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Heat transfer in condensation of flowing vapour on a single horizontal cylinder

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Abstract—Experimental data on heat transfer in condensation of moving vapour on a single horizontal cylinder are presented for a wide range of determining parameters. It is shown that the friction on the vapour–film interface, which determines the thickness of the film and, hence, the heat transfer by condensation of moving vapour, depends substantially on the magnitude of the substance cross flow.

1. ANALYTICAL RELATIONS

IN HIS classical work [1] on condensation of pure saturated vapour, Nusselt considered the effect of flowing vapour on a two-dimensional condensate film on the example of a vertical cooled surface presuming the laminar mode of liquid flow, waveless nature of phase interface motion, constancy of friction factor on this interface, and the smallness of interface flow velocity as compared with that of the vapour flow. Under these conditions, the following unambiguous relationship between the similarity numbers holds

$$\frac{\alpha}{\alpha_0} = f\left(\frac{C_f''\rho''W''^2\alpha_0}{g\Delta\rho\lambda}\right). \quad (1)$$

Here, the efficient value of the amount of heat per kilogram of substance is comprised of the latent heat of evaporation and of the condensate supercooling heat

$$r_{ef} = r + \varphi C\Delta T \quad (2)$$

where $\varphi < 1$.

However, in the case of intense condensation, the boundary layer of vapour on the condensate film surface is a layer with a cross flow of substance whose velocity is

$$v'' = \frac{q}{r\rho''}. \quad (3)$$

This fact was first noted by Cherny [2, 3]. The presence of the cross flow of substance considerably changes the behaviour of flow past a body.

It has been known that in the asymptotic domain the tangential stress on the wall with suction is equal to [4]

$$\tau = -\rho''v''W'' \quad (4)$$

and is independent of the flow mode and viscosity. The dimensionless starting length of the laminar boundary layer on the wall with suction is

$$\bar{X} = \left(\frac{v''}{W''}\right)^2 \frac{W''x}{\nu''} = 4. \quad (5)$$

In the available approaches to the problem of heat transfer by condensation of flowing vapour [2, 3, 5–8], in the boundary-layer theory approximations for the laminar flow of both phases one formulates, for each phase, the continuity, motion and energy equations augmented with boundary conditions on the wall and at infinity and also with conjugate boundary conditions on a smooth interface. In a general case the solution of this problem can be done numerically. The methods and results of such solutions can be found elsewhere [9, 10, 11].

When friction on the phase interface is entirely governed by the momentum transferred during the condensation of a quickly flowing vapour (with allowance for the gravity force), the heat transfer law, obtained in refs. [5, 6], is as follows

$$Nu \to \text{const.}\sqrt{Re'}, \quad (6)$$

which can be directly obtained from Nusselt's solution on having substituted into it the value of C_f'' by formula (4).

The gravity force may be neglected on condition that

$$Z = [C_f''W''^2\rho''/2g\delta(\rho'-\rho'')] \gg 1. \quad (7)$$

It is important to observe that heat transfer in this case is independent of the heat flux. A relative change in heat transfer at ΔT = idem depends then on one complex criterion

$$\alpha/\alpha_0 = f(Fr/Pr\,K). \quad (8)$$

At q = idem, a relative change in transfer can be given as

$$\alpha/\alpha_0 = C\,Re^{1/3}\sqrt{Re'}(Ar)^{-1/3} \quad (9)$$

or

$$\alpha/\alpha_0 = C\frac{q^{1/3}W''^{1/2}}{r^{1/3}\rho^{1/3}\nu^{1/6}g^{1/3}L^{1/6}\overline{\Delta\rho}^{1/3}}. \quad (9a)$$

The relative change in heat transfer is seen to be governed by a number of physical parameters of the liquid, vapour velocity and heat flux, and to be weakly dependent on the characteristic linear dimension of a heat transfer surface.

Another limiting situation is represented by the case of friction determination without account for the vapour cross flow effect.

NOMENCLATURE

Ar, Ar_*	Archimedes criterion, $\frac{gL^3}{\nu^2}(1-\rho''/\rho')$, $\left(\frac{\sigma^3}{\nu^4\rho^3 g\Delta\rho}\right)^{1/2}$
C_f''	friction factor on interface
C_p	specific heat of liquid [kJ kg^{-1} °C^{-1}]
C_q	nondimensional cross flow velocity of vapour, $q/r\rho''W''$
D	cylinder diameter [m]
Fr	Froude number, W''^2/gD
K	phase transition criterion, $r/C\Delta T$
m	liquid flow rate [kg s^{-1}]
Nu, Nu^*	Nusselt numbers, $\frac{\alpha D}{\lambda}, \frac{\alpha}{\lambda}\left(\frac{\nu^2}{g\Delta\rho}\right)^{1/3}$
Pr	Prandtl number, ν'/a
q	specific heat flux [W m^{-2}]
R	thermal resistance
Re'	Reynolds number in equation (6), i.e. $\frac{W''D}{\nu''}\cdot\frac{\nu''}{\nu'} = \frac{W''D}{\nu'}$
Re''	vapour flow Reynolds number, $W''D/\nu''$
r	latent heat of evaporation [kJ kg^{-1}]
T'', $\bar{T}_w$	saturation temperature and mean wall temperature [°C]
ΔT	vapour–wall temperature difference, $T''-T_w$ [°C]
W''_∞, W''	free stream vapour flow velocity and vapour velocity in the flow (narrow) area of channel, respectively [m s^{-1}]
x	coordinate [m].

Greek symbols

$\alpha, \alpha_0, \alpha_{0*}$	coefficients of heat transfer in condensation of flowing vapour and of quiescent vapour calculated from Nusselt's formula and that measured in experiments, respectively [W m^{-1} °C^{-1}]
γ''	specific weight of vapour [N m^{-3}]
μ', μ''	dynamic viscosities of liquid and vapour [Pa s]
ν', ν''	kinematic viscosities of liquid and vapour [m^2 s^{-1}]
ρ', ρ''	densities of liquid and vapour [kg m^{-3}]
χ, N, $\overline{\Delta\rho}$	nondimensional complexes, $1+Pr\,K$, $(\mu''\rho''/\mu'\rho')^{1/2}$, $(\rho'\mu'/\rho''\mu'')^{1/2}$, $(1-\rho''/\rho')$.

