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HEAT TRANSFER AND HYDRAULIC RESISTANCE DURING CONDENSATION OF STEAM IN A HORIZONTAL TUBE AND IN A BUNDLE OF TUBES

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Аннотация—В статье дается теоретический расчет теплообмена при конденсации пара в трубе на основе аналогии Рейнольдса между гидродинамическим сопротивлением и теплообменом. Приводятся экспериментальные результаты, полученные авторами при конденсации водяного пара в трубах диаметром 18 мм и длиной 12 м при давлениях до 90 бар. Теоретические и экспериментальные результаты согласуются удовлетворительно. Кроме того излагаются оригинальные экспериментальные данные о гидравлическом сопротивлении при конденсации водяного пара внутри труб. Приводятся экспериментальные результаты по теплообмену при течении конденсирующегося пара в пучке труб.

NOMENCLATURE

h, local heat-transfer coefficient;
ρ, density;
d, tube diameter;
l, tube length;
K, thermal conductivity;
g, gravitational constant;
Re, Reynolds number;
Pr, Prandtl number;
Nu, Nusselt number;
t, temperature;
Δt, difference of temperature;
c, heat capacity;
$\bar{q}$, average specific thermal load on the inner surface of the tube;
Q, thermal load;
F, heat-transfer surface;
G, mass flow;
p, pressure;
Δp, pressure difference;
ξ, coefficient of hydraulic resistance to the flow;
τ, shear force;
x, steam content.

THE PRESENT paper is in a sense the development of the work [1] in which on the basis of analogy between heat transfer and hydraulic resistance for the case of steam condensation inside a tube, the following equation was obtained

$$h = h_0 \sqrt{\left(\frac{\rho}{\rho_m}\right)} \tag{1}$$

in which ρ and ρ_m are the densities of condensate and steam-and-water mixture, respectively; h_0 is the heat-transfer coefficient from liquid (condensate) to the wall of the tube in that part of the tube where the process of condensation had been completed and where liquid (condensate) flows through the section of the tube. The value of h_0 can be calculated using the well known methods for one-phase flow, for example for ordinary liquids using Mikheev's formula [2]

$$\frac{h_0 \cdot d}{K} = 0{\cdot}021\, Re^{0{\cdot}8} \,.\, Pr^{0{\cdot}43} \left(\frac{Pr_f}{Pr_w}\right)^{0{\cdot}25} \tag{2}$$

where subscripts f and w denote that the value in question relates to the temperature of the stream or of the wall, respectively.

Formula (1) determines the local value of the heat-transfer coefficient. At the entry of the tube where $\rho_m = \rho_v$ the heat-transfer coefficient according to this formula is: $h = h_0 \sqrt{(\rho/\rho_v)}$. At the end of the condensation area $\rho_m = \rho$ and h turns into h_0.

Dr. D. A. Labuntsov has shown through direct calculation that in connection with formula (1) the average value of heat-transfer coefficient is in this case

$$\bar{h} = \frac{h_0}{2}\left[1 + \sqrt{\left(\frac{\rho}{\rho_v}\right)}\right]. \tag{3}$$

The calculation ran like this: with given steam content the ratio of the densities of liquid and homogenized steam-and-water mixture which forms a part of formula (1), can be expressed through the following formula:

$$\frac{\rho}{\rho_m} = 1 + \frac{\rho - \rho_v}{\rho_v} x = 1 + \frac{\Delta\rho}{\rho_v} x. \tag{a}$$

Therefore it follows from formula (1)

$$h = h_0 \sqrt{\left(1 + \frac{\Delta\rho}{\rho_v} x\right)} \equiv h_0 y^{\frac{1}{2}}, \tag{b}$$

in which

$$y = 1 + \frac{\Delta\rho}{\rho_v} x. \tag{c}$$

The average value of the heat-transfer coefficient in section 1–2 of the tube can be expressed by the equation:

$$\bar{h} = \frac{Q_{1-2}}{F_{1-2} \cdot \Delta\bar{t}_{1-2}}, \tag{d}$$

in which Q, $\Delta\bar{t}$, F are the thermal load, average temperature drop and the heat-transfer surface on the said area, respectively. Thermal load can be calculated through the heat balance equation:

$$Q_{1-2} = G_m \cdot L \cdot (x_1 - x_2) = G_m \cdot L(y_1 - y_2)\frac{\rho_v}{\Delta\rho}, \tag{e}$$

in which G_m, L are the mass flow of steam-and-water mixture and latent heat of evaporation, respectively. The value of $\Delta\bar{t}F$ in section 1–2 can be obtained from the heat-transfer equation:

$$h \cdot \Delta t \cdot \mathrm{d}F = -G_m \cdot L \cdot \mathrm{d}x, \tag{f}$$

in which x is the dryness fraction of the mixture as before.

In connection with formula (c) we have

$$\mathrm{d}x = \frac{\rho_v}{\Delta\rho} \cdot \mathrm{d}y. \tag{g}$$

Further substituting (b) and (g) in (f) and integrating over the considered part of the tube we obtain:

$$\Delta\bar{t}_{1-2} \cdot F_{1-2} = \frac{2 \cdot G_m \cdot L \cdot \rho_v}{h_0 \cdot \Delta\rho}(y_1^{\frac{1}{2}} - y_2^{\frac{1}{2}}). \tag{h}$$

Finally the substitution of (e) and (h) in (d) yields:

$$\bar{h} = \frac{h_0(y_1^{\frac{1}{2}} + y_2^{\frac{1}{2}})}{2} = \frac{h_1 + h_2}{2}. \tag{i}$$

At the same time we might mention here the character of the change in the heat-transfer coefficient in this case along the condensation area. For this purpose we determine from equation (b) the change of the coefficient along the differential element of the tube $\mathrm{d}l$:

$$\frac{\mathrm{d}h}{\mathrm{d}l} = \frac{h_0 \cdot \Delta\rho}{2\rho_v \sqrt{[1 + (\Delta\rho/\rho_v) x]}} \cdot \frac{\mathrm{d}x}{\mathrm{d}l}. \tag{k}$$

