{14b}$$

$$\text{Zone III} \quad 0.5 < \frac{\alpha_{I.v}}{\alpha_{w.b}} < 2, \quad \frac{\alpha}{\alpha_{w.b.}} = \frac{4\alpha_{w.b} + \alpha_{I.v.}}{5\alpha_{w.b} - \alpha_{w.b.}} \tag{14c}$$

The formulas (14) satisfactorily generalize the experimental data for a region in which the heat transfer intensity is determined by the mutual influence of q, w_0, and p. These formulas, however, do not specify passages to the limit in the domain of the parameters; this is more suitably handled by S. S. Kutateladze's interpolational relation [22]

$$\frac{\alpha}{\alpha_{w.b.}} = \sqrt[n]{1 + \left(\frac{\alpha'_{I.v}}{\alpha_{w.b}}\right)^n}, \tag{15}$$

which is valid for all three zones existing for boiling in a duct of a liquid heated to the saturation temperature. Here α is the coefficient of heat transfer to a flow of boiling liquid; $\alpha_{w.b}$ is the heat transfer coefficient for forced motion of the liquid without boiling, calculated with respect to w_0; $\alpha_{w.b} = 0.023\ (\lambda/D)\ Re^{0.8} Pr^{0.4}$; $\alpha'_{I.v} = (0.7\text{–}0.8)\,\alpha_{I.v}$; $\alpha_{I.v}$ is the heat transfer coefficient for boiling in a large volume according to the formula (1).

In the determination of the exponent n in the relation (15) use was made of experimental data obtained for conditions in which an increase of vapor content and, consequently, also the vapor flow velocity, did not lead to an increase in the heat transfer intensity.† Formula (15) describes the experimental data quite well when n = 2. However, the authors of [16] claim that much better agreement with the experimental points is obtained for n = 3 in the case of narrow annular passages.

*An analogous influence of a systematic motion and the vapor formation process on the intensity of boiling heat transfer was noted in many experimental papers by Soviet authors and others [12, 23, 24].
†These same data were used in formulating the relations (9) and (10).

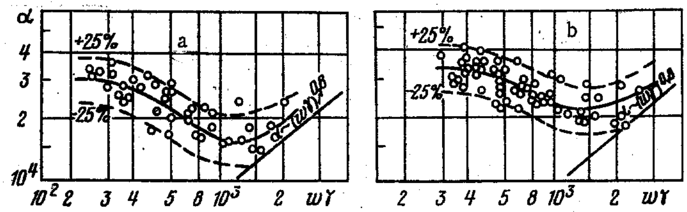

Fig. 7. Heat transfer coefficient versus weight velocity for q = $140 \cdot 10^3$ in Fig. 7a, and q = $200 \cdot 10^3$ kcal $/m^2 \cdot h$ in Fig. 7b (from [27]). The units of $w\gamma$ are kg $/m^2$ sec.

The nature of the dimensionless relationship proposed by S. S. Kutateladze is shown schematically in Fig. 5. Its limiting values are expressed by the following conditions:

$$\frac{\alpha'_{l.v.}}{\alpha_{w.b.}} \to 0; \quad f \to 1; \quad f' \to 0,$$

$$\frac{\alpha'_{l.v.}}{\alpha_{w.b.}} \to \infty; \quad f \to \frac{\alpha'_{l.v.}}{\alpha_{w.b.}}; \quad f' \to 1.$$

It is clear from the above that the equation (13), due to W. M. Rohsenow (see [20]), is a particular case of the relation (15) for n = 1*.

A graphical representation of these relationships appears in Fig. 6. Relation (15) furnishes a more proper approach since its use allows for a smooth passage to the limit for both convection heat transfer (when $\alpha'_{l.v}/\alpha_{w.b} \to 0$, $\alpha \to \alpha_{w.b}$) and developed boiling heat transfer in ducts, when there is no circulation rate effect and the heat transfer is determined by the thermal flow q (when $\alpha'_{l.v}/\alpha_{w.b} \to \infty$, $\alpha \to \alpha'_{l.v}$). From the point of view of the physical process involved the description (15) is more logical than the description of the process given by W. M. Rohsenow. Summation of heat transfer coefficients is apparently allowable when considering independent methods of thermal energy transport (for example, the total influence of radiation and convection). As for the process of boiling for flow in ducts, there are two processes involved here, which cause turbulization of the liquid layer at the wall. Since the vapor formation process is developed in the liquid layer at the wall, it is logical to assume that the relative contribution of the convective transport process to the total heat transfer must drop fairly rapidly with an increase in intensity of the formation process. These considerations are incorporated in the relation (15) but are not accounted for in relation (13).

In the overwhelming majority of papers devoted to the study of boiling heat transfer of a liquid moving in a channel, it has been shown experimentally that increasing the speed of the liquid leads to an intensification of heat transfer. This physically justifiable assumption is reflected in the computational formulas (8)-(15) proposed by various authors, which were considered earlier. The structure of all these relationships is such that an increase in the speed of the liquid leads to an intensification of the boiling heat transfer, although the fraction of the convective heat transfer in the overall heat transport varies depending on the formula considered. Nevertheless, there are a limited number of papers in which it is asserted that an increase in the speed of the liquid leads to a decrease in the heat transfer coefficient, although the authors of these papers give no explanation for the resultant effect.

