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Investigation of heat transfer in film condensation of flowing vapour on horizontal tube banks

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Abstract—This paper reports experimental results on heat transfer in film condensation of flowing vapour on horizontal tube banks. The ranges of dimensionless parameters have been determined within which heat transfer is virtually governed by either the condensate flow rate alone, or vapour velocity alone, or the joint action of these factors.

INTRODUCTION

THE INTENSITY of film condensation heat transfer in a heat-exchanging apparatus is determined by three interrelated factors, i.e.

—condensate flow rate varying over the tube bank height;
—variable velocity of a vapour flow interacting with the condensate film;
—diffusion processes in the presence of non-condensable admixtures in the vapour.

The existence of vapour–liquid interfaces, wave processes and critical phenomena in film flows, the possible entrainment of a liquid phase by a vapour flow as a result of its dispersion, film fragmentation by the vapour flow and return of drops into the film, the effect of substance cross flows on the film hydrodynamics are all important and complicated factors which govern the intensity of heat exchange in the apparatus and which have still not been completely investigated.

In the present state of development, the strict mathematical description of heat and mass transfer processes in condensation is still far from being complete. Apart from pure mathematical difficulties, owing to which these complex problems can be posed and solved only under certain assumptions, one of the main reasons for this is a limited number or complete absence of specially arranged and carefully conducted fundamental experimental investigations of various condensation heat transfer problems.

In a number of works [1–3] the present authors reported experimental results, obtained in a wide range of determining parameters, and important laws derived on their basis.

In condensation of practically quiescent vapour on single horizontal cylinders of different diameters and the bundles of these cylinders, it has been found out that:

(a) The effect of the condensate flow rate is uniquely determined by the film Reynolds number, at least, for a bank of tubes of the same geometry. It is clearly shown that the film Reynolds number varies in a wide range within which the heat transfer remains practically unchanged.

(b) When $Re > 50$, a substantial contribution to the heat transfer comes from the heat exchange in the intertube space in vapour condensation on supercooled liquid droplets and streamlets. The liquid arrives at the lower tube of the bank at a temperature close to the saturation temperature thus leading to the formation of the thermal boundary-layer starting length. Here convective heat transfer takes place.

(c) The relative starting length of the thermal boundary layer depends on both the film Reynolds number and cylinder diameter. Experiments have clearly shown that the convective heat transfer over the starting length and the influence of surface-tension forces lead to an appreciable intensification of condensation heat transfer on a bank of small diameter tubes.

In condensation of flowing vapour on a single horizontal cylinder, it has been found out that:

(a) The friction on the vapour–film interface, that governs the film thickness, and consequently the heat transfer, depend mainly on the substance cross flow. It is in good agreement with the present authors' experimental results and those by other workers [3–6]. The experimental results can be roughly described by approximate expressions of the numerical solutions of the problem with conjugate boundary conditions where the friction is determined with allowance for the substance cross flow [7, 8]:

$$Nu/\sqrt{Re^*} = 0.64\kappa^{1/3}\left[1+\left(1+\frac{1.69\,Pr\,K}{\kappa^{4/3}\,Fr}\right)^{1/2}\right]^{1/2} \tag{1}$$

and

$$Nu/\sqrt{Re^*} = c\kappa^{1/3}\left(1+\frac{0.276\,Pr\,K}{\kappa^{4/3}\,Fr}\right)^{1/4}. \tag{2}$$

At the zero vapour velocity, relations (1) and (2) go over into the Nusselt formula [9].

(b) The values of the critical vapour velocities, which

NOMENCLATURE

c	heat capacity [$\mathrm{J\,kg^{-1}\,°C^{-1}}$]
D	outer diameter of a tube [m]
D_0	diameter of a breaking-off droplet [m]
Fr	Froude number, w''^2/gD
G	condensate flow rate, i.e. amount of condensate flowing through a given cross-section over a 1-m-wide band [$\mathrm{kg\,m^{-1}\,s^{-1}}$]
g	free fall acceleration [$\mathrm{m\,s^{-2}}$]
K	criterion of phase transformation, $r/c\Delta T$
Nu, Nu^*	Nusselt numbers, $\alpha D/\lambda$; $-(\alpha/\lambda)(\nu^2/g\overline{\Delta\rho})^{1/3}$
Pr	Prandtl number, $c\mu/\lambda$
q	specific heat flux [$\mathrm{W\,m^{-2}}$]
Re	film Reynolds number $= \dfrac{G}{2\mu'} \equiv \dfrac{\pi D}{2}\sum_1^n g_i/\mu' r$, where $j = 1, 2, \ldots, n$ is the tube number
Re^*	modified Reynolds number for vapour, $\dfrac{w''D}{\nu''}\dfrac{\nu''}{\nu'} = \dfrac{w''D}{\nu'}$
r	latent heat of evaporation [$\mathrm{J\,kg^{-1}}$]
$T'', \bar{T}_w$	saturation temperature and mean wall temperature [°C]
ΔT	vapour–wall temperature difference, $T'' - \bar{T}_w$ [°C]
w''	vapour velocity calculated in the flow (narrow) area of channel [$\mathrm{m\,s^{-1}}$]
w''_{cr}	critical vapour velocity at which fragmentation of liquid droplets and streamlets by vapour flow begins [$\mathrm{m\,s^{-1}}$]
x	channel width [m].