On the other hand we find from equation (f):

$$\frac{\mathrm{d}x}{\mathrm{d}l} = -\frac{\pi \cdot \mathrm{d}h \cdot \Delta t}{G_m \cdot L}. \tag{l}$$

Substituting from (l) and (b) into (k) we obtain:

$$\frac{\mathrm{d}h}{\mathrm{d}l} = -\frac{\pi \cdot \mathrm{d}h_0 \cdot \Delta\rho \cdot \Delta t}{2 \cdot L \cdot G_m \cdot \rho_v}. \tag{m}$$

In the final equation all the values on the right-hand side except Δt are constant over the whole length of the tube. Owing to that, in the case when the temperature drop Δt is also constant along the length of the tube, the heat-transfer coefficient decreases uniformly along the length of the tube from the maximum value at the beginning to h_0 at the end of the condensation area. Generally, the variation of the local value of the heat-transfer coefficient h along the length of the tube can differ from the straight line distribution only to the extent of the change of the temperature difference. But even in those cases when the temperature difference is not constant along the length of the tube, the heat-transfer coefficient is changing in such a way that its average value [see equation (d)] can still be expressed by equivalent formulas (3) and (i).

This conception was formulated in 1958 [3, 4] and published in [1]. About the same time and quite independently from the first conception, a conception similar in its idea was put forward by Dr. Akers [5, 6]. In that conception as in the previous one, the investigation was carried out for the case of steam condensation inside a cylindrical tube with the flow symmetrical about the axis, i.e. when the gravitational force does not manifest itself. But the regime of flow taken as a model was one with turbulent film on the wall of the tube and turbulent flow of steam in the core. The analysis of the model given in [5] amounts to the following. During the process of steam condensation the main thermal resistance in the model is caused by the film of the liquid and is determined by its hydrodynamics. The latter in its turn is determined mainly by the dynamic impact of the flow of steam moving in the core, upon the film, i.e. by the tangential stress on the boundary between the flow of steam and the film. Therefore in this model "theoretically the vapour core could be replaced by a stream of liquid whose flow would produce the same value for the shear stress" [5].

Thus the system under consideration which has a turbulent film of liquid on the wall of the tube and turbulent flow of steam in the core can, according to the condition of heat transfer, be represented by an equivalent stream of liquid (condensate) with a mass flow G_E equal to the sum of the actual flow of condensate G_L through the given section of the tube and the flow G'_L equivalent in this sense to the actual mass flow of steam G_v through the same section of the tube:

$$G_E = G_L + G'_L. \tag{4}$$

Accordingly, the heat-transfer coefficient in the given section of the tube during steam condensation in it can be determined according to Akers by an ordinary formula of the type (2) by substituting the equivalent flow as given by equation (4) in the Reynolds number. The mass flow G'_L which is a part of formula (4) is calculated by the author on the assumption that the thickness of the film of liquid on the wall of the tube is comparatively small, and therefore the cross section through which the flow of steam moves is approximately equal to the full section of the tube. Under this condition in the case under consideration we can write that

$$G'_L = G_v \left(\frac{\rho}{\rho_v}\right)^{0\cdot5} \cdot \left(\frac{\xi_v}{\xi_1}\right)^{0\cdot5} \tag{5}$$

where ξ_v, ξ_1 are the coefficients of hydraulic resistance to the flow of steam and to the equivalent flow of condensate. On taking the approximately equal values of the said coefficients instead of equation (5) we obtain

$$G'_L = G_v \left(\frac{\rho}{\rho_v}\right)^{0\cdot5}. \tag{6}$$

It is obvious that Akers' theory is based entirely on the relationship between heat transfer and hydrodynamic friction. At the same time, since the details of this relationship were omitted in the given approach to the problem, the theory is, to a very considerable extent, accepted intuitively rather than as a result of a theoretical proof. Dr. Akers' theory differs from the above conception mainly through the choice of the model of steam-and-liquid mixture flow in the tube, and it results in substantially different conclusions. If, on the other hand, we assume in the Akers method that a homogenized steam-and-water mixture moves in the core, the magnitude of the equivalent flow will be determined not by (4) and (6) but by the following equation:

$$G_E = G_m \left(\frac{\rho}{\rho_m}\right)^{0\cdot5} = G_0 \left(\frac{\rho}{\rho_m}\right)^{0\cdot5} \tag{7}$$

where G_m, G_0 are mass flows of the steam-and-water mixture in the given section and that of the condensate after passing the condensation area, respectively. Substituting (7) in (2) we obtain:

$$h = h_0 \left(\frac{\rho}{\rho_m}\right)^{0.4} \tag{8}$$

which almost entirely agrees with equation (1). The difference between equations (8) and (1) is caused by the difference in the estimation of the values of the hydraulic resistance coefficients adopted in these equations.

Having adopted the essence of Akers' method, let us turn again to its results. They are related to the model in which the condensate moves in the turbulent film on the wall of the tube, and the steam in the core of the flow. It is worth mentioning that these results assume a concrete form only in the starting region when it can be assumed that the section occupied by the steam does not differ appreciably from the tube section. Actually, in the adopted model, as the condensation of steam progresses, the thickness of the liquid film on the tube wall increases reaching at the end of the condensation zone the center of the tube. Therefore, in the above approach the change in the thickness of the condensate film along the whole length of the tube should certainly have been taken into account. But since it was not, it is necessary to bear in mind that the results obtained by Akers apply only to the starting region of the tube where the thickness of the film is indeed relatively small. On the other hand, the calculation of the change in the thickness of the condensate film along the whole length of the wall (the entire zone of condensation) in the above model complicates the problem to such an extent that it becomes virtually impossible to reduce it to a definite result in a clear form.