In [26] E. K. Averin and G. N. Kruzhilin, having studied boiling heat transfer of water in an annular passage, established the fact that in a region of high thermal flows ($q > 4 \cdot 10^5$ kcal/$m^2 \cdot h$) with a pressure p in the range from 1 to 9 kg/cm^2 the heat transfer coefficients α decrease with an increase in the rate of flow of the liquid. For the conditions they considered the authors of [26] proposed the following calculational relationship:

$$\alpha = 7.8\, q^{0.6} w_0^{-0.1} p^{0.28}. \tag{16}$$

As for small thermal flows, the usual influence of the liquid speed on the boiling heat transfer was noted, i.e., the heat transfer intensity increased with an increase in w_0.

*The quantity $\alpha_{l.v}$ appears in equation (13) whereas in equation (15) the quantity $\alpha'_{l.v}$. This fact of itself does not interfere with making a comparative analysis; however, in clarifying the fundamental difference between these equations it is more convenient to assume, conditionally, in relation (13) that $\alpha_{l.v} = \alpha'_{l.v}$.

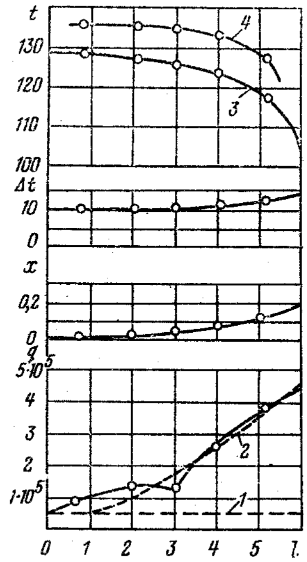

Fig. 8. Variation of local heat flow, temperature head, and weight vapor content along tube length with water flow rate G = 1320 kg/h [30]: Curve 1 is for convective heat transfer with fluid motion; Curve 2 is for convective heat transfer with two-phase flow. Calculation was carried out by the formula for fluid convective heat transfer with velocity equal to that of the two-phase flow; Curve 3 gives the flow temperature t in °C; Curve 4 gives the mean wall temperature t in °C.

It can be assumed that under the conditions involving large thermal flows and flow rates a very strong suppression of the bubble-boiling process takes place. However, the anomalous influence of the liquid speed on the boiling heat transfer could possibly result from peculiarities of the experimental method employed. The length of the heated portion was equal to 60 mm with an annular gap width of 8 mm, i.e., one cannot speak here either of thermal or of hydrodynamic stabilization of the flow.

Yu. A. Zeigarnik and A. S. Komendantov [27] made a study of boiling heat transfer of water for forced flow conditions in a tube under atmospheric pressure. A substantial portion of the experimental material obtained in [27], as well as the conclusions made therein, are at variance with the results obtained by the majority of investigators, both in the Soviet Union and abroad, who studied the influence of the fundamental defining parameters on boiling heat transfer of a liquid in tubes.

In Fig. 7 graphs taken from [27] are presented in which the dependence of the heat transfer coefficient on $w\gamma$ for boiling in tubes has a rather involved character, not even monotonic. From the graph it follows that when $w\gamma < 1000$ kg/m$^2\cdot$sec, a decrease in the outflow leads not to a lowering of heat transfer intensity, as observed by the majority of authors, but rather to an increase; moreover, under these conditions the measured values of α turn out to be substantially higher than those calculated from the formulas (9), (10), (14), and (15). When $w\gamma > 1000$ kg/m$^2\cdot$sec, the coefficient α depends on $w\gamma$ in the usual way and the resultant heat transfer coefficient values agree satisfactorily with the values calculated from the formulas mentioned. In analyzing the results obtained in [27] it should be recalled that the authors pointed out that the majority of their results were accompanied by fluctuations, which vanished for $w\gamma > 1000$ kg/m$^2\cdot$sec.

In our opinion, the conclusions made on the basis of experimental data, obtained from regimes with fluctuations, should in no case be extended to regimes without fluctuations. The fact should also be taken into account that in [27] the influence of vapor formation on the heat transfer coefficient was, apparently, not taken into account with sufficient clarity.

Taking into account the method used in carrying out the experiment, and also the fact that in constructing the graphs in question no differentiation was made in the values of α with regard to the vapor content values, we can assume that the heat transfer coefficient values were obtained more for low $w\gamma$ values than for large vapor content values.

Thus an increase in heat transfer intensity with a decrease in $w\gamma$ (see Fig. 7) can be determined, not by a lowering of the weight velocity, which must decrease the value of the heat transfer coefficient α, but by an increase in the vapor content, leading to an intensification of the thermal transport process.