Then, for $Z \gg 1$, heat transfer is determined from Nusselt's formula

$$Nu \to \text{const.}\left(\frac{PrK}{N}\right)^{1/3}\sqrt{Re'}, \qquad (10)$$

which shows that $\alpha \sim \Delta T^{-1/3}$. It follows from the limiting relations (6) and (10) that the heat transfer rate at a constant vapour velocity changes, depending on the suction intensity (i.e. on the heat flux), not only quantitatively, but also qualitatively. For the calculation of the heat transfer in condensation of flowing vapour on a horizontal single cylinder, the relations have been suggested [6, 8] which approximate the results of the numerical solution, obtained on the assumption of a non-separating flow, which satisfy the limiting relations (6) and (10) and also Nusselt's formula at a zero vapour velocity.

In most of the works on heat transfer in condensation of flowing vapour on a cylinder, the vapour flow is assumed to be intact. The problem is solved for the boundary conditions T_w = const. or q = const. On the other hand, neither in real devices, nor in experiments with condensation on a cylinder can these boundary conditions be strictly fulfilled, since non-uniform distribution of heat flux and temperature on the test section perimeter and along its length is always present. In refs. [12, 13] it was assumed that the heat flux varied stepwise around the cylinder perimeter. The point of vapour boundary-layer separation does not practically depend on the cross flow, and heat transfer in the lower part of the cylinder does not depend on the vapour velocity and is determined from the Nusselt formula for the condensation of quiescent vapour.

The analysis of experimental data also shows that there are modes of flow when the transverse velocity component is of the same order as the vapour free stream velocity, i.e. the boundary-layer approximation turns out to be incorrect.

In the case of the wavy surface of a film, the mechanism of vapour flow interaction with it becomes very complicated. Even the knowledge of the local interface friction is insufficient for the solution of such a problem, since additional information concerning the effect of the condensation process and vapour motion on the critical parameters of film motion is needed. At great flow rates it is also necessary to determine the fraction of the liquid entrained from the condensate liquid film into the vapour flow.

2. EXPERIMENTAL INVESTIGATIONS

A greater amount of research into the vapour velocity effect on condensation heat transfer has been carried out with one working substance, i.e. water vapour or, more precisely, with vapour–air mixtures of various concentrations. The conditions under which the experiments were conducted are summarized in Table 1.

In the authors' opinion, for the analysis of experimental data the following four factors seem to be of basic importance: (a) thorough determination of the amount of noncondensing gases; (b) the technique of wall temperature measurement; (c) the determination of the specific heat flux on the test surface; (d) exact stipulation of the temperature to be used as the determining one for experimental data processing. At present, in some of the works the properties of liquid are taken at the saturation temperature and in others at a mean temperature determined from different relations between the wall and saturation temperatures.

In the majority of studies the experiments were run under vacuum conditions and therefore a rigorous systematic control over air content in vapour needs to be made but this was very often overlooked. The wall temperature was determined: (a) by calculation based on the heat transfer coefficient (condensed vapour—cooling water); (b) by averaging the readings of thermocouples (T); (c) by a wire resistance thermometer (WRT) imbedded in the screw groove of the test section; (d) by using the test section as a resistance thermometer (RT).

It should be noted that the first and the last of these methods of determining the horizontal cylinder wall temperature are the least accurate. The temperature gradient, present in the last case around the perimeter of the test section, does not allow in principle correct measurement of the mean temperature of the section.

The specific heat flux unambiguously specifies the thickness of the film, the Reynolds number and hence the nature of the film flow

$$Re = \frac{\pi D}{2} \bar{q}/\mu r. \tag{11}$$

Generally in experiments, the specific heat flux averaged along the length of the test section is calculated from a change in the enthalpy of the coolant

$$\bar{q} = C_p m \Delta T_1 / F \tag{12}$$

where $\Delta T_1 = T_2 - T_1$ is the difference between the mean temperatures of the coolant at both the exit and entrance of the test section. On the other hand, one may write

$$q = \alpha \Delta T_2, \tag{13}$$

where $\Delta T_2 = T_w - T_C$ is the difference between the wall temperature and the cross-section mean coolant temperature in the test section. In the case of convective heat transfer, the heat transfer coefficient on the side of a coolant at a specified velocity of its motion remains practically constant

$$\alpha = \text{const.}$$

The heat flux is a variable quantity because the vapour–wall temperature difference, ΔT_1, varies. The determination of the conditional heat flux from equation (12) yields a certain mean value, $\bar{q}$, which can differ considerably from the local q and tube-length variable q which is determined from equation (13). The solution of the conjugate problem of condensation heat transfer has shown [33] that the specific heat flux alters along the tube length in an exponential manner, with the exponent depending on the physical properties of liquids and on the ratio between the rates of heat transfer from the side of a condensing vapour and coolant. Whenever a substantial change in the heat flux occurs along the length of the tube, this leads, in turn, to a significant change of the film thickness, hydrodynamics of its flow, and of the transverse vapour velocity component C_q which governs the behaviour of friction at the vapour–film interface. As a rule, the experimental data do not supply information on the values of ΔT_1 and α and hence on the specific heat flux behaviour along the test section length.

It sometimes happens that experiments are carried out at a constant saturation temperature and variable heat flux whose variation is achieved by changing the coolant flow rate over the test section. In almost all cases this means of heat flux variation turns out to be incorrect, since it causes a change in the boundary conditions of an experimental investigation.

Possible inaccurate or incorrect determination of the amount of noncondensing gases in vapour, the mean wall temperature of a test section and of heat flux in experiments, carried out by different authors, leads to the situation that these quantities differ substantially under similar conditions. In the authors' opinion, the reliability and correctness of an experimental study can be indirectly checked by comparing experimentally measured condensation heat transfer coefficients of practically quiescent vapour with those predicted by the Nusselt theory for the same film Reynolds numbers. In the case of laminar film flow ($Re \leqslant 10$), the ratio between experimental and predicted values should be equal to unity. But, as follows from Table 1, in many studies the experimental values of the heat transfer coefficient are smaller than the predicted ones. It can be seen from the table that in a number of publications [15, 16, 21–23, 26–29] the original data are lacking and so analysis is not possible. The results obtained in some of the works are of questionable validity. For example, in ref. [19] the heat transfer coefficient measured for a condensing vapour moving with a velocity of 18 m s^{-1} at $T'' = 20°$C is 10–25% lower than that calculated by the Nusselt theory for quiescent vapour. A similar result is obtained in the experiments of work [12]. At $T'' = 58°$C, vapour–wall temperature difference $\Delta T = 11°$C and vapour velocity of about 10 m s^{-1}, the heat transfer coefficient reaches the value calculated by Nusselt's formula for quiescent vapour. As shown in work [20], for similar conditions the heat transfer intensification is observed at absolute vapour velocities several times below those reported in refs. [12, 19]. The experiments described in refs. [13, 24] were carried out to fit specific applications and therefore involve certain aspects which prevent the results obtained from being