Greek symbols

α_0, α	coefficients of heat transfer in condensation of stationary and flowing vapour, respectively [$\mathrm{W\,m^{-2}\,°C^{-1}}$]
λ	thermal conductivity of liquid [$\mathrm{W\,m^{-1}\,°C^{-1}}$]
μ', μ''	dynamic viscosity of liquid and vapour, respectively [Pa s]
$\kappa = 1 + Pr\,K/N$, $N = (\rho'\mu'/\rho''\mu'')^{1/2}$, $\overline{\Delta\rho} = (1 - \rho''/\rho')$	dimensionless complexes
ν', ν''	kinematic viscosity of liquid and vapour, respectively [$\mathrm{m^2\,s^{-1}}$]
ρ', ρ''	density of liquid and vapour, respectively [$\mathrm{kg\,m^{-3}}$]
σ	surface tension [$\mathrm{N\,m^{-1}}$].

lead to the fragmentation of condensate droplets and streamlets prior to their breakaway from the cylinder surface, can be determined from the relation [10]:

$$\rho'' w''^2_{cr} D_0/\sigma > 10. \qquad (3)$$

As will be shown later, it is extremely important to know the value of the critical velocity, since beyond it there is uncertainty in the film Reynolds number prediction for tube banks.

The present paper, which is a continuation of earlier reported investigations [1–3], describes *experimental results on condensation of a flowing vapour on horizontal tube banks* for the case of the joint influence of vapour velocity and condensate flow rate on heat transfer intensity in condensation of a practically pure vapour (i.e. without noncondensable admixtures). The experiments were run on 10-row tube banks in a cross flow of a coolant (R21 and R12) vapour which was supplied from above. An additional condenser with an extensive heat transfer surface, installed downstream of the test condenser, allowed the generation of a directed vapour flow. The experiments were carried out in a forced-circulation rig with the steam-generating capacity of up to 1 $\mathrm{kg\,s^{-1}}$ [11].

The test sections, which were 16-mm-diameter and 585-mm-long tubes, were differently arranged in rectangular cross-section channels (Fig. 1). The flat walls of the channel were fitted with optical windows mounted flush with the inner surface of the channel. The channel windows were mated with the windows in the

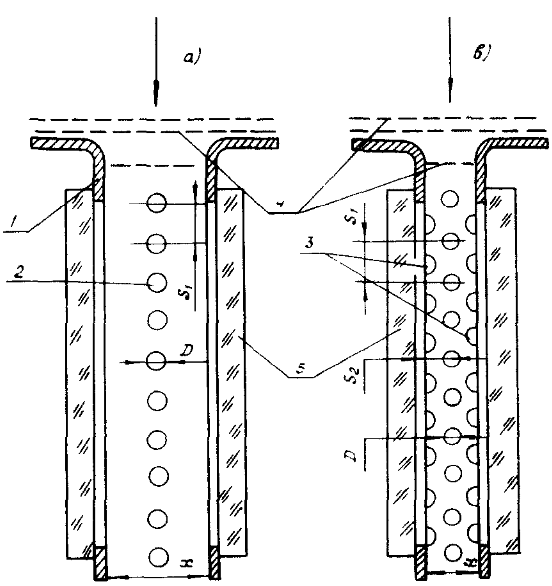

FIG. 1. Schematic diagrams of experimental channels: 1, channel walls; 2, experimental tubes; 3, tube halves; 4, grids; 5, windows.

