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SOME PROBLEMS OF HEAT TRANSFER AND HYDRAULICS IN TWO-PHASE FLOWS

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Abstract—In the paper the results are presented of the study on heat transfer, hydraulic resistance and heat-transfer crisis for two-phase water–vapour and air–water flows in circular tubes, 5–34 mm i.d., the relative length being within 45–100 diameters. The tests were carried out within the following range of parameters

specific heat flux $q = 3 . 10^5\text{–}3 . 10^6$ kcal/m²h;
pressure in a test section $P = 1\text{–}36$ atm;
mass velocity $w\gamma = 180\text{–}5400$ kg/m²h;
steam quality $x = 0{\cdot}05\text{–}0{\cdot}999$.

The study has two aspects:

1. The heat-transfer mechanism and resistance in a two-phase flow have been studied. (Distribution of liquid and gas phases over the tube cross-section; mass transfer between the wall film and the liquid suspended in the flow core; measurement of liquid film thickness and the process of wave formation at the film surface.)
2. Heat transfer and hydraulic resistance have been studied to obtain particular engineering recomendations.

NOMENCLATURE

c,	mean concentration of suspended liquid in mixture;
d,	channel diameter;
l,	channel length;
P,	pressure;
w_0	velocity of circulation;
w_m,	mean velocity of mixture;
We,	Weber number;
x,	steam quality;
α,	heat-transfer coefficient;
ν,	kinematic viscosity;
δ,	film thickness;
σ,	surface tension;
ρ,	density.

AT PRESENT it appears reasonable to use a two-phase gas- or vapour–liquid flow of high gas content moving at high velocities (up to 100–150 m/s) for cooling highly forced devices. At some particular velocity mixture forms uniform emulsion (mist) in a flow core. Under these conditions the heat-transfer surface is permanently streamlined with a thin liquid film. The intense mass transfer between the wall and flow core provides high cooling rate.

Heat transfer in channel two-phase flows is known to be of complicated nature. In particular, the ordered motion affects heat transfer rate even when boiling occurs on walls. The velocity effect depends on the relationship between turbulent disturbances in a wall layer due to the ordered motion of the two-phase flow and vapour generation. The account of these effects is hampered by different flow regimes (nucleate, plug, slug, disperse-annular) of two-phase mixture in tubes. Hence the different nature of heat transfer changes may be supposed. It should also be noted that in a general case mean velocities of phases (liquid and gaseous) are different, due to the so-called "slip". That is why the true gas content (fraction of the section occupied with gas) differs from that calculated by flow rate.

The modern state of knowledge on heat transfer and hydraulics of two-phase systems is well reflected in review papers [1–3]. Of great variety of two-phase flow forms, the present paper deals mainly with a disperse-annular flow.

In the paper the results of investigation into heat transfer, hydraulic resistance and heat transfer crisis of two-phase vapour–water and air–water flows in circular tubes 5, 8, 12, 16, 18, 12, 9, 21, 27·7 and 34 mm i.d. are presented. The relative tube length ranged from 45 to 100 diameters.

The experiments were carried out within the following range of parameters:

specific heat flux $q = 3.10^5$–$3{\cdot}0.10^6$ kcal/m^{2}h;
pressure in working section $P = 1$–36 atm;
reduced velocity of two-phase flow $w_m = 5$–250 m/s;
velocity of circulation $w_0 = 0{\cdot}2$–6 m/s;
mass velocity $w\gamma = 180$–5400 kg/m^{2}h;
steam quality $x = 0{\cdot}05$–$0{\cdot}999$.

Two trends have been kept in researches:

1. The study of physical phenomena accompanying disperse-annular flow of gas–liquid mixtures in circular channels;

2. The experimental study of the problem under conditions close to reality to obtain particular engineering recommendations. I. B. Gavrilov, A. G. Kryuchkov and B. S. Filipovich collaborated in this investigation.

1. The mechanism of processes involved in disperse-annular flow was studied using air–water mixtures in horizontal channels at pressure and temperature close to atmospheric ones.

Liquid was injected into the flow through a sprayer located at the channel axis. In the flow the volumetric water content did not exceed 0·3 per cent. The weight phase ratio ranged from 0 to 3 kg/kg.

Main attention was given to the study of:

(1) distribution of liquid and gaseous phases over the channel section (liquid velocity and concentration fields);

(2) liquid exchange between solid film on the wall and suspended liquid in the flow core;

(3) thickness of liquid film and process of wave formation on its surface.

It was found that when loading the flow with liquid, the gas velocity profiles become less complete. In the flow core the density of two-phase mixture over the channel section may be considered stabilized at a distance of $l/d = 30$ from the location of liquid injection. Under the same conditions the amount of liquid in the wall film may also be considered constant along the channel length.

In case of suspended liquid in the flow core this points to the equal amount of liquid falling out onto the film surface and then detaching from it. It was established from the experiments with air–water mixtures at atmospheric pressure and isothermal conditions that at flow velocity of 60–180 m/s the amount of liquid falling out per unit channel surface G_l may be calculated by the formula

$$G_l = \alpha_0 \bar{c} w_m \tag{1}$$

where w_m is the mean velocity of mixture;
$\bar{c}$ is the mean concentration of suspended liquid in a flow;
α_0 is dimensionless coefficient of falling out.

