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HEAT TRANSFER IN FILM CONDENSATION OF SLOWLY MOVING VAPOUR

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Abstract – Based on considerable experimental evidence, the law (curve) is suggested for film condensation of vapour in the absence of an appreciable external friction on the gas–liquid interface. The curve clearly reveals the region of laminar-wave motion, an extensive region of quasi-selfsimilar heat transfer and the region of fully developed turbulent heat transfer. The law is shown to be qualitatively the same, and quantitatively very similar, for different geometries (a vertical wall, a packet of horizontal tubes).

NOMENCLATURE

C, constant in formula (7);
c, specific heat [kJ kg^{-1} K^{-1}];
D, tube diameter [m];
g, gravitational acceleration [m s^{-2}];
L, tube length [m];
Nu, Nusselt number $\left[= \dfrac{\alpha}{\lambda'}\left(\dfrac{\nu^2}{g}\right)^{1/3}\right]$;
Pr, Prandtl number $\left(= \dfrac{c\mu}{\lambda'}\right)$;
q, q_i, specific heat flux, and specific heat flux for a given tube of the packet, respectively [W m^{-2}];
Re, Reynolds number for vertical and horizontal packets of tubes $(= qL/\mu r$; $= \pi D \sum_1^i q_i/\mu r)$;
Re_*, critical Reynolds number of a film;
r, latent heat of evaporation [kJ kg^{-1}];
S, tube spacing [m];
t'', saturation temperature [°C];
$\bar{t}_s$, average wall temperature [°C];
Δt, vapour-wall temperature difference $(= t'' - \bar{t}_s)$ [°C];
v_s, rate of wall shear stress [m sec^{-1}];
We, Weber number $\left\{= D\left[g\dfrac{(\rho' - \rho'')}{\sigma}\right]\right\}^{1/2}$.

Greek symbols

$\alpha_0, \alpha_v, \alpha, \alpha_i$, coefficients of heat transfer [calculated from the Nusselt formula, formula (1), determined experimentally and determined for a given tube of the packet, respectively] [W m^{-2} °C^{-1}];
η_δ, dimensionless film thickness $\left(= \dfrac{v_s \cdot \delta}{\nu}\right)$;
λ, λ_0, wave length [experimental and calculated from formula (5)] [m];
λ', thermal conductivity of liquid [W m^{-1} K^{-1}];
μ, ν, dynamic and kinematic viscosity, respectively [Ns m^{-2} and m^2 s^{-1}];
ρ', ρ'', density of liquid and vapour, respectively [kg m^{-3}];
δ, film thickness [m];
σ, surface tension [N m^{-1}];
τ_s, shear stress [kg m^{-1} s^{-2}].

INTRODUCTION

THE PRESENT state of the art in condensation heat transfer is most completely covered in a recent publication [1] and in the well-known books on the heat transfer theory [2, 3].

The necessity for experimental studies pertinent to condensation on vertical tubes is dictated, as will be shown below, by a marked spread in the reported data, while to condensation on a horizontal packet of tubes, by a limited body of data obtained in a narrow range of the parameters studied.

The present experiments were performed on a closed Freon circulation loop, the description and testing procedures for which are reported in [4]. The working substances were Freon-21 ($CHFCL_2$) and Freon-12 (CF_2Cl_2). The use of these fluids, boiling far below room temperature and having increased purity (as high as 99.85% of the main product) permit a fairly easy removal of noncondensing impurities from the system.

Regular chromatographic analyses were made in the course of runs to control the liquid and vapour phase compositions.

The experiments involved determination of the temperature difference between the test section wall and the saturated vapour, $\Delta t = t'' - \bar{t}_s$, and of the heat transfer rate on each tube of the test section, q_i, W m^{-2}. The wall temperature was determined from the average readings of 5–12 thermocouples, while heating of the cooling water was measured by differential thermocouples.

In the experiments, the vapour temperature, measured by means of a thermocouple, was maintained in agreement with the saturation temperature, determined from the temperature dependence of pressure and readings of a reference pressure gauge.

Variation in t was achieved by variation of the temperature of the cooling water fed from a constant-level tank. The maximum error in the measurements of the heat transfer coefficient, $\alpha_i = q_i/\Delta t_i$, at $\Delta t \geq 2$ was estimated as 10%.