Table 1. Condensation of moving vapour on a horizontal cylinder

Reference	Substance	T (°C)	D (mm)	$q \times 10^{-3}$ (W m−2)	W'' (m s^{-1})	Air concentration (wt.%)	Measurement, T_w	Original data	$\frac{\alpha_{0*}}{\alpha_0}$
1. Ferguson and Oakden [14]	water	48.8–82.2	19	79–315	0.45–33	—		$1/R = f(q; W'')$	0.7–0.9
2. Gudemchuk [15]	—	102–112	9.6	118–857	4.5–26.3	—	—	—	—
3. Kutateladze [16]	—	100–170	10.0	70–130	4–11.5	—	T	—	1.0
4. Berman and Fuks [17]	—	43–93	19	1.7–83	0.6–67.5	0.03	WRT	$\alpha = f(\gamma'' W'')$	0.7–0.8
5. Fuks [18]	—	—	—	—	—	—	—	—	—
6. Rachko [19]	—	29–60	16	33–62	18–108	0.003	T	$\alpha = f(\gamma'' W''^2)$ q = const.	0.7
7. Berman and Tumanov [20]	—	25–80	19	20.6–150	0.62–22.0	0.008–0.017	WRT	$\alpha = f(W'')$	0.75–0.85
8. Buglayev *et al.* [21]	—	102–105	19	120–180	5–17	—	T	—	1.0
9. Shklover and Grigoryev [22]	—	30–75	14–22	20–100	8–70	0.005–0.02	T	—	
10. Fujii *et al.* [23]	—	10–40	14	50	10–40	—	T	—	—
11. Takashi and Soda [24]	—	30–50	19	5.2–13.6	40–60	—	T	table	—
12. Tushakov *et al.* [25]	—	25, 36	25	36–160	3.4–18.6	—	WRT	—	1.0
13. Marushkin *et al.* [13]	—	143–214	16.7	75–415	1.3–19	—	WRT	—	0.9
14. Nicol and Wallace [12]	—	56–65	19.0	60–180	12–110	—		$\alpha/\alpha_0 = f(W'')$ T = const.	—
15. Buyevich [26]	—	36–85	16, 19	50–150	60–215	—	—		—
16. Nobbs and Mayhew [27]	—	99.7	19	$\Delta T = 13.9$–29.4	0.8–10	—	—	—	—
17. Lee and Rose [28]	R113	—	—	—	—	—	—	—	—
18. Honda *et al.* [29]	R113	39–50	8.0, 37.1	$\Delta T = 3$–14	0.1–9.0	—	—	—	—
19. Gogonin *et al.* [30]	R21	40–60	2.5, 16	10.0–200	0–5.0	0.1	T	$\alpha = f(\Delta T)$ W'' = const.	1.0
20. Gogonin *et al.* [31]	R21 R12	40–90	2.5, 16	5–200	0–5.0	0.1–0.02	T	table	1.0
21. Kutateladze *et al.* [32]	R12	40–70	6.0–12	5–75	0–4.0	0.02–0.03	T	table	1.0

used to evaluate the theoretical relations. A drawback common to all of these works is a drastic change of the specific heat flux along the length of an experimental tube due to considerable heating of the cooling water. Thus, according to the estimates based on the recommendations of ref. [33], where a conjugate condensation heat transfer problem was solved, the specific heat flux along the length of the test section [24] could change at least threefold. This, naturally, led to a marked variation along the test section length of the film thickness, hydrodynamics of its flow, of the dimensionless velocity C_q and, consequently, of friction at the vapour–film interface. Moreover, a distinctive attribute of work [13] is a very close location of tubes in a staggered bank when the space between the tubes was much below the size of a breaking off condensate drop. As shown in ref. [34], under such conditions the effect of surface tension forces makes itself felt appreciably, and steady necks of liquid between the tubes of a bundle appear, thereby contributing specific aspects into the heat transfer process.

It may be noted that methodologically a most correct experiment is that carried out by Berman and Tumanov [20]. The experiments were run on a setup with natural circulation of a working fluid, with a steam generator and condenser being contained in the same receptacle. The latter was placed into an evacuated thermally insulated jacket, where pressure was maintained lower than that in the receptacle. The air concentration in the vapour varied within the range 0.008–0.017%. But in Berman's opinion [31], even this concentration might cause reduction in heat transfer by about 10%.

It follows from the above review of the experimental investigations that:

(1) Some of the works cited do not report the original data making their analysis impossible. The majority of works have been done with condensing vapour–gas mixtures of diverse concentrations whose control was not carefully accomplished for the most part. When describing the experimental procedure, the authors generally fail to cite the parameters of the cooling liquid (its velocity and the difference of temperatures at the outlet and inlet of the test section).

(2) The experiments are carried out on cylinders placed in channels of complicated geometries (simulation of flow past a cylinder in a staggered bank of tubes of a condenser), while theoretical solutions consider a plate (or a cylinder) in an infinite unrestricted flow of vapour. This raises a question concerning the choice of the determining velocity for data processing, since the velocity varies considerably in the case of flow past a cylinder placed in a channel bounded by walls.

(3) The available experimental results concerning the effect of vapour velocity on heat transfer differ considerably for similar governing parameters and therefore it is hard to judge, from these experimental data, the validity of the available correlations and the assumptions made.

3. INVESTIGATIONS CARRIED OUT BY THE AUTHORS

The objective of this study was to obtain reliable experimental data on condensation heat transfer of pure saturated vapour in the case of substantial variations in the physical properties of working bodies, hydrodynamic regimes of vapour and liquid flows, geometrical parameters of the heat transfer surface and confining channels. The results of this investigation have been compared with the well-known theoretical and experimental data of other authors.

The investigations were carried out on a forced-circulation rig whose description is given in ref. [31]. The condensing media were the refrigerants R21 ($CHFCl_2$) and R12 (CF_2Cl_2). The chief merit of these agents, for the investigation of condensation heat transfer, lies in the fact that at room temperature and above the liquid is at saturation, with an excess pressure in the rig preventing the access of air from the atmosphere. Systematic chromatographic analyses, carried out after additional purification of fluids, have shown that the content of the main substance in a liquid phase varies within the range 99.95–99.97% by weight and the content of air in vapour on condensation of R21 and R12 amounts to 0.03–0.1 and 0.02–0.03%, respectively. The measurements of the following quantities were taken: the wall temperature of test sections, the pressure in a working volume, the temperature of liquid and vapour, the temperature and the change in the enthalpy of a cooling water in test sections, the cooling agent flow rate, the steam-generator power, the geometrical characteristics of test sections and channels. These gave the heat transfer governing parameters. The wall temperature was determined by averaging the readings of eight or 10 thermocouples imbedded in two cross sections, four or five in each, uniformly around the test section half-perimeter. The heat flux was calculated by equation (12). The temperature of the water, which cooled the test section, varied from 0.5 to 3°C with a corresponding change in the vapour–wall temperature difference from 2 to 30°C. Under these conditions, the specific heat flux changed insignificantly along the length of the test section.