III. Calculational Procedures in which the Heat Transfer Intensity Depends on the Vapor Content (Velocity of the Two-Phase Flow)

The experimental data, including that obtained as far back as the 1940's by M. A. Kichigin and

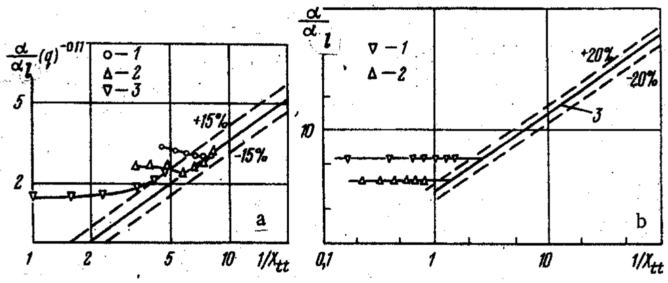

Fig. 9. Heat transfer coefficient ratio versus the Martinelli parameter for various q and $w\gamma$ values in Fig. 9a, and for various thermal flow values q in Fig. 9b. Data in Fig. 9a are from [35]; in Fig. 9b from [32]. Units of q and $w\gamma$ are, respectively, kcal $/m^2 \cdot h$ and $kg/m^2 \cdot h$. In Fig. 9a the (q, $w\gamma$) data are as follows: 1) ($8.6 \cdot 10^4$, $26.3 \cdot 10^4$); 2) ($17.2 \cdot 10^4$, $37.1 \cdot 10^4$); 3) ($5.5 \cdot 10^4$, $7.75 \cdot 10^4$). In Fig. 9b, $w\gamma = 9.76 \cdot 10^5$ and the q data are as follows: 1), $3.39 \cdot 10^5$; 2) $1.68 \cdot 10^5$; Curve 3 is the graph of $\alpha/\alpha_l = 2.9(1/X_{tt})^{0.66}$.

N. Yu. Tobilevich [11] and also that obtained by L. S. Strerman and N. G. Styushin [14], enables us to conclude that starting from a specific vapor content value an increase in the vapor content leads to an essential growth of the heat transfer coefficient (Figs. 2, 3b). The data of Fig. 3b, taken from [11], confirms the absence of a volumetric vapor content influence on the heat transfer for a region of low vapor contents. However, when a definite vapor content is attained, a sharp rise in the heat transfer intensity is observed. As the circulation rate is increased, the vapor content influence on the heat transfer level manifests itself for smaller values of the volumetric vapor content β. This is indicative of the fact that the intensifying influence on the heat transfer is due not only to the vapor content value but also to the faster speed of the two-phase flow. For this reason the vapor content value cannot serve as the only criterion determining the boundaries of the region with a preeminent influence of bubble-boiling on the heat transfer process. However, a separate study of this problem was not made in [11], [12], so that corresponding procedures were not obtained.

An increase in heat transfer intensity with an increase in vapor content up to a value of $x = 0.3$ was noted by N. V. Tarasov, A. A. Armand, and A. S. Kon'kov [28], who studied boiling heat transfer for a water vapor mixture in a tube at a pressure of $p = 170$ kg/cm^2.

The following calculational relationship is recommended by these authors:

$$\alpha = 14q^{0.7}(1 + \gamma'/\gamma'' x). \tag{17}$$

The drawback to the formula (17) is that it does not take into account the influence of the two-phase outflow rate, an increase in which can lead to a substantial intensification of the heat transfer [13, 22, 25].

A fairly strong rise in the heat transfer intensity with an increase in vapor content was obtained experimentally in [29]. However, the authors did not supply a calculational relation for this zone.

In contrast to the very limited number of papers written by Soviet authors in which an attempt is made to take the influence of vapor flow motion into account, a fairly large number of such papers has appeared abroad.

Dengler [30, 31], studying boiling heat transfer of water in a vertical tube heated by a condensing vapor, shows that the substantial increase in the heat transfer coefficient along the tube length cannot be explained by the intensifying effect of the boiling process. He remarks that the two-phase flow motion due to the vapor is a fundamental factor influencing the heat transfer process for high vapor contents (Fig. 8). The conclusion that the influence of the vapor content on the heat transfer intensity manifests itself in terms of an increase in the vapor core velocity is confirmed by the fact that in [30, 31] the heat transfer coefficient growth was fixed for a constant vapor content of the flow with an increase in the thermal carrier weight outflow rate; also fixed was the drop in heat transfer intensity with an increase of the pressure. *

Tong [32] has shown, for two-phase flow in an annulus, that when the volumetric vapor content is sufficiently high the vapor core velocity can be so large, and the turbulence on the vapor—liquid separation boundary can be so intense, that the mechanism of heat transfer from the heating surface to the flow changes in comparison with the conditions obtaining when intense bubble-boiling is observed. For high velocities of the two-phase flow heat is given up by heat conduction through a thin layer of liquid flowing along the channel wall, and, on the boundary separating the film of liquid and the vapor core very intense vaporization takes place.

This was established both with the aid of visual observations [33] and also by uncovering the factors defining the heat transfer intensity in the zone considered, which is referred to in Tong's book as the "zone of vaporization under forced convection."

An analogous conclusion was arrived at earlier by Guerriery and Talty [34], who remarked in their paper that the convective heat transfer, whose intensity is determined by the velocity of the two-phase flow, is the main factor determining the heat transfer mechanism in the region of increased vapor content.