It was found experimentally that α_0 depends on $Re_g = w_m d_e/\nu_g$ and dimensionless concentration of liquid $\bar{c}/\rho_l g$ (ν_g is the kinematic viscosity of air; ρ_l is liquid density at flow temperature). This relationship is presented in Fig. 1. In logarithmic coordinates the value of α_0 decreases with increasing loading. The effect of drop size on the amount of liquid falling out onto the liquid wall has not been studied particularly. The range of drop sizes depends on the flow velocity in the channel. Under test conditions the arithmetic-mean drop size ranged from 8 to 16 μ. In Fig. 2, as an example, the range of drop sizes is shown obtained by the method of light scattering at small angles under conditions

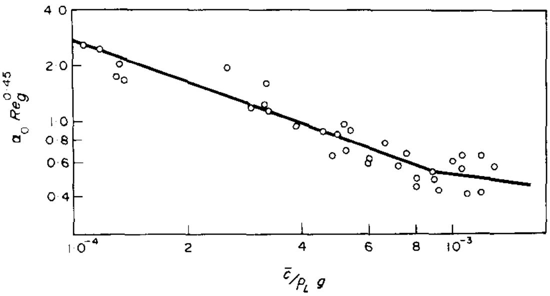

FIG. 1. $\alpha_0 Re_g^{0.75}$ vs dimensionless concentration of liquid suspension in a flow.

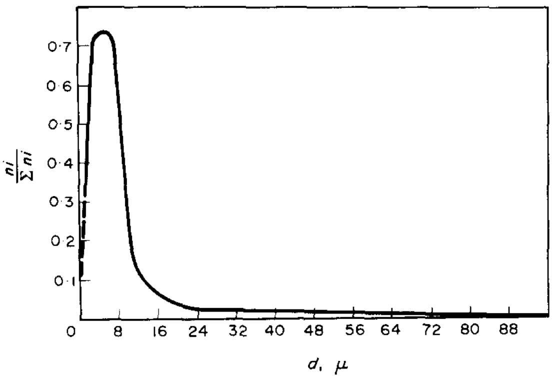

FIG. 2. Size distribution of drops in two-phase flow W_m = 106 m/s, G_l = 93 kg/sm^2

shown in the figure. Dimensionless distance $[y\sqrt{(\tau/\rho)}/v]$ (4) that can be covered by mean-size particles due to inertia in accordance with the Stokes resistance law was more than 30 at all times.

Treatment of velocity profiles of gas with suspended liquid using generally accepted methods for a one-phase flow in rough tubes (which we are not concerned with in this paper) shows that the turbulence constant $\varkappa$ for a two-phase flow is less than that for a one-phase flow.

The problem on liquid distribution between the core and wall layer, stability of motion of the layer with drop detachment from the film and their entrainment by the flow has no theoretical solution at present. Some experimental works [4, 5] are known presenting cinematographic and visual investigation of incipient drop entrainment from the film of liquid carried away by a turbulent flow along the plate.

It was pointed out that drop detachment resulted from long-wavelength disturbances appearing on the surface.

If the film thickness δ and mean velocity v in the film attain such a value that

$$Re^* = \frac{v\delta\rho_l}{\mu_l} < \frac{3{\cdot}8 . 10^3}{M} \quad (2)$$

where M is gas viscosity (μ_g) to liquid viscosity (μ_l) ratio, then the conditions required for detachment do not appear. Formula (2) was obtained experimentally [4] for air and various liquids, liquid being injected to the wall through slits in it. Liquid density ranged from 0·8 to 1·1 g/cm^3, surface tension changed by, approximately, 2 times (3·0–7·4).10^{-2} N/m; air velocity and viscosity ratio ranged from 50 to 200 m/s and from 20 to 2000, respectively. From formula (2) it follows, for example, that with constant viscosity and increasing gas velocity the film gets thinner, velocity in it increases, and liquid flow rate remains practically unchangeable.

In [4] it was mentioned that if a great amount of liquid is supplied to the plate surface, so that $Re_{pl} > Re^*$, the incipient entrainment (spraying) of liquid from the surface depends on the air velocity and liquid surface tension. For the given liquid, the air velocity for entrainment to occur appears to be the lower, the greater the liquid flow rate.

The researches of a disperse-annular channel flow carried out by the present and other authors [6, 7] showed that in a wall layer the liquid flow rate increased with its total content in the flow and decreased with increasing gas velocity.

For any total liquid flow rate there is some minimum liquid flow rate in a wall layer, which is attained at high flow velocities and remains constant with velocity increasing.

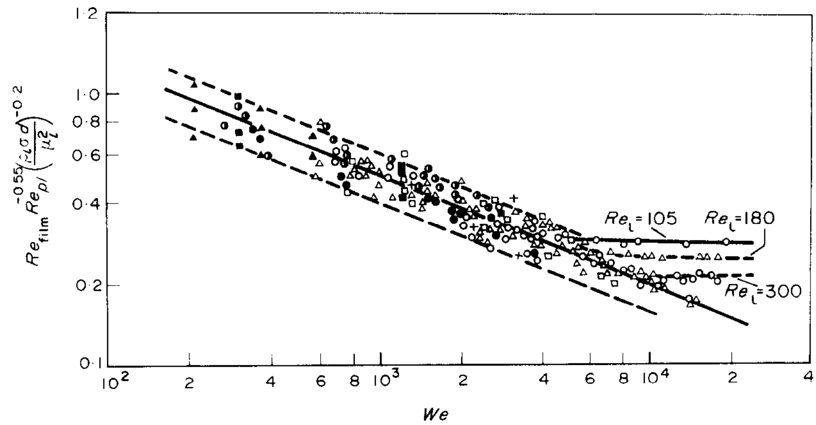

FIG. 3. $Re_{pl}Re_l^{-0.55}(\rho_l\sigma d/\mu_l^2 l)^{-0.2}$ vs Weber number for different channel diameters.