The test sections were cylinders from 2.5 to 17 mm in diameter, installed in rectangular channels. The condensation investigations were run with cocurrent flow past the cylinders. The diagrams of the channels for the test sections are shown in Fig. 1. Figure 2 presents the schematic of a condenser into which the channels with the test sections were installed. The condenser and channels had windows for a high-speed photography. Moving pictures were taken at a speed of 1000 frames s^{-1}. Single cylinders of various diameters (Fig. 1) were placed in channels from 10.5 to 46 mm wide at a distance of 170 mm from the entrance into the channel to their axes. Moreover, the experiments were also conducted with single cylinders, when these constituted one of the tubes of in-line or staggered tube

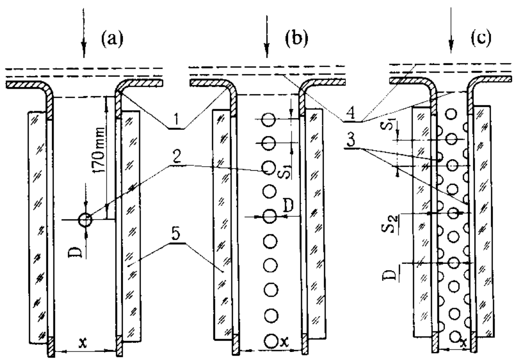

FIG. 1. Schematic diagrams of experimental channels: (1) walls of channel; (2) test sections; (3) halves of tubes; (4) grids; (5) windows.

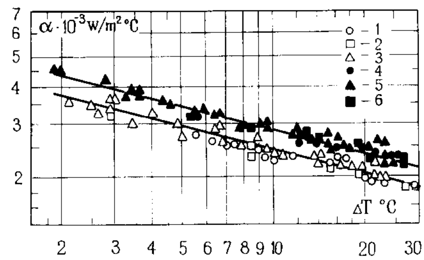

FIG. 3. Dependence of the heat transfer coefficient on vapour–wall temperature difference for the condensation of R21 on a cylinder placed in different channels: $T'' = 60°C$, $D = 16$ mm; $b = 46$ mm, Fig. 1(a), (1) $W'' = 0.61$ m s^{-1}; (6) $W'' = 1.22$ m s^{-1}; $b = 66$ mm, Fig. 1(b), (2) $W'' = 0.57$ m s^{-1}; (4) $W'' = 1.14$ m s^{-1}; $b = 26$ mm, Fig. 1(c), (3) $W'' = 0.57$ m s^{-1}; (5) $W'' = 1.14$ m s^{-1}.

banks, placed in channels from 26 to 66 mm wide. The degree of vapour flow compression, W''/W''_{∞}, in different series of experiments varied from 1.3 to 2.6. The use of two liquids and the conduction of experiments at various temperatures allowed variation of the basic experimental parameters within the following ranges: the vapour density, $12 \leqslant \rho'' \leqslant 160$ kg m^{-3}; the vapour velocity, $0 \leqslant W'' \leqslant 5$ m s^{-1}; the vapour Reynolds number, $2\times10^3 \leqslant Re'' \leqslant 2\times10^5$; the condensate film Reynolds number, $1 \leqslant Re \leqslant 65$; the relative value of the substance cross flow, $4\times10^{-4} \leqslant C_q \leqslant 6\times10^{-2}$.

Figure 3 presents experimental results on condensation of R21 on a single cylinder, with the diameter of 16 mm, situated in a 46-mm wide channel with smooth walls, Fig. 3 (a); on the first, fifth or ninth tube of an in-line tube bank in a 66-mm wide channel, Fig. 3 (b); on the second and subsequent tubes of a staggered tube bank in a 26-mm wide channel, Fig. 3. (c), i.e. with considerably different vapour flow compression factors. Moreover, it might be that the degree of vapour flow turbulence also differed noticeably for vapour flow in the smooth channel or past a cylinder being the second or subsequent tube of the staggered tube bank. As follows from Fig. 3, within the accuracy of the experiments no difference has been found between the heat transfer coefficients for various series of experiments on condensation of vapour moving at the same velocity, based on the flow area of the channel. Similar experiments were carried out with condensation of a moving R12 vapour (Fig. 6). In these experiments, also within the accuracy of observation, heat transfer was the same in channels of various widths when the vapour velocity was based on the narrow area, under otherwise equal conditions. Therefore, in what follows all the calculations of the heat transfer governing parameters, made while processing the present experimental data and those obtained by other authors, were based on vapour velocity in the flow area of the channel. The tables with the measured results and the experimental procedures are given in more detail in refs. [30–32].

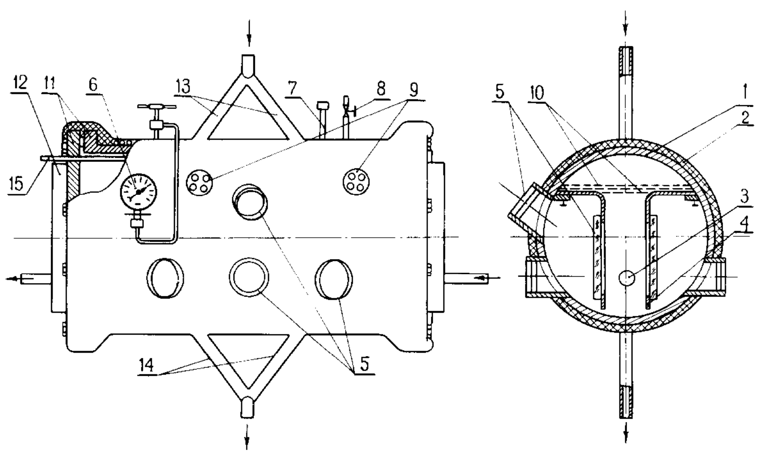

FIG. 2. Schematic diagramof a an experimental condenser: (1) container; (2) insulation; (3) test section; (4) channel; (5) windows; (6) pressure gauge; (7) contact thermometer; (8) blowoff valve; (9) glands for thermocouples; (10) grids; (11) flanges; (12) packing glands of test sections; (13, 14) vapour outlet and inlet; (15) thermocouple sleeve.