We have also conducted an experimental investigation of the intensity of heat transfer in the considered case. The arrangements of the experimental installations were given in [1, 7]. The condensation of saturated steam was conducted inside a tube made of stainless steel inclined at 1° to the horizontal in the direction of the flow. The tube was cooled by boiling water and on account of that the difference in the temperature of condensing steam and boiling water, and hence the temperature drop between the steam and inner wall along the whole length of the tube did not change. With a given pressure of condensing steam this temperature drop was regulated by altering the pressure of the secondary steam. Regulation was achieved with the help of a cooler on the outward surface of which the condensation of secondary steam was carried out and the condensate flowed back into the boiling water which cooled the experimental tube. The heat absorption of this refrigerator plus the outside heat emission of the installation determined the total thermal load of the experimental tube. This load could also be determined by the mass flow of the steam and steam-and-water mixture in the tube and the change in the heat content of the mixture. The difference between these values was, in the experimental conditions, within ± 6 per cent. Therefore we may agree that thermal load in the experiments was measured by sufficiently reliable means. But in processing the results of the experiments the use was made only of the data for the mass flow and the change of its heat content.

In such experiments the measurement of the wall temperature is a most complicated thing. In our tests the tube itself was used as a resistance thermometer [8]. We measured the mean temperature of the metal of the wall. The inner surface temperature was calculated afterwards, the heat conductivity of the metal and measured thermal load being taken into account. Since the experimental tube was made of stainless steel, this method of measuring the temperature of the wall secured sufficiently good sensitivity and precision. Nevertheless during the initial period of the experimental work this method often failed to produce reproducible results [1]. Therefore, a vessel of carbon steel which contained boiling water during the first series of experiments was replaced by a vessel of stainless steel; the electric insulation of the experimental tube from the above vessel was improved as far as possible; during the initial period the vessel was filled occasionally with water from the condensate line of an electric power plant. This was discontinued and steam was condensated directly in the vessel placed on the cooler already mentioned. After this there were no difficulties with the measurement by the described method, and the maximum error in determining the mean value of the heat-transfer coefficient was within ± 10 per cent.

The experiments were conducted with three types of tubes 2500-mm long and $13 \times 1 \cdot 5$ mm; $16 \times 1 \cdot 5$ mm and $20 \times 1 \cdot 5$ mm in diameter, with condensing steam pressure 12·2, 24·5, 59 and 88 bar. During the tests of each of the above mentioned tubes regimes were used with complete condensation of steam, i.e. with the change of steam content from $x_1 = 1 \cdot 0$ in the entrance section of the tube to $x_2 = 0 \cdot 0$ at the exit; regimes with $x_1 = 1 \cdot 0$ and $x_2 = 0 \cdot 20$–$0 \cdot 50$ and regimes with $x_1 = 0 \cdot 3$–$0 \cdot 6$ and $x_2 = 0 \cdot 0$. In addition experiments were conducted with a tube 12·00-mm long and $16 \times 1 \cdot 5$-mm dia., pressures 59 and 88 bar and complete steam condensation [1]. During the experiments the average heat flux changes from $0 \cdot 152 \times 10^6$ W/m^2 to $1 \cdot 58 \times 10^6$ W/m^2. In these experiments heat-transfer coefficients under complete steam condensation reached 38×10^3 W/m^2 degree.

As a result of these experiments we have obtained the average values of heat-transfer coefficients. In accordance with the above equations (l) and (i) these coefficients are represented by the graphs I–IV in Fig. 1 in the logarithmic scale in the following form

$$\bar{K} \equiv \frac{\overline{Nu} \, . \, Pr^{-0 \cdot 43}}{\frac{1}{2}\left[\sqrt{(\rho/\rho_m)_1} + \sqrt{(\rho/\rho_m)_2}\right]} = f(Re), \qquad (9)$$

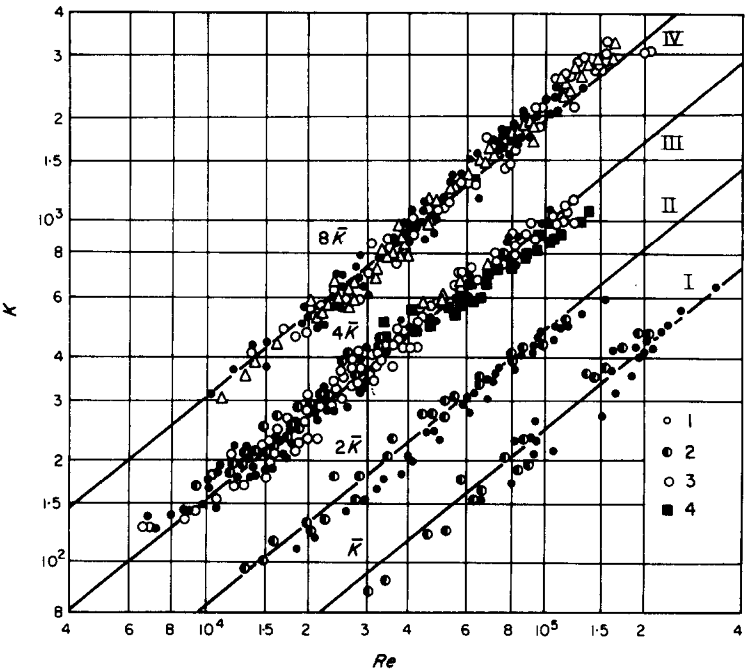

FIG. 1. Relation $\bar{K} = f(Re)$ based on the experimental values of the heat-transfer coefficient for the pipes 2500 mm long and the following diameters: 1. $d = 16 \times 1 \cdot 5$ mm; 2. $d = 20 \times 1 \cdot 5$ mm; 3. $d = 13 \times 1 \cdot 5$ mm; 4. for the pipe 11 950 mm long, $d = 16 \times 1 \cdot 5$ mm.

where values ρ/ρ_m obtained from equation (a) relate to the entrance and exit sections of the tube, respectively, Re is the Reynolds number calculated from condensate flow with complete condensation, $\overline{Nu} = [(\bar{h}.d)/K]$ is the Nusselt number, calculated by the average value of heat-transfer coefficient. All physical constants which are parts of formula (9) are related to the saturation temperature. For clarity the ordinate in graph I is $\overline{K}$, in graph II, $2\overline{K}$ and so on. The averaging curves on these charts have the same inclination and the vertical distances between them correspond to the variance in the said values, i.e. ln 2. Therefore, actually graphs II–IV coincide exactly with chart I.