Thus, a number of foreign authors [30-34] have stated fairly clearly that in the region of high vapor content the basic parameter defining the heat transfer process is, not the vapor content, but the velocity of the two-phase flow. However, in developing their calculational relationships the authors introduce, as the defining quantity, not the vapor core velocity w", but the Martinelli parameter

$$X_{tt}=\left(\frac{1-x}{x}\right)^{0.9}\left(\frac{v'}{v''}\right)^{0.5}\left(\frac{\mu'}{\mu''}\right)^{0.1}.$$

Thus, according to Dengler [30, 31], the formula for calculating the heat transfer coefficient for two-phase flow in tubes for conditions in which the heat transfer intensity is determined by the two-phase flow motion, and does not depend on the bubble-boiling process, has the form

$$\frac{\alpha}{\alpha_l}=3.5\left(\frac{1}{X_{tt}}\right)^{0.5}, \tag{18}$$

where α_l is the convective heat transfer coefficient calculated according to the circulation rate.

Formula (18) is valid for variation of the quantity $1/X_{tt}$ between the limits of 0.25 and 70.

Very close to this is the relation

$$\frac{\alpha}{\alpha_l}=3.4\left(\frac{1}{X_{tt}}\right)^{0.45} \tag{19}$$

proposed by Guerriery and Talty [34]. However the calculation of the convective coefficient of heat transfer to the liquid, is given, in this case, not for the circulation rate w_0, but for the reduced velocity of the liquid phase $w_0(1-x)$.

Heat transfer for the flow of a water—steam mixture in an annular channel with internal heating and pressures close to atmospheric was studied in [35] by Bennett, Collier, Pratt, and Thornton. Experimental data, obtained for q = const, was processed in the form

$$\frac{\alpha}{\alpha_l}=f\left(\frac{1}{X_{tt}}\right),$$

where the calculation of α_l was carried out both with respect to the circulation rate w_0 and the reduced velocity of the liquid $w_0(1-x)$. A better generalization of the experimental data was obtained when the reduced velocity was used.

A number of authors [43, 44, 45] propose calculational procedures in which the heat transfer coefficient is a function of the average velocity of the two-phase flow. † The main drawback to these formulas

*An increase in the outflow rate for x = const leads to an increase in the two-phase flow velocity, whereas an increase in the pressure, lowering the value of the specific volume of the first phase, decreases the two-phase flow velocity.

†We shall not give these formulas here since they are not widely recognized and are available in Collier's survey paper [39].

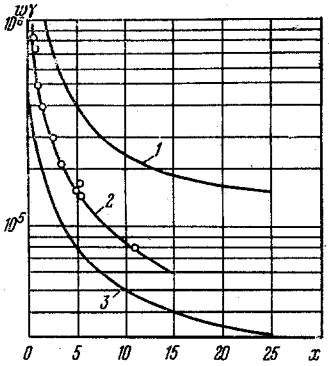

Fig. 10. Boundary (Curve 2) between regions of bubble-boiling and vaporization under forced convection [32]. q = 16 .8 · 10^4 kcal/m^2 · h; p = 1.03 atm; x is the weight vapor content; Curve 1 is for dispersed flow; Curve 2 for annular flow; Curve 3 for slug flow.

is that they cannot account for the process involving vapor content variation along the channel length and its influence on the heat transfer intensity.

Tong [32], in considering the large number of papers devoted to the study of heat transfer in two-phase flows, came to the conclusion that the heat transfer calculation for the vaporization zone in forced convection should follow a functional dependence of the form

$$\frac{\alpha}{\alpha_l} = A\left(\frac{1}{X_{tt}}\right)^n,$$

where A and n are constant coefficients having differing values for the data of the various authors: A = 3.5 and n = 0.5 for Dengler's data for tubes [30]; A = 3.4 and n = 0.45 for Guerriery's and Talty's data for tubes [34]; A = 3.5 and n = 0.79 for the data of Collier et al for longitudinally-streamlined bundles for s/d = 1.64 [40]; A = 3.5 and n = 0.75 for the data of Collier et al for longitudinally-streamlined bundles for s/d = 1.18 [40]; A = 2.17 and n = 0.070 for the data of Collier et al for annular passages [41]; A = 2.72 and n = 0.58 for Wright's data for tubes [42].

The formulas (18)-(20) are applicable only for a region in which the influence is primarily that of forced convection (vaporization zone for forced convection); however, the boundaries of this zone cannot be clearly defined by authors of relationships of the type (20).

Thus, it was shown in [35] that for values of the parameter $1/X_{tt}$ in the range from 2 to 5, the vapor content varies from 7 to 15%, and deviations of the experimental data from the relationship (20) are observed. These data, obtained for various weight velocity values $w\gamma$ and thermal flows q (Fig. 9a), attest to the change in the nature of the heat transfer mechanism and indicate that, depending on the values of $w\gamma$ and q, the vaporization condition result for forced convection is accomplished for various values of the parameter $1/X_{tt}$ and, consequently, for various vapor contents (all the experiments were carried out with p = const). An even more descriptive confirmation of this fact, namely, that the vaporization condition result for forced convection depends on the bubble-boiling intensity, is furnished by the data shown in Fig. 9b, taken from [32]. Both of the regimes considered differ only in their thermal flow values q, since their pressures and mass rates are identical. The experimental data show that the larger the intensity of the vapor formation process, defined for the conditions p = const by the value of the specific thermal flow q, the larger the values of the parameter $1/X_{tt}$, and consequently, for large vapor contents the vaporization condition result is observed for the forced convection of a vapor—liquid flow.