● – d = 16·0 mm	○ – d = 19·9 mm
△ – d = 33·8 mm	▲ – d = 10·8 mm [8]
■ – d = 26·0 mm [10]	◑ – d = 31·8 mm [7]
+ – d = 25·0 mm [9]	□ d = 27·7 mm

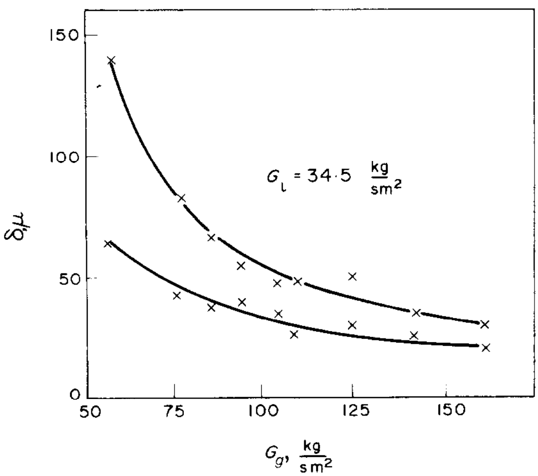

FIG. 4. Liquid film thickness on channel wall vs mass velocity of gas d = 27 mm, P = 1 atm.

The Re_{pl} number corresponding to the minimum flow rate is referred to as a limit one. According to the data of the present paper authors, for moving air–water mixtures in the channel the limit Reynolds number corresponds to Re^* calculated by formula (2) within ± 10 per cent.

For isothermal flow conditions in horizontal circular channels, experimental relationship is obtained

$$Re_{pl} = f\left(Re_l, We, \frac{\rho_l \sigma d}{\mu_l^2}\right)$$

$We = \rho_g w_g^2 d/\sigma$ is Weber number; σ, is surface tension; d is channel diameter; $Re_l = G_l/\pi d \mu_l g$ is Reynolds number calculated by the total liquid flow rate.

This relationship is presented in Fig. 3. Within ± 20 per cent it may be described by the formula

$$Re_{pl} = 8.2\, Re_l^{0.55} We^{-0.4} \left(\frac{\rho_l \sigma d}{\mu_l^2}\right)^{0.2}. \quad (3)$$

Formula (3) is valid while $Re_{pl} > Re^*$ (at flow temperature of 20°C $Re^* = 70$).

It is seen from formula (3) that the We number at which Re_{pl} is equal to 70, increases with the total liquid flow rate. In experiments the Weber number ranged from 200 to 20000.

Figure 4 shows variation of film thickness of liquid moving over the wall of a horizontal channel depending on the air flow rate, the water flow rate being the same. The measurements have been carried out on a lower generating line of the circular tube by the probe method. In the figure the lower line shows the thickness of a continuous liquid layer, and the upper one corresponds to 50 per cent the time of the probe and liquid contact. The change in the liquid layer thickness becomes less with increasing flow velocity. The disturbances of the film surface are retained even at high velocities, though being smoothed a little.

These investigations make it possible to suppose that increasing hydraulic losses in a disperse-annular flow are due to the acceleration losses of liquid detaching from the film surface and losses occuring when the flow core is moving relative to a greatly disturbed film surface, rather than due to decrease in the effective channel section because of the presence of liquid film on the wall. Total losses are determined experimentally. These losses may be divided into components only through calculation. Such a calculation has been performed. By experimental determination of the amount of liquid suspended in the flow and neglection of the phase slip in the flow core, the losses for longitudinal flow acceleration may be calculated.

The losses for acceleration of detaching liquid were calculated by assuming that the amount of liquid detaching from unit film surface and that falling out onto it are equal. The latter, as was already noted, has been determined experimentally (see formula (1)). The liquid is considered to be accelerated from the velocity on the film surface to the mean flow core velocity.

The calculations showed that the fraction of losses for acceleration of detaching liquid did not exceed 30 per cent of the total pressure losses. Dominating are the losses when flow core is moving relative to the "rough" film surface. This, probably, may explain some success of a homogeneous flow mode when calculating hydraulic losses and heat transfer for two-phase mixtures. It should be noted here that the resistance coefficient will be a complex function of the flow rate, viscosity and surface tension of liquid, since the film surface roughness depends on regime parameters and liquid properties.

Cinematographic investigation of the surface of a plate under heating located in the initial section of a two-phase flow showed that the heat transfer crisis and burnout of plate are associated with liquid film drying out on its surface. This may occur if the amount of heat supplied exceeds the heat spent for evaporation of falling out liquid. In this case the critical heat flux for vapour–water mixture may be calculated by formula

$$q_{cr} = \frac{\Delta i}{\pi d_e l_{cr}} \; G_{s0} \; 1 - e^{-\alpha_0 \frac{4 l_{cr}}{d_t}} \; + G_{pl0} \qquad (4)$$

where Δi is heat of vaporization; G_{s0} is amount of liquid suspended in a flow at channel entrance; l is channel length. Formula (4) includes two empirical values; α_0 is coefficient characterizing falling out; G_{pl0} is liquid flow rate corresponding to mixture conditions at the entrance.

In Fig. 5 the critical heat flux versus water mass flow rate is plotted for various vapour flow rates. In the section the mean pressure was 3 atm, the section being heated by alternating current. In this figure the lines are drawn by

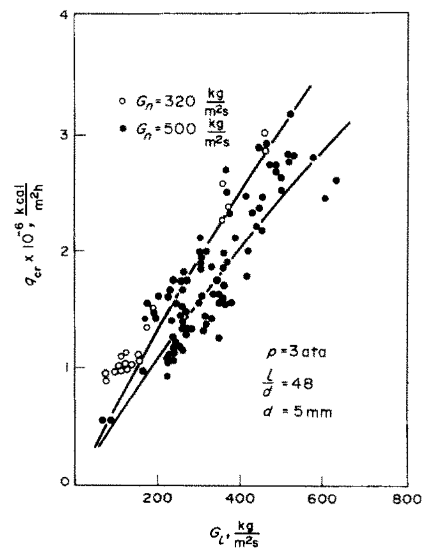

FIG. 5. q_{cr} vs water and vapour mass flow rate within entrance working section.

formula (4). It is seen from the figure that experimental points are concentrated, though with sufficient scatter, near the calculation lines.