HEAT TRANSFER IN FILM CONDENSATION OF QUIESCENT VAPOUR ON VERTICAL SURFACES

It is known that the Nusselt formula [5] has a very limited region of applicability, since the condensate film fall in a purely laminar flow is realized at very small Reynolds numbers ($Re = qL/\mu r$).

At $Re \sim 5$, formation of waves is observed in a falling film that enhance the heat transfer rate. For the film Reynolds number, which characterizes the onset of rippling, P. L. Kapitsa [6] suggested a relation obtained by assuming a capillary nature of waves. Recent studies [7] have discovered gravitational waves on the surface of the falling film which are accompanied by capillary waves. To calculate heat transfer over the range of the film Reynolds number $5 < Re < 100$, the authors of [1, 10] recommend adding an empirical correction term in the Nusselt formula to allow for the effect of the waves on enhancement of heat transfer

$$\alpha_v/\alpha_0 = \varepsilon_v = Re^{0.04}. \quad (1)$$

With further growth of the film Reynolds number, the wave mode of the condensate flow is replaced by the turbulent mode. For the latter case, the process of heat transfer was studied in [2, 8–12]. The relationships suggested in these publications depict, to a certain extent, the following functional relation:

$$\frac{\alpha}{\lambda'}\left(\frac{\nu^2}{g}\right)^{1/3} = f(Re, Pr). \quad (2)$$

Condensation of quiescent vapours of different liquids on vertical tubes has been studied experimentally by many authors. Figure 1 shows the data for water vapour obtained in [13–19]. Attention is directed to the large spread of experimental points, especially in the range $10^2 < Re < 10^3$, which corresponds to the transition regime. A similar figure could be presented for vapours of other substances. The results of most of the studies were obtained in a narrow range of the Reynolds numbers of a film. Figure 1 also presents a comparison of the experimental data with the relationships given in [2, 8–10].

It can be seen from Fig. 1 that in the region of the laminar and laminar-wave regimes of the condensate film flow the experimental data are fairly well described by the Nusselt formula with the correction (1) for the wave flow of the film. Formulae suggested for the turbulent regime [2, 8–10] differ significantly in the region which corresponds to transition from the laminar-wave to the turbulent flow of the film. The validity of these formulae cannot be fully assessed on the basis of the experimental data presented due to the large spread of the experimental points.

Figure 2 shows the results of the present authors for water vapour [14] and Freon-21. The experiments with Freon were conducted on tubes of different lengths and diameters at condensation temperatures from 40 to 125°C. The Reynolds number in the experiments with water vapour ($t'' \simeq 100$°C) was varied from 25 to 1200, while with Freon-21, it lay in the range $10 < Re < 4300$, thus covering the laminar-wave, transition and turbulent film flow regimes. It can be seen that the above experiments are fairly well generalized in the coordinates $(\alpha/\lambda')(\nu^2/g)^{1/3} = f(Re)$. Comparison with the theoretical relations shows that, up to $Re \sim 100$, the experimental results agree very closely with those computed from the Nusselt formula with the correction for the wave motion, ε_v, equation (1). At $Re > 100$, the closest coincidence with experiment is provided by calculation according to [2] for a mixed regime of the condensate film flow, i.e. when the local heat transfer coefficient in the turbulent flow was calculated by the formula