It is natural that the pressure must also have an influence on the value of the parameters for which bubble-boiling is completely suppressed, although the influence of the pressure must be weaker than the influence of the thermal flow. Thus the situation which must obtain in order for the two-phase motion to have a primary influence on the heat transfer must be one involving a relationship among all the basic parameters, and, in particular, the parameters $w\gamma$, q, p, and x.

To make it possible to carry out calculations with relationships of the type (20), a graph was suggested in [32] (see Fig. 19) giving an experimentally determined boundary between the bubble-boiling region and the vaporization region for forced convection of a stream-water flow (q = 16.8 · 10^4 kcal/m^2 · h; p = 1.03 atm).

This graph is found to be inadmissible for any other relationship involving the quantities q and p. Thus, it may be concluded that, actually, there are in the literature no specific procedures for determining the "points of transition" to the condition of vaporization in forced convection, a situation which makes the use of relations of the type (20) computationally difficult. It is necessary to keep in mind that the concept of a "point of transition" introduced here is a simplification of the problem in question since, in point of fact, a transition zone exists between the region of bubble-boiling and the vaporization region in which both mechanisms have an influence on the heat transfer intensity.

IV. Calculational Procedures in which the Heat Transfer Intensity Depends on the Values of the Specific Thermal Loading, the Pressure, the Circulation Rate, and the Vapor Content

From the above survey of the papers of Soviet authors and others we can conclude that, in the general case, the heat transfer intensity for steam—water two-phase flow in tubes is a function of the specific thermal loading q, the circulation rate w_0, the weight vapor content x, and the pressure p. The existence of a similar relationship for boiling in tubes was demonstrated experimentally for the first time in [12]. However, in [12] attention was directed mainly towards identifying the influence on the convection heat transfer intensity for forced motion of a liquid. In this connection, both in [12] and the much later paper [13], the influence of x on α in the calculational procedures was not taken into account. Later on, in the introduction to [39], an attempt was made by L. S. Sterman to extend his relations (9) and (10) to the region where an increase in the vapor content leads to an intensification of the heat transfer process. He recommended introducing into the calculational relationships a true mean velocity of the liquid in place of the circulation rate w_0. This velocity is defined for the flow in question as the following function of the true vapor content φ:

$$w' = \frac{w_0 (1 - x)}{(1 - \varphi)} . \tag{21}$$

This shows that relationships employing the true mean velocity of the liquid w', can be applied so long as the core-like flow regime of the two-phase flow is not violated, i.e., up until the onset of conditions under which liquid drops begin to break away from the film surface and become part of the vapor flow.

However, the relations (9), (10), and (21) are computationally difficult to use in practice owing to the necessity of knowing the distribution of the true volumetric vapor content along the channel length and, also, owing to the fact that the data concerning the boundaries for the existence of the various two-phase flow regimes reflect, to a considerable degree, the subjective opinions of the individual investigators.

In a number of papers [30, 31], in which the experiments were advantageously carried out in a high vapor content region, it was noted that for small values of the parameter $1/X_{tt}$ (zone of combined influence of forced convection and vapor formation) the values of the experimentally determined heat transfer coefficients turned out to be substantially higher than those calculated from a relationship of the type (20).

As a consequence of this, Dengler [30, 31] was forced to introduce into the calculational relation (18) a correction coefficient A_q, taking into account the influence of bubble-boiling on the heat transfer intensity:

$$\frac{\alpha}{\alpha_l} = A_q \cdot 3.5 \left(\frac{1}{X_{tt}}\right)^{0.5} , \tag{22}$$

where

$$A_q = 0.67 \left\{(\Delta t - \Delta t_L) \left[\left(\frac{\partial p}{\partial T}\right)_s \frac{d}{\sigma}\right]_{t_W}\right\}^{0.1} ;$$

$\partial p / \partial T$ gives the variation of the pressure with the temperature on the curve of saturation; σ is the surface tension of the liquid.

The correction coefficient A_q is used only in cases when its value is greater than one. Calculation of the coefficient A_q was based on a number of far-from-obvious assumptions which cast doubt onto the reliability of the proposed reliationship. For instance, one of these assumptions is that the effective temperature head of the boiling process is not the total temperature head $\Delta t = t_W - t_S$, but rather the difference $(\Delta t - \Delta t_L)$, where Δt_L is a provisional temperature head between the wall and the boiling liquid, depending on the local flow velocity and corresponding to the conditions under which the influence of bubble-boiling on the heat transfer does not appear (the bubble-boiling is suppressed).

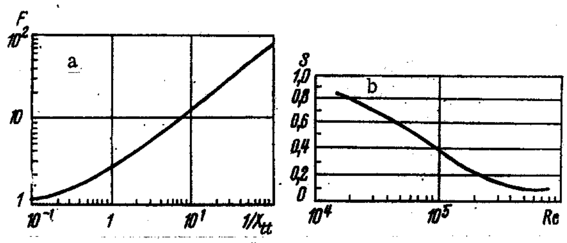

Fig. 11. Dimensionless functions F and S in Figs. 11a and 11b, respectively. $1/X_{tt} = (x/1-x)^{0.9}(\rho'/\rho'')^{0.5} \times (\mu'/\mu'')^{0.1}$; $Re = Re_l F^{1.25}$.