A similar formula may be derived for a gas–liquid mixture. The liquid flow rate on the wall and hydraulic resistance being known, the convective heat-transfer coefficient for disperse-annular gas and vapour–liquid mixture flow may be calculated by usual methods of the boundary-layer theory. Such a calculation was carried out on the basis of the above data and compared with the experiments on heat transfer at pressures not higher than 3 atm.

These materials are not presented in the paper. But it should be noted that the results of calculation agree satisfactorily with experimental data.

2. In order to obtain particular recommendations for calculation, heat transfer and hydraulic resistance of a two-phase vapour–water flow in channels have been studied experimentally. The experiments were carried out using an open circulation contour consisting of the following main elements:

(1) working section (with electric heating);
(2) mixer designed to obtain vapour–water flow with required degree of dryness;
(3) system supplying water and vapour of given parameters to the mixer.

In the work wall temperature distribution has been studied and total pressure drop for given heat flux and definite two-phase flow parameters has been measured. For some flow regimes sharp deterioration of heat transfer was observed which nature was like that of crisis and proceeded, as a rule, at the end of the working section. The regimes of such a kind were not considered when the data were treated finally.

Though at present there are more than 20 relationships available in literature which are recommended to calculate boiling heat transfer under conditions of an ordered vapour–water flow in channels, all these formula are not universal and may be used only within particular range of parameters controlling heat transfer. Below two types of similar relationships are considered which, to the authors' mind, are mostly spread in heat-transfer literature.

The first type includes heat transfer equations, similar to the Martinelli equation, for calculation of the pressure drop.

The equations of this type introduced for tube flow calculations by Dengler and Addoms [11], Guerrieri and Talty [12] and quite recently verified by Collier, Lacey and Pulling [13, 14] for flows in annuli and longitudinally streamlined rod bundles, are of the form

$$\frac{\alpha}{\alpha_k} = A\left(\frac{1}{x_{tt}}\right)^n \tag{5}$$

$$\frac{1}{x_{tt}} = \left(\frac{x}{1-x}\right)^{0\cdot 9}\left(\frac{v''}{v'}\right)^{0\cdot 5}\cdot\left(\frac{\mu''}{\mu'}\right)^{0\cdot 8} \tag{6}$$

where α_k is heat transfer coefficient at forced liquid flow without boiling, kcal/m^{2}h°C; α_k is calculated by formula

$$\alpha_k = 0{\cdot}023.\frac{\lambda}{d}.Re^{0\cdot 8}.Pr^{0\cdot 4}. \tag{7}$$

The above relationship does not account for the effect of the specific heat flux q on the boiling heat transfer rate, which is its advantage. To account for the q effect, a correlation factor should be introduced which is difficult to be calculated.

In the Soviet Union the most known formulae presented by S. S. Kutateladze [15] and L. S. Sterman [16]† are derived by analyzing the experiments with no marked effect of vapour content on heat transfer (experiments being carried out at small x). In this case at constant pressure the heat-transfer coefficient was determined by the specific heat flux q and rate of circulation w_0.

The formula suggested by S. S. Kutateladze

† Sterman's formulae are analysed in detail by Collier. I. G. COLLIER. *A Review of Two-Phase Heat Transfer* (1935–1957).

for these conditions is of the form

$$\frac{\alpha_{pk}}{\alpha_k} = \sqrt{\left[1 + \left(\frac{\alpha_{00}}{\alpha_k}\right)^2\right]} \tag{8}$$

where α_k is heat-transfer coefficient at forced liquid motion at velocity w_0 without boiling, kcal/m²h°C; α_{00} is heat-transfer coefficient with boiling in channels for the region where circulation rate exerts yet no effect on boiling heat-transfer rate, kcal/m²h°C.

As V. M. Borishansky, A. P. Kozyrev and L. S. Svetlova [1, 19] showed

$$\alpha_{00} = 0{\cdot}7\alpha_{b0} \tag{9}$$

where

$$\alpha_{b0} = 3(P^{0\cdot 14} + 1{\cdot}83 . 10^{-4}p^2)q^{0\cdot 7}. \tag{10}$$

Relationship (8) is a rather satisfactory correlation of experimental data for the range of comparatively low vapour contents. At the same time, some works have appeared in literature recently presenting the results of heat-transfer investigations for the range of essential vapour contents (high-velocity flow).

It was established that experimental data on heat transfer to a high-velocity two-phase flow lie higher than the values recommended by formula (8). An increase in the heat-transfer coefficient (for the case of intensifying effect of vapour content) over that calculated by formula (8) depends both on the rate of circulation and vapour content of a two-phase flow.

Under these conditions the circulation rate w_0 cannot be used any longer as the only parameter accounting for the effect of forced motion on heat transfer.

The analysis of experimental data obtained for the region of higher vapour contents showed that they can be best generalized if the mean true velocity of vapour nucleus w_p'' (with regard for phase slip) is assumed to be the characteristic one. This indicates, that the heat transfer increase at high velocities of a two-phase flow results from turbulizing effect of vapour nucleus on a liquid wall layer. In a number of particular calculations, however, it is difficult to determine the true vapour velocity.