Table 1. Experimental conditions

Substance	T'' (°C)	q (kW m^{-2})	ΔT (°C)	w'' (m s^{-1})	Re	$\kappa^{4/3}$ Fr/Pr K	Channel width x (mm)	Schematic diagram	Refs.
		Range of variation							
R21	60	7.4–140	1.7–31.4	0.57, 1.14, 2.3, 4.3	3.0–300	0.15–7.7	26	b	[12]
R21	60	7.4–130	1.7–30	0.57	3.0–200	0.15	66	a	[12]
R12	40	5.0–28	2.6–21	0.49	3–200	0.15	46	a	[13]
R12	60	4.0–35	2.1–28	0.4	4–370	0.15	46	a	[13]
R12	40	6.7–27.6	2.3–20	0.49, 0.88, 1.9	3–240	0.15–2.8	26	b	[13]
R12	40	4.7–26	2.2–22	0.49, 0.88, 1.9	3–240	0.15–2.8	26	a	[13]
R12	60	4.1–35	2.1–27	0.79, 2.0	4–250	0.60–4.7	26	a	[13]

housing of the experimental condenser which allowed visual observations by stroboscopic illumination and high-speed photography. Three grids with different sized meshes were installed at the channel inlet providing, together with a smooth entrance, a uniform distribution of vapour over the channel cross-section.

In-line tube banks [Fig. 1(a)] were arranged in one vertical row with the relative spacing $S_1/D = 1.87$ in channels 26, 46 and 66 mm wide. Staggered banks [Fig. 1(b)] were placed in a 26-mm-wide channel, to whose walls uncooled inserts of the halves of 16-mm-diameter tubes were fixed. The geometric characteristics of the staggered bank are: $S_1/D = 1.87$ and $S_2/D = 0.81$.

The following parameters were experimentally measured: the pressure and temperature of the vapour in the condenser and steam-generator, the flow rate, temperature and heating of cooling water in each tube of the bank, the temperature of the tube wall, and the coolant flow rate. These measurements allowed the prediction of the specific heat flux and vapour–wall temperature difference in each region, the vapour velocity in the channel and the saturation temperature. Below, the vapour velocity is given, calculated in the flow (narrow) area of the channel.

The measurements were made in the steady-state operating conditions of the set-up, with good thermostatic protection of its basic units. The vapour velocity at the channel inlet was constant, while the specific heat flux on the tubes of the bank was varied by changing the temperature of the cooling water at the tube inlets. Different condensate flow rates were obtained by both changing the vapour–wall temperature difference and varying the quantity of cooled tubes in the bank.

The use of different cooling agents and carrying out the experiments at different saturation temperatures allowed the vapour density to be varied by almost a factor of 10 (from 12 to 90 kg m^{-3}) and its velocity from 0 to 4.3 m s^{-1}. The conditions of carrying out different series of experiments are given in Table 1. In the experiments, the dependence of the heat flux on the vapour–wall temperature difference, vapour velocity, physical properties of the cooling agents, location of a tube in a bank and the geometry of the bank was determined.

Preliminary results of one series of experiments for tubes 1, 3, 5 and 9 (counting from above) are given in Fig. 2 in dimensional coordinates. One can see that at a constant heat flux the vapour–wall temperature difference increases with the number of a tube in a bank and the slope of the curves changes.

In condensation of substances with a high vapour density (as in our experiments) the effect of vapour velocity on heat transfer makes itself felt appreciably at its low absolute values. This fact results in the following peculiarity that has been discovered in the experiments. The estimates of the film velocity from ref. [9] show that at small vapour velocities at the channel inlet the relative velocity of phases tends to zero as the film Reynolds number increases. The experimental value of the heat transfer coefficient under these conditions is close to that in stationary vapour condensation (Fig. 3). This figure presents experimental results on condensation of R21 vapour on banks of tubes placed in channels of different widths. One can see that the effect of vapour velocity is practically lacking at $Re > 70$ and the heat transfer in this case is governed by the condensate flow rate alone. The results obtained for banks placed in channels of different widths practically

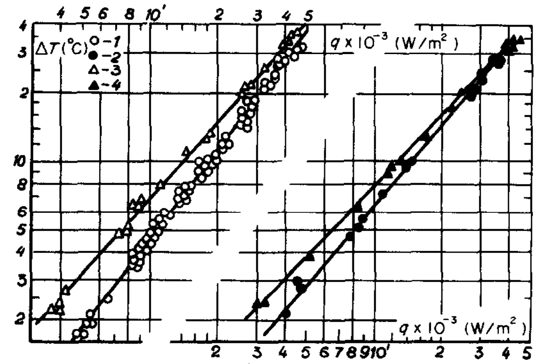

FIG. 2. Dependence of the vapour–wall temperature difference on heat flux in condensation of R12 vapour on a bank of tubes. $T'' = 60°C$, $x = 46$ mm (diagram 'a'), $w'' = 0.4$ m s^{-1}.