The results of measurements at the maximum, for the present experiments, vapour velocities merit particular discussion. Figure 4 contains the experimental data on heat transfer in condensation of R21 vapour on a cylinder with $D = 2.5$ mm under the following conditions: (a) $T'' = 40°C$, $W'' = 5$ m s^{-1}; (b) $T'' = 60°C$, $W'' = 3.8$ m s^{-1}.

The curves numbered 4 are the curves which average the experimental results under the above-mentioned conditions and those numbered 3 show the calculation from the Nusselt theory for quiescent vapour. A characteristic feature of these experiments is a pronounced discontinuity of the curve $\alpha = f(\Delta T)$. The conditions of the experiments make it possible to ignore the effect of the gravity force and to compare the experimental data with the results obtained from the limiting relations (6) and (10). In Fig. 4, the line labelled 1 shows calculation by relation (4) for $C = 0.9$. Line 2 presents calculation by relation of type (10) [8]. As is seen from the experiments, at a small heat flux the experimentally obtained curve has the form

$$\alpha \sim \Delta T^{-1/3},$$

which is of the same nature as predicted by relation (10). At a high heat flux the heat transfer is almost independent of it, as follows from relation (6). In the experiments presented in Fig. 4, the relative magnitude of the

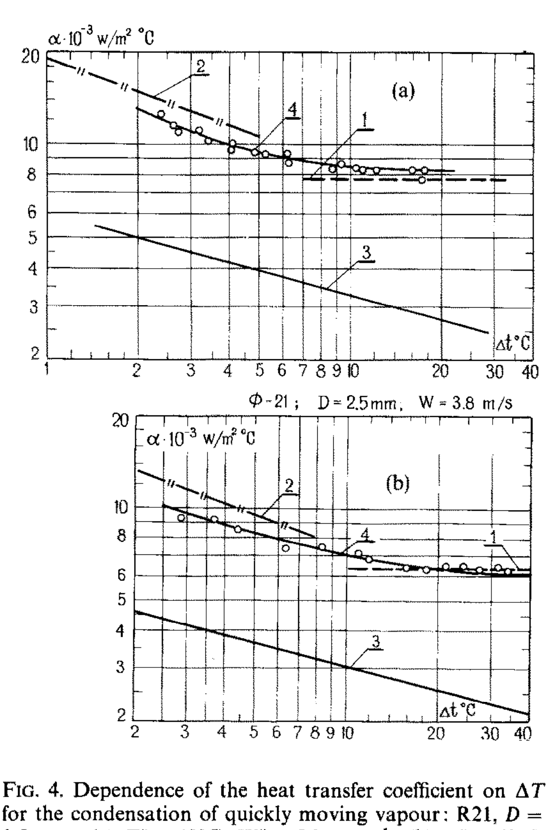

FIG. 4. Dependence of the heat transfer coefficient on ΔT for the condensation of quickly moving vapour: R21, $D =$ 2.5 mm: (a) $T'' = 40°C$, $W'' = 5.0$ m s^{-1}; (b) $T'' = 60°C$, $W'' = 3.81$ m s^{-1}; (1) calculation by equation (6); (2) by equation (10); (3) by the Nusselt theory, $W'' = 0$, (4) line averaging the experiment.

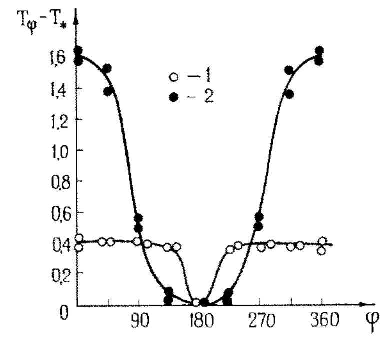

FIG. 5. Measurement of the wall temperature around the perimeter of a 12-mm diameter cylinder: R12, $T'' = 83°C$, $q = 7 \times 10^4$ W m^{-2}; (1) $W'' = 0$; (2) $W'' = 1.2$ m s^{-1}; T_* for $\beta = 180°$.

substance cross flow at the phase interface changed by about an order, $0.15 \times 10^{-2} \leqslant C_q \leqslant 1.25 \times 10^{-2}$, while dimensionless starting length, calculated from equation (5), varied within the range $0.06 \leqslant \bar{X} \leqslant 4.1$. Of course, the parameter $\bar{X}$, calculated by equation (5), can only qualitatively (by the order of magnitude) characterize the same parameter for condensation on a cylinder, since relation (5) implies a separation-free flow past a cylinder.

A pronounced change in the nature of the function $\alpha = f(\Delta T)$ seems to be associated with a substantial change in the behaviour of friction on the phase interface and is due to a change in the transverse velocity of condensing vapour. A satisfactory fit of the prediction to the data confirms this suggestion.

The effect of vapour on a condensate film results not only in heat transfer intensification, but also in a considerable redistribution of the film around the perimeter of the cylinder, as evidenced by the cylinder-wall temperatures measured at various points of the perimeter. The angle is read off from the frontal point. Figure 5 shows the manner in which the wall temperature of a 12-mm diameter copper cylinder (with 1.5-mm thick wall) varies under the influence of vapour.

Figures 6 and 7 present the experimental data on condensation of R12 vapour on a single 6-mm diameter cylinder installed in 12- and 21-mm wide channels and on a 12-mm diameter cylinder placed in a 21-mm wide channel. The experiments were carried out at three saturation temperatures and are presented in the coordinates

$$Nu^* = f(Re). \tag{14}$$

Here

$$Nu^* = (\alpha/\lambda)[\nu^2/g(1-\rho''/\rho')]^{1/3}$$

and

$$Re = \frac{\pi D}{2}\bar{q}/\mu r(1+\varphi/K)$$

where $\varphi = 3/8$ and $K = r/C\Delta T$ are the parameters allowing for the condensate supercooling [10]. The physical properties of liquid were taken at the