G. E. Kholodovsky [18], S. S. Kutateladze and M. A. Styrikovich [20] have showed that there is a single-valued relationship between the true vapour velocity and reduced two-phase flow velocity w_m in a wide range of gas–liquid flow parameters. In this connection the value

$$w_m = w_0\left[1 + \frac{\gamma' - \gamma''}{\gamma''}x\right]$$

attains a certain physical sense and may be used instead of w_p''.

In Fig. 6, as an example, a typical dependence

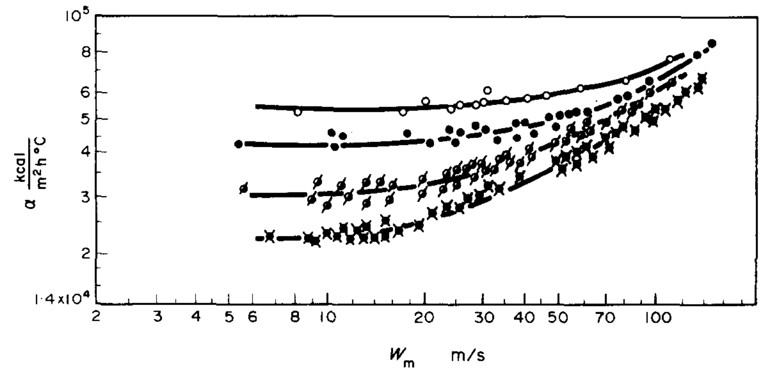

FIG. 6. Heat transfer coefficient vs mixture velocity and heat flux. P_{en} = 19 atm.
1 ✱ – q = 300 . 10³ kcal/m²h
2 ⌀ – q = 500 . 10³ kcal/m²h
3 ● – q = 800 . 10³ kcal/m²h
4 ○ – q = 1200 . 10³ kcal/m²h

of the heat-transfer coefficient on the specific heat flux q and mixture velocity w_m is plotted. Two limit regions may be distinguished in the dependence of the heat-transfer coefficient on q and w_m. At low velocities of a two-phase flow α is independent of w_m and is determined by the heat flux on the heating surface (and, probably, by circulation rate w_0). The authors failed to reveal the effect of w_0 on α because the effect of water convection on heat-transfer within the studied velocity range ($w_0 < 5$ m/s) proved to be insufficient as compared to the effect of heat flux ($q = 3.10^5$–$1{\cdot}2.10^6$ kcal/m^{2}h). This assumption is verified by the fact that calculations carried out in this region (low values of w_m) by formula (8) completely agreed with the experimental values of the heat-transfer coefficients (Fig. 7).

For high reduced velocities w_m heat transfer is no longer dependent on the specific heat flux q and when the pressure is prescribed it is determined only by w_m. This indicates that in this region the effect of vapour nucleus velocity on boiling heat transfer is dominating. Between these two regions transitions is smooth. The nature of transition depends on the simultaneous effect of the mixture velocity w_m and the specific heat flux q. An attempt was made to qualitatively describe this transition by interpreting the experimental heat transfer data in dimensionless coordinates of the form

$$\frac{\alpha}{\alpha_{pk}} = f\left(\frac{w_m r . \gamma'}{q}\right). \tag{11}$$

In this relationship α_{pk} is some conventional heat transfer coefficient calculated by formula (8). At present only very general observations can be made of the physical sense of the dimensionless complex $(w_m . r . \gamma')/q$. For a disperse-annular high-velocity two-phase flow in channel the intensity of the nucleus effect on the kind of motion of the wall liquid film and, hence, on heat-transfer is proportional to the nucleus velocity or, with regard for the data of [18, 20], to w_m. With such an approach the dimensionless complex $(w_m . r . \gamma')/q$ should be considered as the ratio of the value proportional to the mass velocity of liquid $w_m \gamma'$ to that of vapour q/r generated in the liquid wall layer.

The results of treatment of experimental data in coordinates of (11) are presented in Fig. 8. In addition to the data obtained by the authors of the present paper on vapour–water mixture boiling in tubes 5–18 mm i.d., for the pressure range from 2 to 40 atm the results presented in [21–29] should be generalized. In the graph both the points obtained from experiments showing the effect of vapour content on heat transfer [21, 24, 29] and the points from the experiments where no such effect was observed [22, 23, 25–27] are plotted. It follows from the graph that the experimental points are satis-

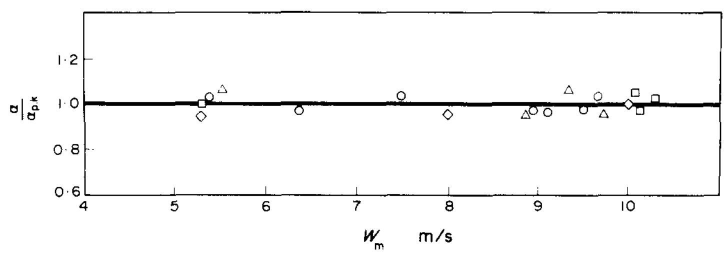

FIG. 7. Ratio α/α_{pk} for range of low mixture velocities.
○ – q = 300 000 kcal/m^{2}h
△ – q = 500 000 kcal/m^{2}h
□ – q = 800 000 kcal/m^{2}h
◇ – q = 1200 000 kcal/m^{2}h