According to Bennett et al [35], the weak influence of the thermal flow on the heat transfer appears throughout the region where the main effect on the heat transport process is found to be the velocity of the two-phase flow.

For zones where $1/X_{tt}$ varies over an interval from 5 to 40, the authors of [35] recommended the following calculational formula:

$$\frac{\alpha}{\alpha_l}(q)^{-0.11} = 0.64\left(\frac{1}{X_{tt}}\right)^{0.74}. \tag{23}$$

The very structure of the relation (23), wherein an increase in the heat flow leads to a proportional increase in the heat transfer intensity over the whole range of the parameters studied, casts great doubt on its validity. This relationship, as well as those employing correctional coefficients like A_q, have not found wide usage in computational work.

Schrock and Grossman [43] made an attempt to introduce a single formula both for the case of bubble-boiling and the case of vaporization under forced convection, wherein they assumed a relation of the form

$$\frac{Nu}{Re_l^{0.8} Pr_l^{0.4}} = k_1 B_0 + k_2\left(\frac{1}{X_{tt}}\right)^n, \tag{24}$$

where k_1, k_2, and n are constant coefficients;

$$B_0 = \frac{q}{w\gamma r} = \frac{q/r\gamma}{w_0}.$$

Formula (24) suffers from a whole series of essential defects.

1. It is based on the principle that for arbitrary two-phase flow regimes bubble-boiling is not suppressed and the amount of heat transmitted as the result of vapor formation on the channel wall is additive with the heat transmitted as the result of vaporization from the film surface at high two-phase flow velocities. *

2. According to formula (24) the heat transfer coefficient depends on the thermal flow to the first power, although the great majority of investigators have shown that $\alpha \sim q^{0.7}$.

3. For the case in which the bubble-boiling process is the primary influence on the heat transfer intensity, the dependence of the heat transfer coefficient on the circulation rate w_0 and the channel diameter is maintained. In the papers considered above, it was shown that in this case neither w_0 nor d should have an influence on the quantity of heat transmitted.

An attempt to take into account the combined influence of the two-phase flow motion and the bubble-boiling process on the quantity of heat transmitted was also made by Chen [4]. By empirical means he obtained the values of the two dimensionless functions (Fig. 11):

$$F = f\left(\frac{1}{X_{tt}}\right) \text{ and } S = f_1\left(\frac{1}{X_{tt}};\quad Re_l\right),$$

which take into account the variation of the heat transfer due to boiling and to forced convection. Chen proposed a relation, representable symbolically as follows:

$$\alpha = \alpha_{boil} S + \alpha_{conv} F.$$

Tong, in analyzing the formula (25), concluded that it could be recommended for calculations of heat transfer in two-phase flows, since it implies, when X_{tt} = const, that the contribution of convection to the overall heat transfer does not depend on the intensity of boiling.

In the survey we have presented 23 relationships, recommended by various authors for calculating

*An analogous principle of addition of effects, along with its drawbacks, was examined in detail above in connection with relation (12), proposed by W. M. Rohsenow for taking into account the joint influence of bubble-boiling and the fluid flow motion.

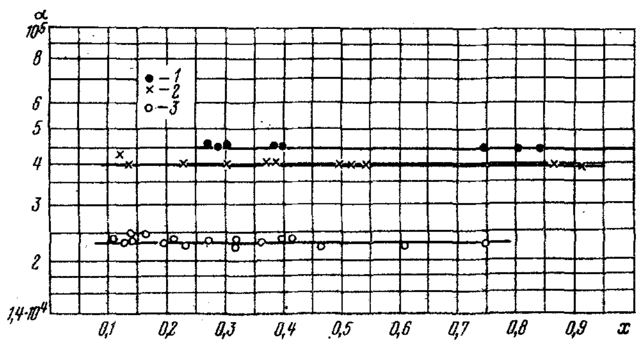

Fig. 12. Illustration of the absence of an explicit effect of vapor content on the heat transfer, keeping the mixture velocity constant. For the data 1) the values of the triple of parameters (w_{mix}, p, q) are (95, 11, $300 \cdot 10^3$); for the data 2) they are (60, 19, $300 \cdot 10^3$); for the data 3) they are (15, 19, $300 \cdot 10^3$). The units of the parameters w_{mix}, p, and q are, respectively, m/sec, kg/cm^2, and kcal/m$^2 \cdot$h.

the heat transfer for two-phase flows in tubes and ducts. With regard to the relations (1)-(20), not all of them are universal relations; they can be used with confidence only in restricted intervals and under specified conditions.

For zones in which the heat transfer intensity is determined by physical properties, namely, the heat flow q, the pressure p, the circulation rate w_0, and does not depend on the vapor flow motion, the most effective calculational formula, in our opinion, is given by the relation (15), although even for it specific boundaries of applicability have not been defined.