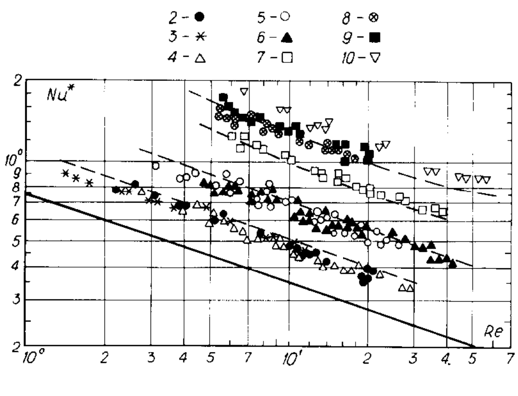

FIG. 6. Heat transfer in condensation of R12 on a single horizontal cylinder: (1) calculation according to ref. [1] for $W'' = 0$, $M = 0.61 \pm 0.05$, $D = 6$ mm; (2) $T'' = 70°C$; $b = 21$ mm; (3) $T'' = 40°C$; $b = 12$ mm, $D = 12$ mm; (4) $T'' = 40°C$, $M = 2.8 \pm 0.3$, $D = 6$ mm; (5) $T'' = 70°C$; $b = 21$ mm and $b = 12$ mm; $D = 12$ mm; (6) $T'' = 40°C$, $T'' = 55°C$, $T'' = 70°C$, $M = 8.1 \pm 0.8$, $D = 6.0$ mm; (7) $T'' = 70°C$, $b = 12$ mm, $M = 17.5 \pm 1.0$, $D = 6.0$ mm; (8) $T'' = 40°C$, $b = 12$ mm; (9) $T = 40°C$, $b = 21$ mm; (10) $T'' = 70°C$, $b = 12$ mm. Dashed line, calculation according to ref. [6].

saturation temperature [36], their variations across the film thickness were incorporated into the relation which approximates the numerical solution given in ref. [37] when calculating the criteria of relation (14). The corrections for the supercooling of liquid and change in the film physical properties did not exceed 15 and 5%, respectively. As follows from the approximating relations suggested in refs. [6, 8], a relative change in heat transfer is a function of one complex parameter $\chi^{4/3}Fr/Pr\ K = M$. For this reason, the vapour velocities in different series of experiments given in Fig. 6 (different properties of liquids, diameters of cylinders, widths of channels) were taken at certain specified values of the determining complex $\chi^{4/3}Fr/Pr\ K$.

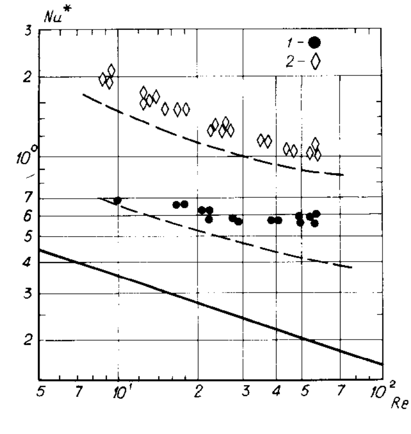

FIG. 7. Heat transfer in condensation of R12 on a single horizontal cylinder: $M = 2.70 \pm 0.3$, $D = 12$ mm; (1) $T'' = 83°C$, $W'' = 1.18$ m s^{-1}, $M = 27 \pm 3$, $D = 6.0$ mm; (2) $T'' = 70°C$, $b = 12$ mm, $W'' = 2.9$ m s^{-1}. Solid line, calculation according to ref. [1] for $W'' = 0$. Dashed line, calculation according to ref. [6].

It is evident that at the maximum vapour velocity, the heat transfer increases by more than a factor of four the value for a quiescent vapour. The experimentally obtained relationships for condensation of a moving vapour have the form

$$Nu^* \sim Re^{-1/3},$$

i.e. the same as for a laminar flow of a film. The dash-dotted line in this figure presents the calculation for some vapour velocities by the relation suggested in ref. [16]. The agreement between the predicted results and experimental data is satisfactory. It has been found in the experiments that at a certain critical vapour velocity there occurs the break-up of condensate droplets by a vapour flow. The value of the critical velocity in the present experiments was picked up on the movie film of the condensate process. The motion pictures were taken at constant saturation temperature and heat flux, while the vapour velocity changed in a stepwise manner increasing with each shooting. Figure 8 shows the characteristic photos of the flow of liquid from a horizontal cylinder at a constant heat flux and different velocities of vapour. The movie film revealed that before a liquid droplet broke off it moved time and again along the cylinder generatrix under the action of vapour and even vibrated. As the vapour velocity increased, the amplitude and frequency of the vibrations of the droplet grew and its shape became less distinct (Fig. 8). The movie film makes it possible to find out that at $T'' = 40°C$ the R12 vapour velocity, at which the breakup of condensate droplets started, was $W''_{\infty} = 0.7$–0.75 m s^{-1} and at $T'' = 70°C$ it was about 0.6 m s^{-1}. The critical velocity is given here per entire cross-section of the channel. In the literature dealing with the condensation heat transfer, the influence of the breakup of condensate droplets by vapour flow has not been virtually touched upon. The vapour velocities at which the breakup of droplets begins can be estimated only by the relations describing the fragmentation of liquid droplets in a noncondensing gas flow. Calculations made by relations suggested in refs. [38, 39] show that the critical vapour velocities W''_{∞}, obtained in the present experiments, approximately correspond to those predicted. When vapour condenses at a rate, exceeding the critical one, the condensate on the lower generatrix of the cylinder wets the glass walls of the channel. At vapour velocities within the range $1.0 \leqslant W'' \leqslant 2.0$ (Fig. 8), the liquid is sprayed from separate points as if from nearly equally spaced nozzles. At $W'' > 2.0$ m s^{-1} these centres disappear and the breakup occurs uniformly from the entire lower generatrix of the cylinder. At low heat fluxes the condensate film is not seen and the tube seems to be dry. Figure 6 presents the data measured at sub- and supercritical vapour velocities. These have been compared with correlations developed on the assumption of laminar film flow and laminar separation-free vapour flow past a cylinder [6]. Rough agreement between the measured data and the predicted results shows that the breakup of the