The tangent of the angle of inclination of graphs I–IV to the horizontal line is equal to 0·8 and hence the heat-transfer coefficient is proportional to the value of $Re^{0\cdot 8}$. The constant in the type (1) formula can be obtained from graph I and is equal to 0·024. Therefore the experimental data obtained can be expressed by the formula:

$$Nu = 0{\cdot}024\, Re^{0\cdot 8}\,.\,Pr^{0\cdot 43}\,\frac{[\sqrt{(\rho/\rho_m)_1} + \sqrt{(\rho/\rho_m)_2}]}{2}. \tag{10}$$

As we see in Fig. 1 the discrepancy between the experimental data and the latter formula is within ± 15 per cent. The calculated results can also be compared with the experimental data for the local values of heat-transfer coefficient. In the given calculations the value of the latter can be expressed by equation (1). Experimental data of the local values were obtained by Miropolsky [9]. The cooling of steam-and-water mixture was carried out, and the determination of the heat-transfer coefficient was conducted along the 50–55-mm long part of the tube. The steam content of the stream changed very little within these areas, so that it is possible to consider the results obtained as local. Tubes of copper and carbon steel with 8-mm i.d. were used during these experiments. The pressure in the flow path changed from 7 to 200 bar, relative weight of steam content from 1·0 to 0·0m, and the speed of flow from 400 to 2000 kg/m^{2}s. The results of the experiments which are rather voluminous are given in Fig. 2 (in logarithmic coordinates) where the value

$$K = \frac{Nu}{Pr^{0\cdot 43}\,(\rho/\rho_m)^{0\cdot 5}}$$

is marked on the ordinate in accordance with equation (1). The experimental points are represented quite satisfactorily by a straight line with a slope coefficient 0·8, and the scattering

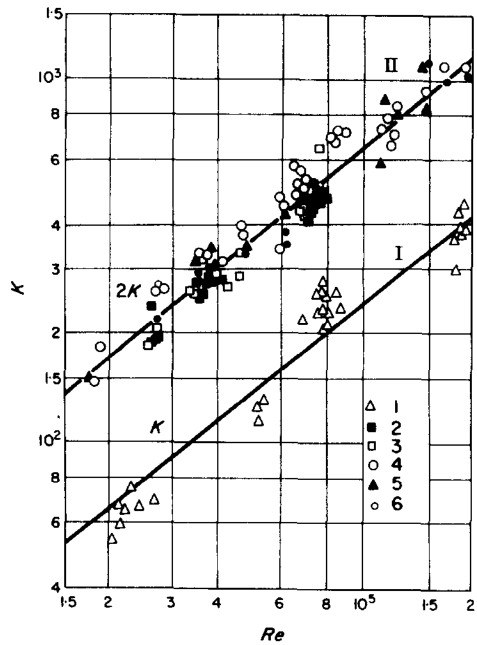

FIG. 2. Relation $K = f(Re)$ based on experimental data [9] for pipes with $d = 8$ mm and the following lengths: 1. l = 135 mm; 2. l = 28 mm; 3. l = 142 mm; 4. l = 32 mm; 5. l = 80 mm; 6. l = 135 mm (I, carbon steel, II, copper).

of experimental data does not exceed ± 20 per cent. Thus a satisfactory correlation between the calculated and experimental data exists also in this case. The only specific factor is, that the proportionality coefficient in the formula of type (2) is equal to 0·024 for the experiments with a steel tube and 0·032 for a copper tube.

To make the picture complete let us consider the direct connection between the heat transfer and hydraulic resistance in the case considered on the basis of the experimental data. We present for this purpose the local heat-transfer coefficient in the following form [1]:

$$h = EgC_p\sqrt{\left(\frac{\xi_1\rho}{8}\right)}\sqrt{(\tau_m)}. \tag{11}$$

Further, assuming with a certain degree of approximation that the value of ξ_1 is constant along the whole length of the tube and is equal to a certain average value, we find from formula (11) that the average values are related by the following formula

$$\bar{h} = EgC_p\sqrt{\left(\frac{\xi_1\rho}{8}\right)}\overline{\sqrt{(\tau_m)}}. \tag{12}$$

From the corresponding measurements during the experiments were obtained the average heat-transfer coefficient $\bar{h}$ and the hydraulic resistance ΔP_f related to the average specific friction force on the wall along the length of the tube by the following formula:

$$\bar{\tau}_m = \frac{d}{4l}\Delta P_f. \tag{13}$$

Therefore we will have to substitute the average value of $\overline{\sqrt{(\tau_m)}}$ for the average value of $\bar{\tau}_m$ in order to be able to utilize formula (12). Under the conditions of the experiments the temperature difference between the wall and condensate was constant over the whole length of the tube. Therefore, as it was already mentioned in connection with equation (m), the value of the heat-transfer coefficient changed linearly along the pipe. It follows from equation (11) that the value of $\sqrt{(\tau_m)}$ also varied as a straight line. Therefore, denoting the friction force in the entrance section of the tube by τ_{m_1}, and by τ_{m_2} the friction force in the exit section of the tube, and designating the coordinate of the entrance section by z, we obtain:

$$\sqrt{(\tau_m)} = (1-\theta)\sqrt{(\tau_{m_1})} + \theta\sqrt{(\tau_{m_2})} \tag{14}$$

$$\overline{\sqrt{(\tau_m)}} = \tfrac{1}{2}[\sqrt{(\tau_{m_1})} + \sqrt{(\tau_{m_2})}], \tag{15}$$

in which $\theta = (z/l)$. After raising equation (14) to the second power we find:

$$\tau_m = (1-\theta)^2\tau_{m_1} + 2\theta(1-\theta)\sqrt{(\tau_{m_1}\,.\,\tau_{m_2})} + \theta^2\tau_{m_2}. \tag{16}$$