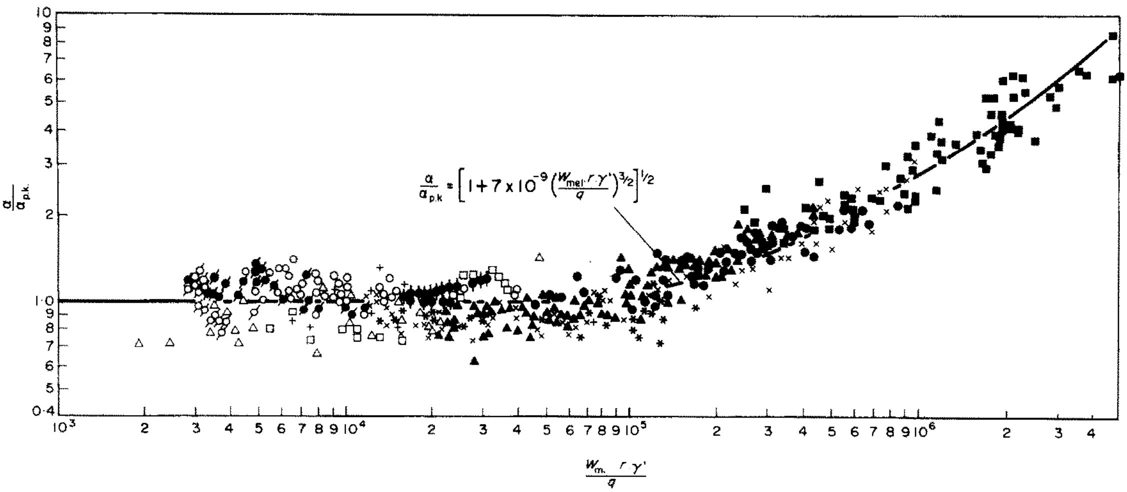

FIG. 8. Heat transfer from two-phase flow moving in tubes and channels (summary graph)

$$\frac{\alpha}{\alpha_{pk}} = \sqrt{\left[1 + 7.10^{-9}\left(\frac{W_m r\gamma'}{q}\right)^{\frac{3}{2}}\right]}.$$

● – tube ∅ 8 mm, $P = 5$–31 atm, $q = 300.10^3$–1200.10^3 kcal/m²h
× – tube ∅ 12 mm, $P = 5$–31 atm, $q = 300.10^3$–1200.10^3 kcal/m²h
▲ – tube ∅ 18 mm, $P = 5$–31 atm, $q = 300.10^3$–1200.10^3 kcal/m²h [21]
○ – tube 32 mm, $P = 31$–41 atm, $q = 100.10^3$–450.10^3 kcal/m²h [22]
\+ slit $d_e = 5{\cdot}76$ mm; 3·87 mm, $P = 50$–100 atm, $q = 300.10^3$–740.10^3 kcal/m²h [23]
■ – tube ∅ 5 mm; 6·9 mm; $P = 2$–7 atm; $q = 200.10^3$–1100.10^3 kcal/m²h [24]
* – tube ∅ 13·75 mm; $P = 20$–80 atm; $q = 70.10^3$–500.10^3 kcal/m²h [25]
✱ – tube ∅ 6 mm; $P = 7$ atm; $q = 700.10^3$ kcal/m²h [26]
□ – tube ∅ 10 mm; $P = 32$–100 atm; $q = 190.10^3$–340.10^3 kcal/m²h [27]
△ – slits $d_e = 0{\cdot}5$ mm; 1 mm; 1·5 mm; $P = 48$–144 atm; $q = 390.10^3$–1500.10^3 kcal/m²h [27]
∅ – tubè ∅ 4 mm; $P = 3$–9 atm; $q = 1600.10^3$–4800.10^3 kcal/m²h [28]
● – tube ∅ 8 mm; $P = 170$ atm; $q = 200.10^3$–800.10^3 kcal/m²h [29]

factorily concentrated around a single averaging curve, which may be described by the following empirical equation

$$\frac{\alpha}{\alpha_{pk}} = \sqrt{\left[1 + 7.10^{-9}\left(\frac{w_m . r . \gamma'}{q}\right)^{\frac{3}{2}}\right]}. \quad (12)$$

Relationship (12) shows that the motion of a vapour nucleus of two-phase flow which intensifies heat-transfer, starts in case when the dimensionless complex $(w_m . r . \gamma')/q > 5.10^4$.

This relationship is verified with experimental data in the following range of parameters:

$$P = 2\text{–}170\ \text{kg/cm}^2;$$

$$q = 70.10^3 + 5.10^6\ \text{kcal/m}^2\text{h};$$

$$w_m = 1\text{–}300\ \text{m/s}.$$

Thus, a single formula is obtained for calculation of heat transfer in tubes and chambers over the whole range of conditions without crises.

The experiments on hydraulic resistance were carried out with smooth tubes of stainless steel. The absolute roughness did not exceed 3μ with internal tube diameters equal to 8 and 18 mm. The tests run both with isothermal flow and with heat release within the working section ($q = 300000$–800000 kcal/m² h).

In literature for treatment of the experimental data on two-phase hydraulics it is accepted to present these data in dimensionless coordinate system

$$\frac{\Delta p_{2p}}{\Delta p_0} = f(x).$$

Here p_{2p} is friction resistance in case of two-phase flow of given vapour content in a channel; Δp_0 is friction resistance in case of liquid flow (with saturation parameters) of the same mass flow rate.

At zeroth vapour content ($x = 0$) the ratio $\Delta p_{2p}/\Delta p_0 = 1$ for any pressure, mass flow rates of heat-transfer agent and relative drop roughness, which favours the method. Its disadvantage is that for the second limit case ($x = 1$) the experimental data expressed in this coordinate system scatter depending on the pressure and relative roughness of the channel. Indeed, at $x = 1$ for a flow in rough tubes

$$\frac{\Delta p_{2p}}{\Delta p_0} = \frac{\Delta p''}{\Delta p_0} = \gamma'/\gamma'' = f_1(p)$$

for a flow in smooth tubes

$$\frac{\Delta p_{2p}}{\Delta p_0} = \frac{\Delta p''}{\Delta p_0} = \left(\frac{\mu''}{\mu'}\right)^{0\cdot25} \cdot \left(\frac{\gamma'}{\gamma''}\right) = f_2(p).$$

It should also be taken into account that as far as hydraulic roughness is considered, the case may occur with real constructions when at $x = 0$ the channel is smooth, and at $x = 1$ it is hydraulically rough.