Relation (15) cannot be used to calculate the heat transfer for high velocity two-phase flows. For the conditions considered here the two-phase flow motion is found to have a substantial influence on the heat transfer; moreover, as follows from [30, 32, 34, et al], its intensity can turn out to be many times higher than the values obtained in calculating the heat transfer coefficient from formula (15). In [30, 32, 34], in calculating the heat transfer in high velocity two-phase flows it is proposed that relations of the type (20) be used. They are suitable for calculation of the heat transfer coefficient in a zone where the heat transfer intensity is determined solely by the two-phase flow motion and is independent of bubble-boiling. However, the boundaries of this zone, which depend on the relationships among the fundamental defining parameters, namely, w_0, q, p, and x, have not as yet been clearly defined.

As for the formulas (22)-(25), which are of a universal nature, they suffer, as noted above, from a number of very substantial flaws, so that their use can lead to significant errors.

The absence of a simple and reliable relationship for calculating the heat transfer coefficient for two-phase steam—water flow in ducts over a range of parameter variation from x = 0 to the onset of a crisis has occasioned the necessity of writing an appropriate paper; this was accomplished over a period of years at the I. I. Polzunov Central Control Technology Institute.

According to the data in [45-47], the heat transfer coefficient for a two-phase flow in ducts is a function of

a) the heat transfer intensity arising from the turbulization of the wall boundary layer by vapor bubbles which form during boiling;

b) the heat transfer intensity due to the turbulent exchange arising during the forced motion of the two-phase flow.

The effect of the first factor on the heat transfer can be taken into account through a separate thermal loading. The extent of the influence of the second factor depends on the total weight outflow rate of the two-phase flow (on the circulation rate) and on the vapor core velocity (for a dispersed annular flow regime).

In [45-47] it was shown (Fig. 12) that in a region in which the heat transfer is affected by the motion of a high velocity steam—water flow the combined influence of the thermal carrier outflow rate (circulation

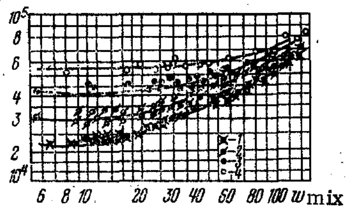

Fig. 13. Heat transfer coefficient α versus the mixture velocity w_{mix} and the thermal loading q (see [46]). Units of p, q, and w_{mix} are, respectively, kg/cm^2, $kcal/m^2 \cdot h$, and m/sec. Here p = 19, and the values of q for the Data 1, 2, and 3 are, respectively, $300 \cdot 10^3$, $500 \cdot 10^3$, and $1200 \cdot 10^3$.

rate) and the vapor content on the heat transfer intensity is best taken into account by the reduced velocity of the two-phase mixture, W_{mix}, where $w_{mix} = w_0 (1 + (\gamma' - \gamma''/\gamma'')x)$; as $x \to 0$, $w_{mix} \to w_0$; as $x \to 1$, $w_{mix} \to w''$.

The experimental data obtained in [45-47] (see Fig. 13) enable us to make the following conclusions.

1. In a region where the two-phase flow velocity has a small influence on the heat transfer, an increase in the thermal loading leads to a significant increase in the heat transfer coefficient. Under these conditions the relationship obtained for conditions of boiling in a large volume are valid.

2. The influence of the pressure on the heat transfer coefficient in a region of developed boiling for the forced motion of a two-phase flow in tubes may be expressed by the same relationship as that used for boiling in a large volume.

3. In a region of high velocities of a two-phase flow ($w_{mix} > 100$ m/sec) the thermal loading effect is comparatively small and the heat transfer intensity is determined by the forced convection, which depends on the mixture flow velocity w_{mix}.

4. There exists a transition region where the influence on the heat transfer is due both to bubble-boiling, whose intensity for p = const depends on the thermal flow q, and to the two-phase flow motion. The smaller the thermal flow q, the earlier the influence of the two-phase flow motion manifests itself on the heat transfer and the stronger this influence.

Noting the nature of the influence of the fundamental parameters on the heat transfer intensity, the authors of [45-47] used a method for generalizing the experimental data analogous to that employed by S. S. Kutateladze in deriving the relation (15). For this an additional term α'_K was introduced, which takes into account the intensifying effect of the vapor core velocity on the heat transfer.

With this taken into account, the calculational relationship, suitable for determining the heat transfer coefficient in an arbitrary region of mutual influence of the parameters q, w_0, x, and p, assumes the following form:

$$\alpha = \sqrt{(\alpha'_{I.v.})^2 + \alpha_k^2 + (\alpha'_k)^2} \tag{26}$$

or

$$\frac{\alpha}{\alpha_k} = \sqrt{1 + \left(\frac{\alpha'_k}{\alpha'_{I.v.}}\right)^2 \left(\frac{\alpha'_{I.v.}}{\alpha_k}\right)^2}, \tag{27}$$

where α_K is the value of the heat transfer coefficient calculated from S. S. Kutateladze's formula (15).

According to the data of [45-47], the ratio $\alpha'_k/\alpha'_{I.v}$ can be represented as a function of the dimensionless grouping $w_{mix}\gamma' r/q$, which should be regarded as the ratio of a quantity proportional to the weight velocity in the film ($w_{mix}\gamma'$) to the weight velocity of the vapor generated in the liquid layer at the wall.