Further in connection with equation (16) we obtain

$$\bar{\tau}_m = \int_0^1 \tau_m\,.\,d\theta = \tfrac{1}{3}[\tau_{m_1} + 2\sqrt{(\tau_{m_1}\,.\,\tau_{m_2})} + \tau_{m_2} - \sqrt{(\tau_{m_1}\,.\,\tau_{m_2})}]. \tag{17}$$

The sum of the first three terms in the brackets on the right-hand side of equation (17) is according to equation (15) equal to $(2\overline{\sqrt{\tau_m}})^2$. Thus we will have instead of (17):

$$\overline{\sqrt{(\tau_m)}} = \tfrac{1}{2}\sqrt{[3\bar{\tau}_m + \sqrt{(\tau_{m_1}\,.\,\tau_{m_1})}]}. \tag{18}$$

The specific friction force on the wall can be roughly expressed by the equation

$$\tau_m = \frac{\xi_m}{8}(\rho\,.\,W^2)_m = \tau_0\frac{\xi_m\,.\,\rho}{\xi_0\,.\,\rho_m} \tag{19}$$

and after taking into account the latter and formula (a), we obtain

$$\overline{\sqrt{(\tau_m)}} = \frac{1}{2}\sqrt{\left\{3\bar{\tau}_m + \tau_0\sqrt{\left[\left(1+\frac{\rho-\rho_v}{\rho_v}x_1\right)\left(1+\frac{\rho-\rho_v}{\rho_v}x_2\right)\right]}\sqrt{\left(\frac{\xi_{m_1}\,.\,\xi_{m_2}}{\xi_0^2}\right)}\right\}} \tag{20}$$

where τ_0 is as before the specific friction force for a continuous flow of the condensate. Under these conditions the second term on the right-hand side of the equation is considerably smaller than the first. We can therefore neglect in equation (20) the value of $\sqrt{(\xi_{m_1}\,.\,\xi_{m_2}/\xi_0^2)}$ which plays the role of a correction factor and is close to one.

Substituting (20) in (12) after taking into account (13) we find:

$$\frac{\bar{h}\,.\,d}{K} = Nu^* = \frac{1}{2}\frac{Eg\,Cp\,d}{K}\sqrt{\left(\frac{\xi_1\,.\,\rho}{8}\right)} \times\sqrt{\left\{\frac{3}{4}\frac{l}{d}\Delta P_f + \frac{\xi_0\rho W_0^2}{8}\sqrt{\left[\left(1+\frac{\rho-\rho_v}{\rho_v}x_1\right)\times\left(1+\frac{\rho-\rho_v}{\rho_v}x_2\right)\right]}\right\}}. \tag{21}$$

Let us analyse equation (21) making use of the experimental data of the hydraulic resistance of the tubes in the case considered. These data were obtained during the experiments already described on heat transfer using the same experimental tubes. The pressure differential along the whole length of the tube was measured with a differential tubular manometer. The value of ΔP_f which is a part of equation (21) was determined afterwards, having taken into account "acceleration resistance". Further, the values of Nu^* were obtained from equation (21) in which the value of ξ_1 was assumed to be equal to the value ξ_0 in accordance with the theory discussed earlier. The values obtained are shown in Fig. 3, where K^* is determined by equation (9) after substituting in the latter the value of Nu^* instead of $\overline{Nu}$. We can see from Fig. 3 that the values of Nu^* are proportional to the values of $Re^{0\cdot 8}$. In this case the proportionality coefficient in formula of type (10) is equal to 0·029, although according to the experimental formula (2) for one-phase flow, it is equal to 0·021. Therefore their ratio is 1·37. According to equation (21) the value of Nu^* is roughly proportional to $(\Delta P_f)^{0\cdot 5}$. Therefore we may conclude that this ratio depends on the fact that in the case considered hydraulic resistance is about 1·5 times larger than the average value for a homogeneous flow. In connection with this Fig. 4 gives data for the hydraulic resistance obtained from the experiments described. They are given in relative values. One can clearly see that when the Reynolds numbers are small the measured values of hydraulic resistance are close to the

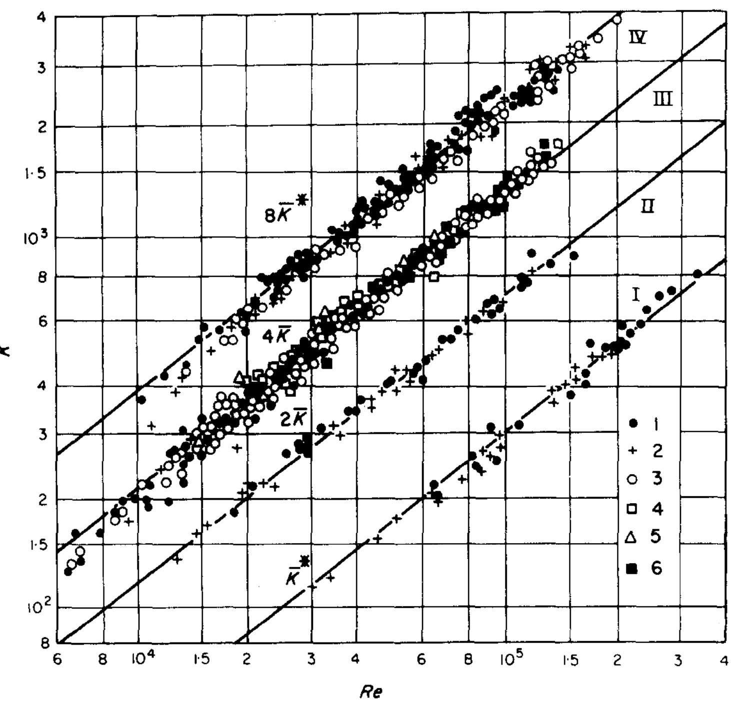

FIG. 3. Relation $K^* = f(Re)$ based on the experimental values of hydraulic resistance for steel pipes 1X18H9T 2500-mm long and the following diameters: 1. $d = 16 \times 1{\cdot}5$ mm; 2. $d = 20 \times 1{\cdot}5$ mm; 3. $13 \times 1{\cdot}5$ mm and for pipes with $d = 16 \times 1{\cdot}5$ mm, $l = 11950$ mm and the pipes with $l = 7650$ mm and $l = 10630$ mm (for the two latter cases $d = 16 \times 1{\cdot}5$ mm).