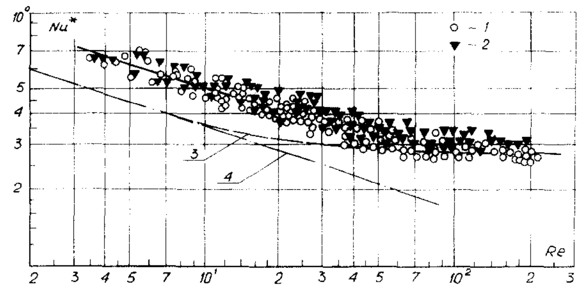

FIG. 3. Heat transfer in condensation of a flowing R21 vapour on banks placed in channels of different widths: $T'' = 60°C$, $w'' = 0.57$ m s^{-1}. 1, $x = 66$ (schematic diagram 'a'); 2, $x = 26$ mm (schematic diagram 'b'); 3, stationary vapour [1] (experiment); 4, calculation by the Nusselt theory.

coincided. One should bear in mind that at maximum heat fluxes the vapour velocity at the outlet from a narrow channel decreased by more than a factor of two from the original value, while in a wide channel it remained practically constant. The calculations show that at maximum heat fluxes in these experiments the value $Re \approx 70$ is attained on the third tube of the bank. Under these conditions the vapour velocity near the third tube of the bank changes only slightly even in the narrow channel. It is apparently for this reason that the experimental results obtained for condensation on banks placed in channels of different widths practically coincide.

Figure 4 shows the dependence of the averaged heat transfer coefficient on the vapour velocity at two substantially different vapour–wall temperature differences for all the tubes of the bank. Attention is turned to the different nature of the curves and to the weak effect of the quantity of tubes in the bank at high heat loads. At low vapour–wall temperature differences the effect of the number of tubes in a bank decreases with an increasing vapour velocity. At a maximum (in this series of experiments) vapour velocity of 4.3 m s^{-1}, there occurs either atomization of condensate droplets on the lower generatrix of the cylinder or their fragmentation. A portion of the liquid condensed did not reach the lower tube but flowed down the channel walls, as was observed visually. This phenomenon imparted an uncertainty into the condensate flow rate calculation and did not subsequently allow such experimental data to be processed in dimensionless coordinates. For this reason at this velocity the coefficients of heat transfer on the first and 10th tubes of the bank differed by no more than 20%.

The experimental results on condensation of R12 at the vapour velocities $w'' = 1.9$ m s^{-1} ($T'' = 40°C$) and $w'' = 2.0$ m s^{-1} ($T'' = 60°C$) have shown that liquid droplets generated under these conditions also disintegrate before their breakup under the action of the vapour flow. An appreciable amount of liquid flows down the channel walls. The results of these experiments are not presented in dimensionless form either.

The experimental data on condensation of flowing vapour of R21 on a 10-row bank (Fig. 5) show that at all vapour velocities the obtained curves $Nu^* = f(Re)$ tend to curve 2 which averages the experimental data on condensation of a stationary vapour. As the vapour velocity at the channel inlet increases, the value of the Reynolds number, at which the effect of vapour velocity on heat transfer practically disappears, increases too. As has already been mentioned above, it may be attributed to the fact that the vapour–liquid relative velocity tends to zero.

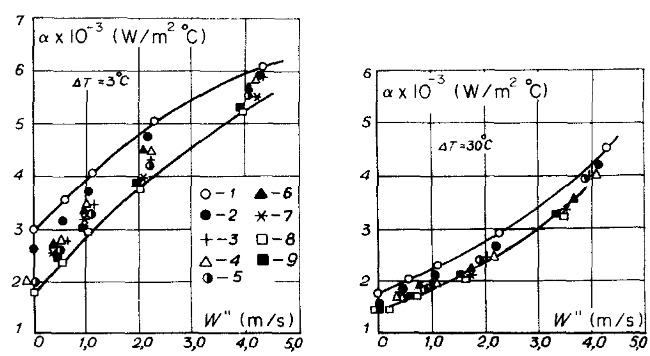

FIG. 4. Dependence of the averaged heat transfer coefficient on vapour velocity and on the number of a tube in a bank. R21, $T'' = 60°C$, $D = 16$ mm, $x = 26$ mm. Figures are numbers of tubes in the bank.