Table 1

Pressure (atm)	1	30	70	100	150
Rough channel $\frac{\Delta p''}{\Delta p_0} = \frac{\gamma'}{\gamma''}$	1650	56	21	12·8	6·5
Smooth channel $\frac{\Delta p''}{\Delta p_0} = \left(\frac{\mu''}{\mu'}\right)^{0\cdot25}\left(\frac{\gamma'}{\gamma''}\right)$	745	35	14	9	4·9

The figures listed in Table 1 give visual picture of variation of the ratio $\Delta p''/\Delta p_0$ depending on pressure for vapour–water mixture.

Thus, the authors think that the system of coordinates $\Delta p_{2p}/\Delta p_0$ is not quite successful to correlate the experimental data. In this connection in the present paper to correlate the experimental data the following system of coordinates is suggested

$$F(x) = \frac{\Delta p_{2p} - \Delta p_0}{\Delta p'' - \Delta p_0}. \qquad (13)$$

The suggested system of coordinates has the advantage that for any p, $w\gamma$ and $\bar{\nabla}$ the limit values of the complex $(\Delta p_{2p} - \Delta p_0)/(\Delta p'' - \Delta p_0)$ are constant.

Thus at $x = 0$

$$\frac{\Delta p_{2p} - \Delta p_0}{\Delta p'' - \Delta p_0} = \frac{\Delta p_0 - \Delta p_0}{\Delta p'' - \Delta p_0} = 0$$

at $x = 1$

$$\frac{\Delta p_{2p} - \Delta p_0}{\Delta p'' - \Delta p_0} = \frac{\Delta p'' - \Delta p_0}{\Delta p'' - \Delta p_0} = 1.$$

In Fig. 9 the experimental data obtained in the present work are compared in coordinates $(\Delta p_{2p} - \Delta p_0)/(\Delta p'' - \Delta p_0)$ and x. It is seen from the graph that all the experimental points independent of pressure, channel diameter and specific heat flux are satisfactorily concentrated around a single averaging curve. The presence of a single curve of reduced friction losses $F(x)$ allows development of a relatively simple procedure to calculate the hydraulic resistance coefficient for a two-phase flow in channels.

To calculate the hydraulic losses for friction in case of a two-phase flow in tubes, it is generally accepted to use the Darcy–Weissbach relationship written in the form

$$\Delta p_{2p} = \xi_m \frac{\gamma_m w_m^2}{2g} \frac{l}{d} = \xi_m \frac{\gamma' w_0^2}{2g} \frac{l}{d} \times \left[1 + \left(\frac{\gamma'}{\gamma''} - 1\right)x\right] \ldots \qquad (14)$$

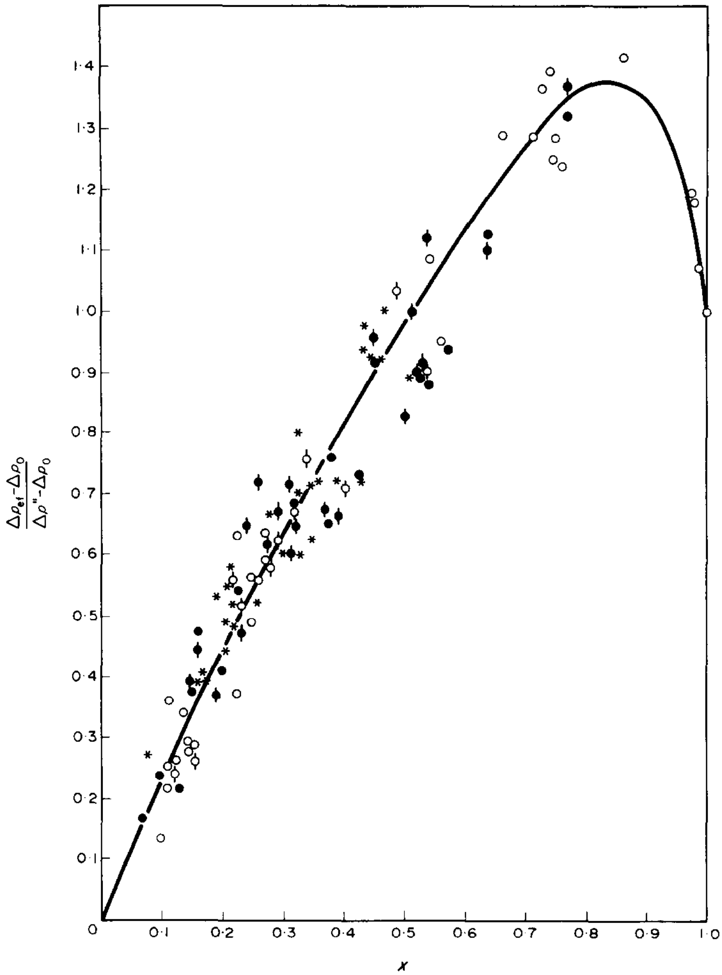

FIG. 9. Reduced resistence vs vapour content.