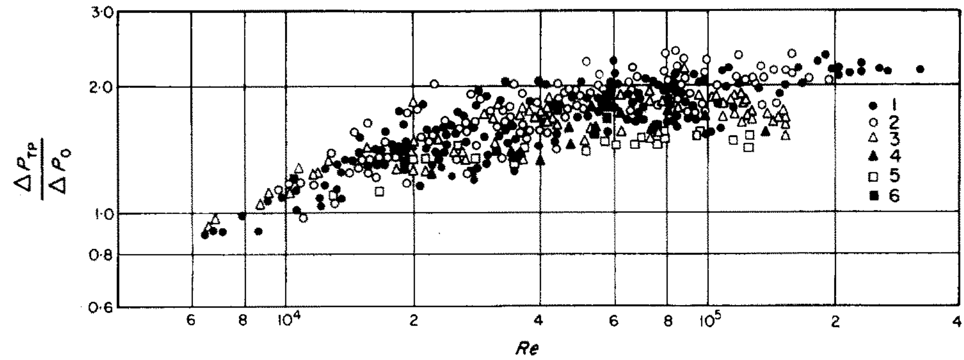

FIG. 4. Hydraulic resistance for steam condensation at 12·2–88 bar pressure in horizontal tubes: 1. $l = 2{\cdot}5$ m, $d = 16 \times 1{\cdot}5$ mm; 2. $l = 2{\cdot}5$ m, $d = 20 \times 1{\cdot}5$ mm; 3. $l = 2{\cdot}5$ m, $d = 13 \times 1{\cdot}5$ mm; 4. $l = 11{\cdot}95$ m, $d = 16 \times 1{\cdot}5$ mm; 5. $l = 10{\cdot}625$ m, $d = 16 \times 1{\cdot}5$ mm; 6. $l = 7{\cdot}650$ m, $d = 16 \times 1{\cdot}5$ mm.

rated resistance of homogeneous flow. But for the majority of the experiments the ratio of these values varies from 1·0 to 2·0.

As stated before, according to direct measurements of heat-transfer intensity the proportionality coefficient in formula (10) is equal to 0·024, i.e. less than 0·029 by 20 per cent. But this difference is comparatively small and one can not draw a conclusion that discrepancy exists in the analogy between hydraulic resistance and heat transfer on the basis of the given experimental data.

Let us dwell now on condensation of steam in the parallel tubes in a bundle of tubes. Heat transfer in steam condensation within a bundle of tubes has certain peculiarities. Heat exchangers of this type are widely employed in industry, especially those with U-shape arrangements of tubes in the bundle. The tubes in the bundles of such heat-exchangers are of different length, the outside tubes being longer than the inside tubes. The maximum difference in the length of the tubes in the U-type heat exchanger reaches the value of $\pi(R_1 - R_2)$ where R_1 and R_2 are respectively the radii of the bend of the inner and outer rows of tubes in the bundle.

This difference in the length of the pipes becomes essential for high-power heat-exchangers. As shown by experimental data this difference in the length of the pipes in the bundle produces a certain variation in the conditions of steam condensation inside them. It results in a certain decrease in heat-transfer intensity in the bundle as a whole compared to heat transfer in a tube of the average length.

With a complete condensation of steam the condensate gathers in the outlet chamber of the heat-exchanger, and there is a certain amount of steam above the condensate. In this case the temperature of steam and condensate corresponds to the saturation conditions at the pressure P_2 in that chamber and which is lower than P_1, the pressure in the inlet chamber. The pressure difference $\Delta P = P_1 - P_2$ is caused by the hydraulic resistance of the heat-exchanger, which in its turn depends on the steam condensation conditions. We can consider as typical the conditions such that, when steam condenses in the tubes of the intermediate length, the condensate leaves the tubes precisely under the above-mentioned temperature of saturation corresponding to the pressure P_2 of steam. Equal amounts of steam condensing along the whole length of the tube go through each of them. Since the difference in pressure in the inlet and outlet chambers is at the same time the difference in pressure in the tubes, the latter has a constant value in all tubes. It is evident therefore that when a heat-exchanger of this type is working a correspondingly smaller amount of steam passes through the longer tubes, and a larger amount of steam through shorter ones, than through the tubes of intermediate length. Since longer tubes have larger heat-transfer surface, the condensate leaving them has a temperature below the saturation point under pressure P_2, i.e. it is overcooled. On the contrary, the shorter tubes lack the necessary surface to be able to condensate the whole flow of steam passing through them. This results in steam leaving these shorter tubes together with condensate. This extra steam condenses in the outlet chamber under the influence of the jets of overcooled condensate pouring from the longer tubes. In this way the overcooling of condensate in longer tubes is compensated by the slipping of steam through the shorter tubes.

The experiments were conducted with two parallel tubes made of stainless steel of 16 × 1·5-mm dia. inclined by 1° to the horizontal in the direction of the flow of steam which was done for the purpose of clearing the tubes from condensate between the tests. The length of one of the tubes was 11950 mm, the length of the other varied and was equal successively to 10625, 7650 and 4945 mm. Saturated steam condensed inside the tubes and the condensate was collected in separate vessels. The level of condensate in each of them was maintained at a precalculated height to prevent steam from escaping outside the vessels. From these vessels condensate was removed and fed into heat-exchangers for cooling, and was afterwards fed into measuring tanks for the determination of its amount by the volumetric method.