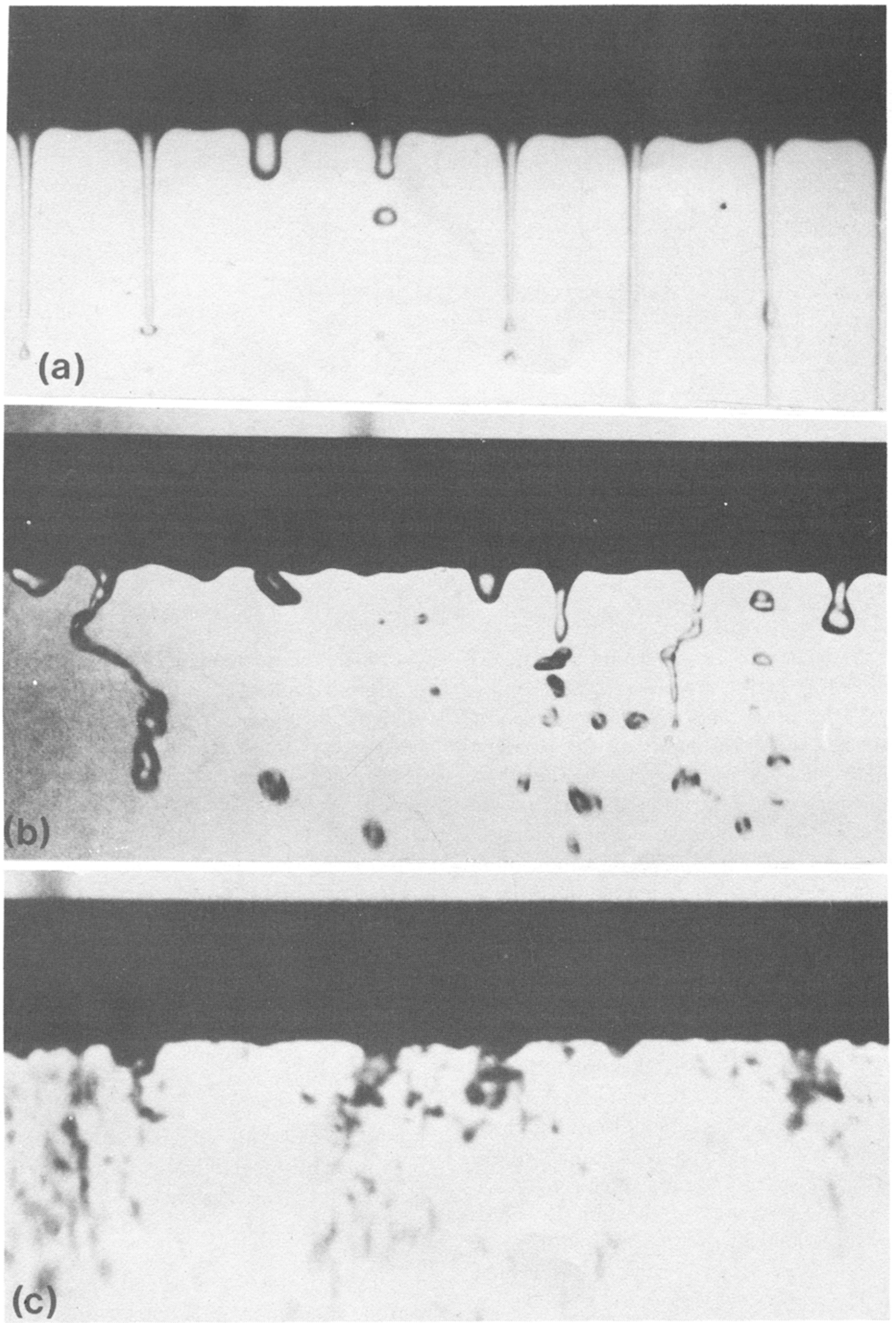

FIG. 8. Moving pictures of R12 vapour condensation: $T'' = 40°C$, $D = 6.0$ mm, $Re = 15$, $b = 12$ mm. (a) $W'' = 0$; (b) $W''_{\infty} = 0.62$ m s^{-1}; (c) $W''_{\infty} = 1.1$ m s^{-1}. Speed of shooting, 1000 frames s^{-1}.

condensate droplets does not cause a drastic change in heat transfer and exerts a relatively weak effect on condensate heat transfer for a single cylinder. Exceptions to this are the experiments carried out at $T'' = 83$ and 70°C and shown in Fig. 7. These differ noticeably from the experiments conducted at the same determining parameter, but at a much lower saturation temperature. As the temperature increases, the liquid surface tension decreases substantially and the vapour density markedly increases. It can be assumed that under these conditions the liquid is entrained from the film surface into the vapour flow thus causing an increase in the heat transfer rate and a change in the character of the dependence $Nu^* = f(Re)$, revealed in the experiments which are presented in Fig. 7.

The data, obtained in this study for the condensation of cooling agents on cylinders of different diameters at various vapour densities and velocities, allowed a wide

Table 2. Range of the limiting variations of basic experimental parameters

$q \times 10^{-3}$ (W m^{-2})	ΔT (°C)	W'' (m s^{-1})	Re	$Re'' \times 10^{-3}$	$C_q \times 10^3$	α/α_0	$\chi^{4/3} Fr/Pr\, K$	$\bar{X}$
2	3	4	5	6	7	8	9	10
R21, $T'' = 60$°C, $D = 2.5$ mm, $b = 10.5$ mm [Fig. 1 (a)], $W''/W''_\infty = 1.31$, ref. [30]								
19.5	3.1	0.48	1.35	2.14	9.1	1.77	0.61	0.28
132	38	0.48	9.0	2.14	61.5	1.87	0.92	12.85
25.4	2.8	3.81	1.75	17.0	1.49	3.42	36.0	0.06
212	24.3	3.81	14.5	17.0	12.4	3.87	55.0	4.1
R12, $T'' = 40$°C, $D = 6.0$ mm, $b = 12$ mm [Fig. 1 (a)], $W''/W''_\infty = 2.0$, ref. [32]								
8.3	2.59	0.55	2.6	12.5	2.1	1.4	0.61	0.086
44.5	21.5	0.55	14.0	12.5	11.0	1.57	0.52	2.35
14.4	2.05	3.0	4.53	68	0.66	3.8	20	0.047
59.25	12.26	3.0	18.6	68	2.7	3.6	16.5	0.78
water vapour, $T'' = 80$°C, $D = 19$ mm, $b = 28$ mm [Fig. 1 (c)], $W''/W''_\infty = 3.1$, ref. [20]								
49.5	2.5	0.78	1.8	0.38	95	1.13	0.023	4.3
62.8	2.5	3.4	2.3	2.17	21.2	1.55	0.73	1.55
122	7.4	1.33	4.5	0.65	137	1.27	0.010	19.8
148	7.4	4.15	5.5	2.05	53.5	1.65	0.98	9.4

variation of the heat transfer governing parameters. Table 2 lists the range of the parameters used in some of the present experiments and also shows a conspicuous difference between the experiments on the condensation of cooling agents and water vapour. In Table 2 are set out the substance, the saturation temperature, the cylinder diameter, the width of the channel where the test section is situated, the degree of vapour flow restriction, the minimum and maximum vapour velocities in each series of experiments, the minimum and maximum values of the specific heat fluxes and respective vapour–wall temperature differences, the film and vapour Reynolds numbers, the variation of the determining complex $\chi^{4/3}Fr/PrK$, the relative value of the cross flow of substance C_q, the dimensionless starting length $\bar{X}$ calculated from equation (5).