In this case, according to [47], the equation (27) assumes the form

$$\frac{\alpha}{\alpha_k} = \sqrt{1 + 7 \cdot 10^{-9} \left(\frac{w_{mix} r \gamma'}{q}\right)^{\frac{3}{2}} \left(\frac{\alpha'_{I.v.}}{\alpha_k}\right)^2}. \tag{28}$$

A noticeable effect of the two-phase flow velocity on the heat transfer begins to manifest itself for a value of the grouping

$$\left(\frac{w_{mix} r \gamma'}{q}\right) \left(\frac{\alpha'_{I.v.}}{\alpha_k}\right)^{\frac{4}{3}} > 5 \cdot 10^4.$$

For smaller values of this grouping, $\alpha = \alpha_k$, and the heat transfer coefficient can be calculated from the much simpler formula (15).

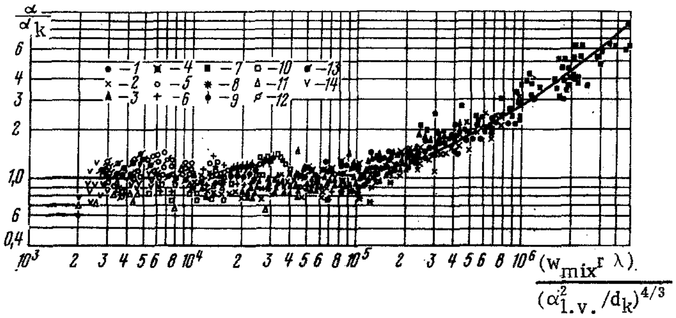

Fig. 14. Heat transfer for two-phase flow in tubes and ducts [46]. $\alpha/\alpha_k = \sqrt{(1 + 7 \cdot 10^{-9} (w_{mix} r \gamma'/q)^{3/2} (\alpha'_{l.v}/\alpha_k)^{2'})}$. The data shown are the following: 1) Tube, d = 8 mm (p = 5 to 31 atm, $q = 300 \cdot 10^3$ to $1200 \cdot 10^3$ kcal/$m^2 \cdot$ h) [46]; 2) Tube, d = 12 mm (p = 5 to 31 atm, $q = 300 \cdot 10^3$ to $1200 \cdot 10^3$ kcal/$m^2 \cdot$ h) [45]; 3) Tube, d = 18 mm (p and q as in 1 and 2) [45]; 4) Bundle, d_{equiv} = 5.46 mm (p = 11 to 31 atm, q as in 1) [46]; 5) Tube, d = 32 mm (p = 31.4 atm, $q = 100 \cdot 10^3$ to $450 \cdot 10^3$ kcal/$m^2 \cdot$ h) [17]; 6) Annular passage, d_{equiv} = 5.75; 3.87 mm (p = 50 atm, $q = 300 \cdot 10^3$ to $740 \cdot 10^3$ kcal/$m^2 \cdot$ h) [18]; 7) Tube, d = 5; 6.9 mm (p = 2 to 7 atm, $q = 200 \cdot 10^3$ to $1100 \cdot 10^3$ kcal/$m^2 \cdot$ h [48]; 8) Tube, d = 13.75 mm (p = 20 to 80 atm, $q = 70 \cdot 10^3$ kcal/$m^2 \cdot$ h) [49]; 9) Tube, d = 6 mm (p = 6 atm, $q = 700 \cdot 10^3$ kcal/$m^2 \cdot$ h) [50]; 10) Tube, d = 10 mm (p = 32 to 100 atm, $q = 190 \cdot 10^3$ to $340 \cdot 10^3$ kcal/$m^2 \cdot$ h) [16]; 11) Annular passage, d_{equiv} = 0.5; 1; 1.5 mm (p = 48 atm, $q = 390 \cdot 10^3$ to $1500 \cdot 10^3$ kcal/$m^2 \cdot$ h) [16]; 12) Tube, d = 4 mm (p = 3 to 9 atm, $q = 1600 \cdot 10^3$ to $4800 \cdot 10^3$ kcal/$m^2 \cdot$ h) [26]; 13) Tube, d = 8 mm (p = 170 atm, $q = 200 \cdot 10^3$ to $800 \cdot 10^3$ kcal/$m^2 \cdot$ h) [28]; 14) Bundle, d_{equiv} = 9.47 (p = 50; 100 atm, $q = 500 \cdot 10^3$ to $800 \cdot 10^3$ kcal/m $\cdot$ h) [18].

A comparison of the relation (28) with the experimental data for heat transfer with forced motion of a two-phase steam—water flow in tubes and ducts (Fig. 14) shows that the experimental points agree entirely satisfactorily with the calculational relation (28). The relation (28) is valid for sub-crisis heat transfer regimes and has been confirmed by experimental data in the following ranges of the parameters: $p = 2 \cdot 10^5$ to $170 \cdot 10^5$ N/m^2; $q = 0.8 \cdot 10^5$ to $6 \cdot 10^6$ W/m^2; w_{mix} = 1 to 300 m/sec. From an analysis of the relation (28) it is evident that it satisfies the specific transitions with respect to the circulation rate w_0, the specific heat flow q, the mixture velocity w_{mix}, and is devoid of the basic defects inherent in the 23 calculational formulas considered above.

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