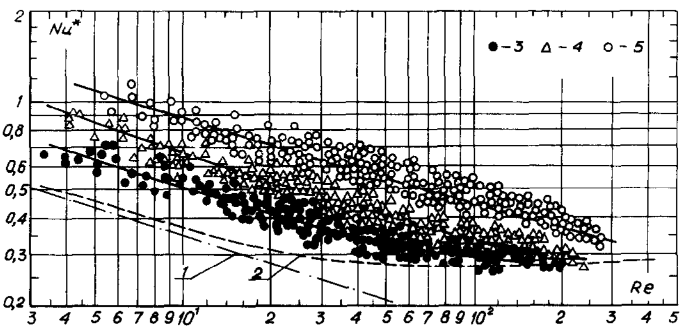

FIG. 5. Heat transfer in condensation of a flowing R21 vapour on a tube bank. 1, calculation by the Nusselt theory; 2, stationary vapour [1]; 3, $w'' = 0.57$ m s^{-1}; 4, $w'' = 1.14$ m s^{-1}; 5, $w'' = 2.3$ m s^{-1}.

If one refers approximate expressions (1) and (2) to Nusselt's formula, it may be seen that the relative value of the heat transfer coefficient in condensation of flowing vapour depends on one complex criterion which characterizes the friction on the phase interface:

$$M = \kappa^{4/3}\, Fr/Pr\, K \tag{4}$$

The program of experiments on heat transfer in condensation of R12 envisages the choice of such values of the vapour velocity at the channel inlet, that the values of the determining complex (4) be close to those obtained in the experiments with R21 (see Table 1). However, at a constant vapour velocity, the value of the parameter (4), first, somewhat varies with heat flux variation and, second, the nature of its change with variation of Re is different for different liquids. A change in complex (4) in condensation of R21 and R12 vapours attained $\pm 20\%$ of a certain mean value. Therefore, the measurement results on condensation of vapour of various liquids can be compared to a first approximation only.

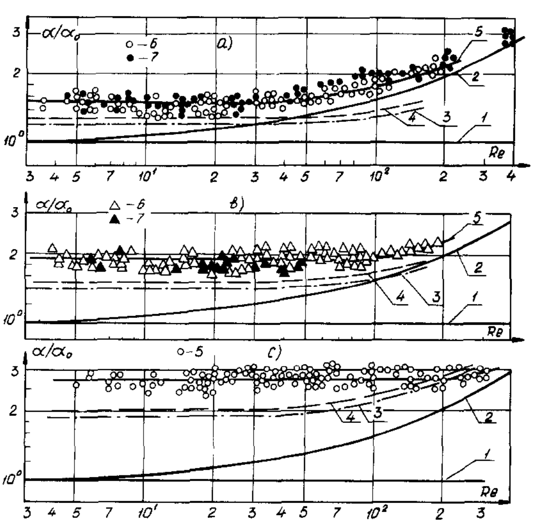

FIG. 6. Relative change in heat transfer by condensation of a flowing vapour. a, $M = 0.15$; b, $M = 0.6$; c, $M = 2.4$. 1, calculation by the Nusselt theory; 2, experimental data for a stationary vapour [2]; 3, calculation from (2); 4, calculation from (1); 5, experiments on R21 condensation [12]; 6, R12, $T'' = 40°C$; 7, R12, $T'' = 60°C$ [13].

Figure 6 presents the results of the processing of experimental data in the coordinates $Nu^* = f(Re)$ for three values of the parameter M. The value of α_0 has been calculated from Nusselt's formula. Here the values of the relative change in the heat transfer coefficient calculated from relations (1) and (2) have also been plotted. The comparison of the experimental and calculated coefficients of heat transfer has been carried out for the condition Re = idem. Both analytical relations do not take into account the effect of wave formation in a film on heat transfer and, therefore, in some cases when Re numbers are high, they give the values which are lower than those obtained in the experiment on condensation of stationary vapour. The pattern of curves for different coolants is similar. Lines 5 which average the experimental data for R21 are close to the experimental data on R12 condensation.