○ tube ∅ 12 mm; $q = 0$
∅ tube ∅ 12 mm; $q = /300\text{–}800 . 10^3$ kcal/m²h
● tube ∅ 18 mm; $q = 0$
● tube ∅ 18 mm; $q = /300\text{–}800 . 10^3$ kcal/m²h
* tube ∅ 8 mm; $q = /300\text{–}800 . 10^3$ kcal/m²h

Definition of the reduced resistance (13) yields

$$\Delta p_{2p} = \Delta p_0 \left[1 + \left(\frac{\Delta p''}{\Delta p_0} - 1\right) . F(x)\right]. \quad (15)$$

By equating relations (14) and (15), obtain an expression relating the hydraulic resistance coefficients for two- and one-phase flows

$$\xi_m = \xi' \frac{1 + \left(\frac{\Delta p''}{\Delta p_0} - 1\right) . F(x)}{1 + \left(\frac{\gamma'}{\gamma''} - 1\right) x}. \quad (16)$$

For a two-phase flow moving in smooth tubes equation (16) may be written in the form

$$\xi_m = \xi' \frac{1 + \left[\left(\frac{\mu''}{\mu'}\right)^{0\cdot 25}\left(\frac{\gamma'}{\gamma''}\right) - 1\right].F(x)}{1 + \left[\left(\frac{\gamma'}{\gamma''}\right) - 1\right]x}. \quad (17)$$

For rough tubes

$$\xi_m = \xi' \frac{1 + \left[\left(\frac{\gamma'}{\gamma''}\right) - 1\right].F(x)}{1 + \left[\left(\frac{\gamma'}{\gamma''}\right) - 1\right]x}. \quad (18)$$

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QUELQUES PROBLEMES DE TRANSFERT THERMIQUE ET D'HYDRAULIQUE DANS LES ECOULEMENTS BIPHASIQUES

Résumé—On présente dans cet article les résultats de l'étude du transfert thermique, de la résistance hydraulique et de la crise du transfert thermique pour des écoulements biphasiques eau–vapeur et air–eau dans des tubes circulaires de 5–34 mm de diamètre intérieur, la longueur relative variant de 45 à 100 diamètres. Les essais ont été faits dans le domaine suivant des paramètres:

flux thermique spécifique $q = 3.10^5$–3.10^6 kcal/m^2 h

pression dans une section de mesure $P = 1–36$ atm
vitesse massique $W_\gamma = 180–5400$ kg/m² h
qualité de la vapeur $x = 0{,}05–0{,}999$
L'étude a deux aspects
1. On a étudié le mécanisme de transfert thermique et la résistance dans un écoulement biphasique. (Distribution des phases liquide et gazeuse dans la section droite du tube; transfert massique entre le film pariétal et le liquide en suspension dans le noyau de l'écoulement; mesure de l'épaisseur du film liquide et processus de formation d'ondes à la surface du film).
2. On a étudié le transfert thermique et la résistance hydraulique pour obtenir des recommandations particulières technologiques.

EINIGE PROBLEME DES WÄRMETRANSPORTES UND DES HYDRAULISCHEN WIDERSTANDES BEI ZWEIPHASENSTRÖMUNGEN

Zusammenfassung—In dieser Arbeit werden die Ergebnisse einer Untersuchung über den Wärmeübergang, den hydraulischen Widerstand und die Wärmeübergangskrisis bei Zweiphasen-Wasser-Dampf- und Luft-Wasserströmungen in Kreisrohren von 5–34 mm Durchmesser und einer auf den Rohrdurchmesser bezogenen Länge von 45–100 vorgestellt.

Die Versuche wurden über den folgenden Parameterbereich durchgeführt:

Wärmestromdichte $q = 3 \cdot 10^5–3 \cdot 10^6$ kcal/m²h;
Druck in der Teststrecke $p = 1–36$ ata;
Massenstromdichte $W_j = 180–5400$ kg/m² h;
Dampfgehalt $x = 0{,}05–0{,}999$,

Die Untersuchung erfolgt nach zwei Gesichtspunkten:

1. Den Wärmeübergangsmechanismus und den Widerstand in einer Zweiphasenströmung zu erhalten. (Verteilungen der Flüssig- und Gasphase über den Rohrquerschnitt; Massentransport zwischen dem Wandfilm und der in der Kernströmung verteilten Flüssigkeit; Messung der Flüssigkeitsfilmdicken und der Vorgänge bei Wellenbildung an der Filmoberfläche.)
2. Den Wärmeübergang und den hydraulischen Widerstand im Hinblick auf spezielle Empfehlungen für den Ingenieur zu betrachten.

НЕКОТОРЫЕ ВОПРОСЫ ТЕПЛООБМЕНА И ГИДРАВЛИКИ ПРИ ТЕЧЕНИИ ДВУХФАЗНЫХ ПОТОКОВ

Аннотация—В статье излагаются результаты исследования теплоотдачи, гидравлического сопротивления и кризиса теплообмена при движении двухфазного пароводяного и воздуховодяного потоков в круглых трубах диаметром 5–34 мм. Относительная длина труб изменялась в пределах от 45 до 100 калибров. Опыты проводились в следующем диапазоне параметров: удельная тепловая нагрузка $q = 3.10^5–3.10^6$ ккал/м² час; давление в рабочем участке $P = 1–36$ ата; весовая скорость $w\gamma = 180–5400$ кг/м² час; весовое паросодержание $x = 0{,}05–0{,}999$.

Исслкдования проводились в двух направлениях:

(1) Изучался механизм теплообмена и сопротивление в двухфазном потоке (распределение жидкой и газообразной фазы по сечению канала; обмен жидкостью между пленкой на стенке и взвешенной жидкостью в ядре потока; измерение толщин жидкой пленки и характера волнообразования на ее поверхности).

(2) Изучалась теплоотдача и гидравлическое сопротивление в целях получения конкретных инженерных рекомендаций.