The comparison of the experimental results on the condensation of cooling agents and water vapour shows that the use of the cooling agents makes it possible to extend the range of the film Reynolds numbers by more than an order of magnitude, the vapour flow Re'' number by two orders and to change the dimensionless cross flow vapour velocity also by two orders of magnitude.

For the first time in the experiments on condensation this allowed the authors to approximate that limiting situation when friction of flow past a 'dry' cylinder contributes appreciably to the total friction. As is seen from Table 2, the cross flow of the substance has the minimum value when heavy $R12$ vapour condenses and it varies within the range 10^{-4}–10^{-3}, and has the maximum value when water vapour condenses and it varies within the range 10^{-2}–10^{-1}. The calculation of the dimensionless starting length shows that in the case of R12 condensation, the vapour flow past a cylinder occurs entirely in the starting length region; in the case of water vapour condensation, it occurs in the asymptotic region when $\bar{X} > 4.0$. The parameters of the experiments with the condensation of R21 occupy a transitional region.

This requires the friction on the phase interface to be nearly always considered as the total 'dry' friction plus the friction of flow past a surface with boundary layer suction, while at $\bar{X} > 4.0$ it can be incorporated by relation (4).

In Fig. 9, the experimental data on condensation of moving vapour on single tubes are presented in the coordinates

$$\alpha/\alpha_{0*} = f(\chi^{4/3}Fr/Pr\,K), \tag{15}$$

provided Re = idem. In the authors' opinion, the most correct is the condition of equality of the film Reynolds numbers in calculations of the relative change in transfer by condensation of moving vapour as compared with quiescent vapour. Really, in the case of laminar flow of a film, its thickness is unambiguously determined by Re; in the case of laminar wavy flow, it as yet does not strongly depend on Ar^* and, in a first approximation, can also be determined only by Re.

While processing the experimental results on the condensation of R12 and R21, the value of α_{0*} is taken from the experimental data, because at the maximum Reynolds numbers, for the condensation of quiescent vapour, the heat transfer intensification amounted approximately to 30% as compared with the Nusselt prediction. In the experiments on water vapour condensation [20, 25], the film Reynolds number did not exceed 6 and the value of α_0 was calculated by Nusselt's formula. The majority of points with the spread of $\pm 25\%$ are correlated in the coordinates of equation (15) and agree satisfactorily with the approximating relations for numerical solutions [6, 8].

4. CONCLUSIONS

A good agreement between the experimental data obtained on the condensation of different substances in

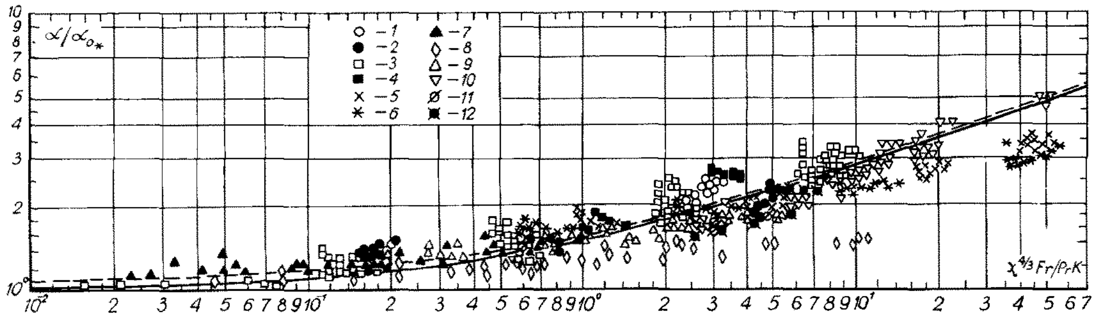

FIG. 9. Relative change in heat transfer by condensation of moving vapour as a function of the parameter $\chi^{4/3} Fr/Pr\ K$. Re = idem. Dash-dotted line, calculation according to ref. [6], solid line, according to ref. [8]: R12, D = 16 mm; (1) T'' = 40°C; (2) T'' = 60°C; R21, D = 16 mm, (3) T'' = 60°C; (4) 90°C, D = 2.5 mm; (5) T'' = 40°C; (6) 60°C, ref. [31]. Water vapour, D = 19 mm; (7) T'' = 80°C; (8) T'' = 25°C, ref. [20]; (9) D = 25 mm, T'' = 36°C, ref. [25]; R12, D = 6.0 mm; (10) T'' = 40°C; (11) T'' = 70°C; (12) D = 12 mm, T'' = 70°C, ref. [32].

test sections of different diameters placed into channels of diverse geometries for different vapour and film Reynolds numbers and also their approximate agreement with the predicted relations confirm the validity of the basic assumptions made for the solution of condensation heat transfer problems in the works of Cherny, Cess and other authors. It is apparent that the friction on the vapour–film interface, which determines the thickness of the film and thereby the condensation heat transfer of flowing vapour depends appreciably on the magnitude of the cross flow of substance.

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TRANSFERT THERMIQUE EN CONDENSATION D'UN ECOULEMENT DE VAPEUR SUR UN UNIQUE CYLINDRE HORIZONTAL

Résumé—On présente des résultats expérimentaux sur le transfert thermique en condensation d'une vapeur mouvante sur un unique cylindre horizontal, pour un large domaine de paramètres. On montre que le frottement sur l'interface vapeur–film qui détermine l'épaisseur du film, et par suite le transfert thermique par condensation, dépend fortement de l'importance du débit transversal de matière.

WÄRMEÜBERGANG BEI DER KONDENSATION VON STRÖMENDEM DAMPF AN EINEM EINZELNEN WAAGERECHTEN ZYLINDER

Zusammenfassung—Es werden Versuchsdaten für den Wärmeübergang bei der Kondensation von strömendem Dampf in einem weiten Parameterbereich angegeben. Es zeigt sich, daß die Reibung an der Grenzfläche zwischen Dampf und Film, welche die Filmdicke und damit auch den Wärmeübergang bei der Kondensation bestimmt, im wesentlichen von der Größe des Querstroms abhängt.

ТЕПЛООТДАЧА ПРИ КОНДЕНСАЦИИ ДВИЖУЩЕГОСЯ ПАРА НА ОДИНОЧНОМ ГОРИЗОНТАЛЬНОМ ЦИЛИНДРЕ

Аннотация Приведены экспериментальные данные по теплообмену при конденсации движущегося пара на одиночном горизонтальном цилиндре в широком диапазоне изменения определяющих параметров. Показано, что трение на границе раздела пар-пленка, определяющее толщину пленки, а, следовательно, теплообмен при конденсации движущегося пара, существенным образом зависит от величины поперечного потока вещества.