Figure 6 shows that the vapour velocity exerts an effect only up to certain values of the film Reynolds number, above which the heat transfer is determined only by the condensate flow rate. The influence of vapour velocity leads not only to an absolute increase in the heat transfer coefficient, but also to a change in the pattern of the function $Nu^* = f(Re)$. This change may conditionally be called the 'laminarization' of the condensate film. The dependence of the form $Nu^* \sim Re^{-1/3}$, which is valid for a laminar film, remains

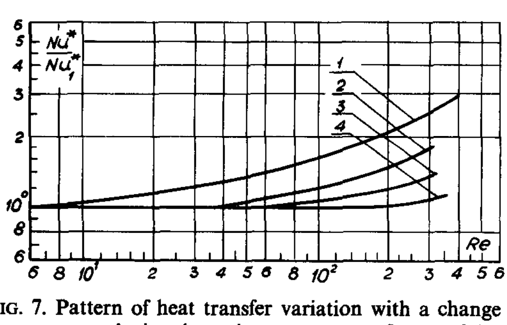

FIG. 7. Pattern of heat transfer variation with a change of the vapour velocity. 1, stationary vapour; 2, $M = 0.15$; 3, $M = 0.6$; 4, $M = 2.4$.

unaffected up to ever increasing values of Re with increasing vapour velocity.

The above fact is most clearly illustrated by Fig. 7. Here Nu^* is the averaged Nusselt number in the experiment on vapour condensation on a bank at a specified vapour velocity, Nu_1^* is the value of the Nusselt number determined by the relationship $Nu^* \sim Re^{-1/3}$ for the condition $Re = \text{idem}$ at the same vapour velocity. For a stationary vapour, the deviation of experimental data in the present experiments from a laminar mode is observed at $Re = 5$. At $M = 2.4$ the relationship $Nu^* \sim Re^{-1/3}$ is valid up to $Re = 200$.

CONCLUSIONS

(1) The results reported in the present paper and in refs. [1–3] make it possible to affirm that at the value of the complex criterion smaller than 0.05, vapour may be considered as practically quiescent. In this case the heat transfer in condensation on tube banks depends only on the condensate flow rate.

(2) When the vapour velocity is higher than the critical one, which is determined from equation (3), fragmentation of condensate droplets and streamlets by a vapour flow occurs. The heat transfer on each tube of a bank can be predicted to a first approximation as condensation on a single cylinder without taking into account the effect of the condensate flow rate.

(3) A relative change of heat transfer within the velocity range $0 < w'' < w''_{cr}$ depends on both the condensate flow rate, characterized by the film Reynolds number, and the friction on the vapour–film phase interface, characterized by the dimensionless complex $M = \kappa^{4/3}\,Fr/Pr\,K$.

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ETUDE DU TRANSFERT THERMIQUE DANS LA CONDENSATION EN FILM D'UNE VAPEUR TRAVERSANT UN ARRANGEMENT DE TUBES HORIZONTAUX

Résumé—On présente des résultats expérimentaux sur le transfert thermique avec condensation en film d'une vapeur qui s'écoule à travers un arrangement de tubes horizontaux. Les domaines de valeur des paramètres adimensionnels ont été déterminés de façon que le transfert thermique soit virtuellement gouverné par le débit de condensat seul, ou la vitesse de vapeur seule, ou l'action commune de ces facteurs.

UNTERSUCHUNG DES WÄRMEÜBERGANGS BEI DER FILMKONDENSATION VON STRÖMENDEM DAMPF AUF HORIZONTALEN ROHRBÜNDELN

Zusammenfassung—Die vorliegende Arbeit berichtet über experimentelle Ergebnisse des Wärmeübergangs bei der Filmkondensation von strömendem Dampf auf horizontalen Rohrbündeln. Es werden die Bereiche der dimensionslosen Parameter bestimmt, bei denen der Wärmeübergang im wesentlichen allein durch die Kondensat-Strömungsgeschwindigkeit, allein durch die Dampfgeschwindigkeit oder gemeinsam durch beide Faktoren bestimmt wird.

ИССЛЕДОВАНИЕ ТЕПЛООБМЕНА ПРИ ПЛЕНОЧНОЙ КОНДЕНСАЦИИ ДВИЖУЩЕГОСЯ ПАРА НА ПАКЕТАХ ГОРИЗОНТАЛЬНЫХ ТРУБ

Аннотация—Излагаются результаты опытов по теплообмену при пленочной конденсации движущегося пара на пакетах горизонтальных труб. Установлены диапазоны значений безразмерных параметров, в которых теплообмен практически определяется только плотностью орошения или только скоростью пара, или совместным воздействием плотности орошения и скорости пара.