The steam spaces of the separating vessels were connected by a short tube and a valve, the diameters of both being 20 mm. Owing to this arrangement the two experimental tubes could work either separately when the valve was closed, or in parallel when the valve was open: in the latter case the steam pressure at the tubes outlets equalized and was practically the same.

The tubes of the separating tee junction at the entrance to the experimental tubes had the same 20-mm i.d. and minimum length of about 300 mm. We can assume on this ground that the pressure of steam at the entrance to the parallel experimental tubes was equal.

The results of these experiments have enabled us to compile charts showing loading characteristics for each experimental tube when operating both separately and in parallel, in the form of formula $\bar{q} = f(\Delta t)$ where $\bar{q}$ is the average specific thermal load on the inner surface of the tube, $\Delta t = t_1 - t_2$ the temperature drop between condensing steam and coolant (water) boiling in the inter-tube space. The above loading characteristics for a tube of 16 × 1·5-mm dia. and 11950-mm long under different pressures of steam condensation are shown in Fig. 5. It can be clearly seen that, other conditions being equal, the average specific thermal load of the heating surface increases as the pressure of condensing steam goes up. Analogous results were obtained on all other tubes operating separately.

With steam condensation inside tubes operating in parallel, as stated before, a re-distribution of specific thermal flows occurs and hence also a re-distribution takes place of heat transfer intensities as compared with their values during separate working of the tubes under the same conditions. This can be clearly seen in Fig. 6 where loading characteristics are plotted for the tubes 11950-mm and 4945-mm long, respectively for their separate and parallel work during the condensation of steam at a pressure of 88 bar. From these diagrams it follows that when the tubes operate in parallel the specific thermal load of the long tube decreases to about one half while the specific thermal load of the short tube, on the contrary, increases, but no more than 1·1–1·2 times. The total amount of heat transmitted from condensing steam to boiling water during parallel operation of the tubes is considerably smaller than when the tubes operate separately. For example, during the condensation of steam at 88 bar in separate tubes 11950-mm and 4945-mm long and with a temperature drop $t = 45$ deg their total thermal load amounted to 220000 W while during

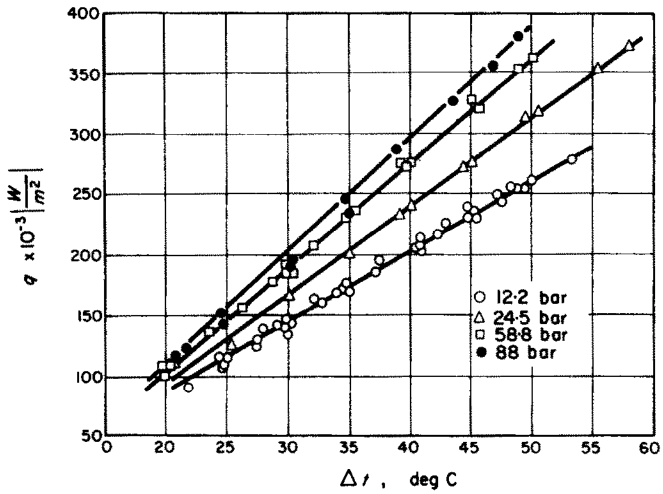

FIG. 5. Loading characteristics of the tube with $d = 16 \times 1{\cdot}5$ mm and $l = 11{\cdot}950$ mm.

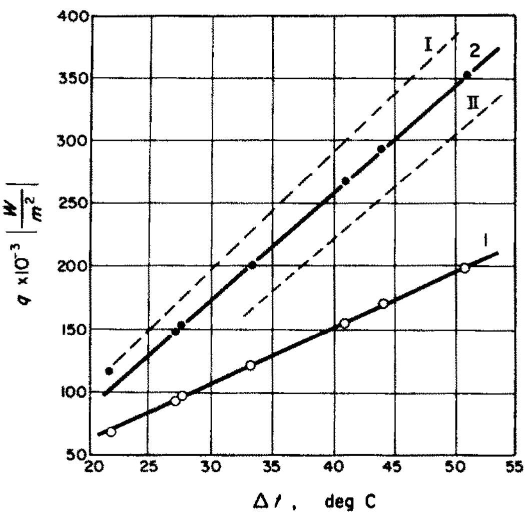

FIG. 6. Loading characteristics for steam condensation under 88 bar pressure of tubes operating in parallel with $d = 16 \times 1{\cdot}5$ mm and $l_1 = 11950$ mm (line 1) and $l = 4945$ mm (line 2). Broken lines I and II correspond to the separate operation of tubes.

parallel work of the tubes their total thermal load under the same conditions amounted to only 142000 W, i.e. 36 per cent less.

If we further cut down the length of the short tube, it will inevitably result in the further decrease of the thermal load of the long tube operating in parallel with the short tube. Obviously the thermal load of the latter reaches its minimum when the length of the short tube comes close to zero. The latter condition was obtained in the experiments by connecting the entrance part of the long experimental tube to its separator via a by-pass tube of 20-mm dia. and 300-mm long, laid outside the evaporators and covered with thermal insulating material. The existence of such a tube practically equalizes the pressure of steam at the entrance and the exit of the experimental tube. Thus in this case the thermal load is essentially determined by the condition of free flow of condensate inside the tube. The results of the experiments are shown in Fig. 7 where one can see the loading characteristics for an experimental tube 11950-mm long and steam condensation at 76 bar for its separate work and when operating in parallel with the above by-pass tube. As one can see from Fig. 7, the existence of the by-pass tube equalizing the pressure of the steam at the entrance results in a sharp decrease in the thermal load of the experimental tube which goes down about three times as compared to that for a separately working tube.

When the tubes operate in parallel and the

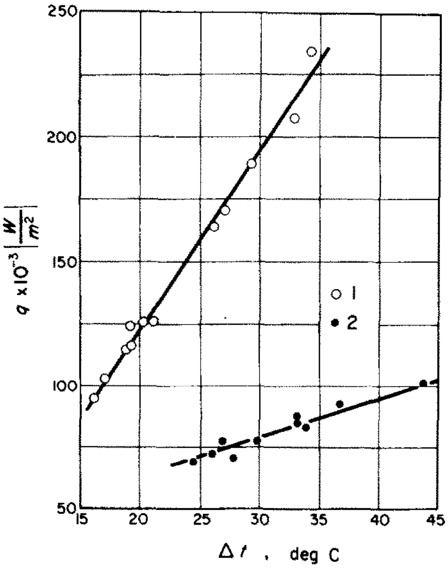

FIG. 7. Loading characteristics of steam condensation with 76 bar pressure in tube with the following parameters: $d = 16 \times 1{\cdot}5$ mm and $l = 11950$ mm.

1. separate operation of tube in total condensation regime;
2. the case when the entrance and exit sections of the tube are connected by a by-pass tube.

valve between the separators of the installation is open, the overcooling of condensate in the longer tube was measured with the help of a chromel–copper thermocouple, which was installed in a sheath so as to meet the flow of condensate just as it left in the tube. Apart from this in the measuring tank the corresponding total amount of condensate was measured. This condensate formed both during steam condensation in the long tube itself and in the condensation of the steam through the shorter tube via the valve, and giving the additional heating to the overcooled condensate bringing it up to the saturation temperature which corresponds to the pressure in the separators. From these measurements were calculated the true amount of condensate actually generated in the long tube, the heat of overcooling of the condensate, and the thermal load of the tube. In the case of the points shown in Fig. 6 the heat of overcooling of condensate amounted to 15 per cent of the total thermal load of the tube. Finally one might say here that the analogous measurements of the temperature of condensate at the exit of the short tube always coincided with saturation temperature.

Again in Fig. 8 are shown data in the form of the dependence of the total thermal load in absolute figures on the pressure of the condensing steam and the relative length of the short tube operating in parallel with the long one, where l_1 and l_2 are the length of the longer and shorter tubes, respectively. The points marked on the chart when $l_1/l_2 = 0$ were obtained from the experiments with the by-pass tube, when the pressure of steam at the entrance and exit of the experimental tube were, to all intents and purposes, equal. This diagram illustrates the essence of the matter and illustrates the fact that the possibilities for steam condensation inside the longer tube which exist in this case in accordance with the conditions of heat transfer, cannot in fact be realized because due to the hydraulic conditions the steam flow through the tube is in this case lower than that required by the conditions of its cooling.

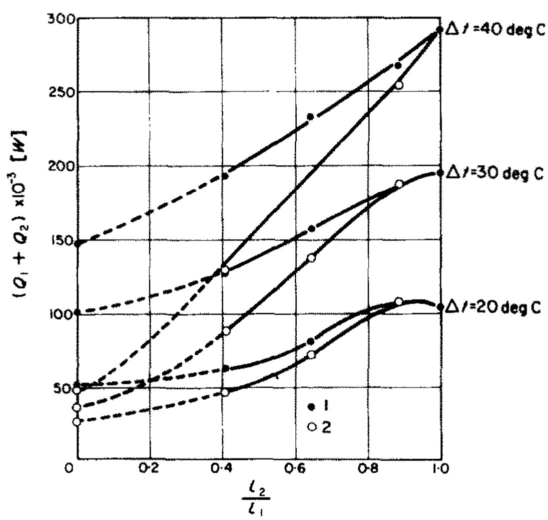

FIG. 8. Dependence of total thermal load on the ratio of tube lengths l_2/l_1 under pressure of condensing steam at 88 bar.

1. separate operation of tubes;
2. parallel operation of tubes.

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Abstract—The paper presents an approximate prediction of heat transfer during steam condensation inside a tube on the basis of the analogy between hydraulic resistance and heat transfer in accordance with Reynolds' theory. It describes experimental results obtained by the authors during the condensation of steam inside tubes with diameter of 18 mm and lengths up to 12 m at pressures up to 90 bar. Theoretical and experimental results agree satisfactorily. It also describes original experimental results on tube hydraulic resistance during condensation of steam therein. Finally, it describes experimental results on the conditions of heat transfer during condensatory flow of steam inside tube bundle.

Résumé—On présente un calcul théorique approché du transport de chaleur pendant la condensation de la vapeur d'eau à l'intérieur d'un tube sur la base de l'analogie entre la résistance hydraulique et le transport de chaleur en accord avec la théorie de Reynolds. On décrit les résultats expérimentaux obtenus par les auteurs pendant la condensation de la vapeur d'eau à l'intérieur de tubes de 18 mm de diamètre et de 12 mm de longueur pour des pressions allant jusqu'à 90 bars. Les résultats théoriques et expérimentaux sont en accord satisfaisant. On décrit également des résultats expérimentaux originaux, sur la résistance hydraulique dans un tube avec de la vapeur d'eau se condensant à l'intérieur. Finalement, on décrit les résultats expérimentaux d'échange de chaleur pendant l'ecoulement de la vapeur au cours de sa condensation un faisceau de tubes.

Zusammenfassung—Die Arbeit bringt eine ungefähre Voraussage über den Wärmeübergang bei der Kondensation von Dampf in einem Rohr. Als Ausgangspunkt dient die Analogie zwischen hydraulischem Widerstand und Wärmetransport in Übereinstimmung mit Reynolds' Theorie. Experimentelle Ergebnisse bei Kondensation von Dampf in Rohren mit Durchmessern von 18 mm und Längen bis 12 m bei Drücken bis 90 bar werden beschrieben. Theorie und Experimente stimmen zufriedenstellend überein. Ausserdem werden experimentelle Ergebnisse von Messungen des hydraulischen Widerstandes während der Kondensation in Rohren wierdergegeben. Schliesslich werden Experimente über den Wärmeübergang bei Kondensation bei der Strömung von Dampf in Rohrbündeln beschrieben.