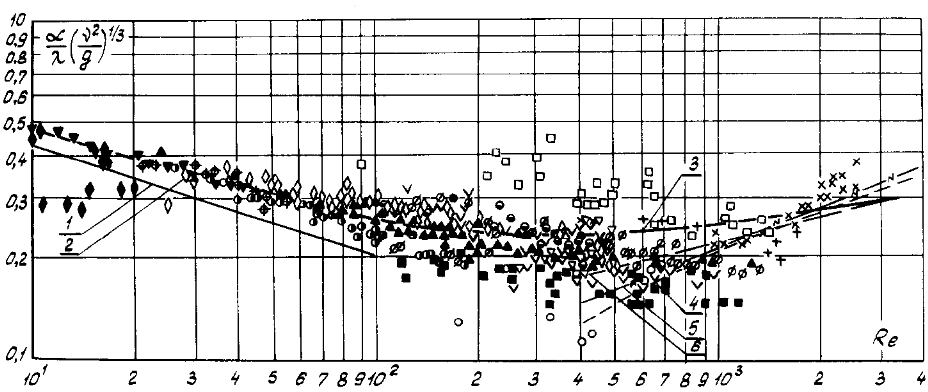

FIG. 1. Water vapour condensation on vertical tubes. ⌀—[13]; ▲—[14]; ×—[15]; ◑—[16]; ◇—[17]; ▼—[18]; ⊖—[19]. The data of the following authors are taken from [8, 17]: ●—Callendar and Nicolson; ○—Jordan; V—Hebbard and Badger; ▽—Lozhkin and Kanaev; ■—Baker and Stroebe; ⊕—Shea and Krase; ◆—English and Donkin; □—Fragen. Line 1, calculation according to [5]; 2, by formula (1); 3, [2]; 4, [9]; 5, [8]; 6, [10]. Curves 3–6, $Pr = 2$.

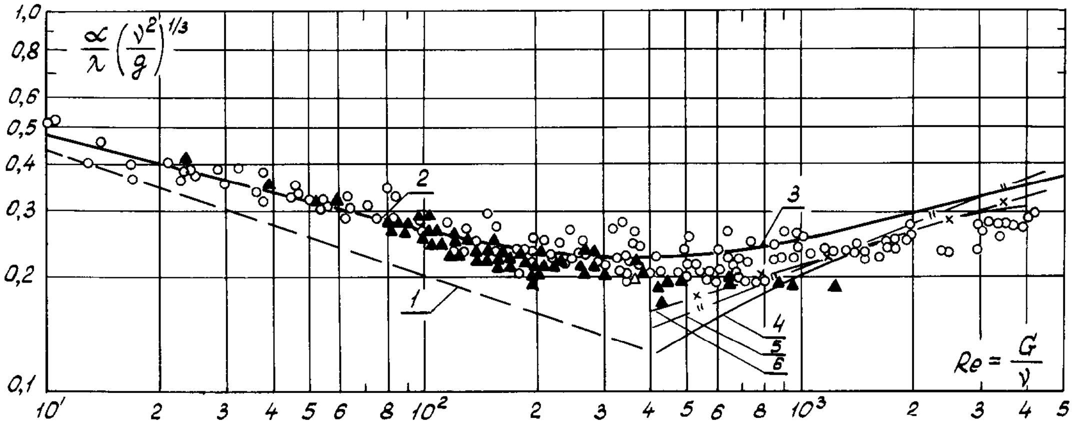

FIG. 2. Water vapour and Freon-21 condensation according to the present authors' data. ▲—water vapour [14], Pr = 1.75; ○—Freon-21, Pr = 3.3–3.5; 1, calculation according to [5]; 2, by formula (1); 3, [2]; 4, [9]; 5, [8]; 6, [10]. Curves 3–6, Pr = 3.

$$\frac{\alpha}{\lambda'}\left(\frac{v^2}{g}\right)^{1/3} = \frac{0.4\,Pr\,\eta_\delta^{1/3}}{\ln\dfrac{\sqrt{\eta_\delta}+\sqrt{\eta_\delta-11.6}}{\sqrt{\eta_\delta}-\sqrt{\eta_\delta-11.6}}+4.65\,Pr} \qquad (3)$$

where $\eta_\delta = (v_s\,.\,\delta/v)$ is the dimensionless film thickness,

$$v_s = \sqrt{\frac{\tau_s}{\rho}} = \sqrt{g\delta\left(1-\frac{\rho''}{\rho'}\right)}$$

is the rate of the wall shear stress. The critical Reynolds number was taken here to be $Re_* = 100$. At the same time it is known that Re_* for the gravitational film flow is about 400 [2, 20]. This discrepancy may be explained by the assumption that the waves on the outer surface of the condensate film extend the contact area and increase the effective value of its heat transfer coefficient. The latter may be regarded as evidence of the incipience of quasi-turbulent heat conduction, which allows formula (3) to be extended to the Reynolds numbers starting from $Re = 100$.

As is seen from Fig. 2, there is a vast region of the Reynolds numbers, $100 \leqslant Re \leqslant 1000$, in which heat transfer rate is practically constant; this appears to be clearly discovered for the first time in [23].

HEAT TRANSFER IN FILM CONDENSATION ON A SINGLE HORIZONTAL CYLINDER

It is known that, besides the gravity forces and viscosity allowed for by the Nusselt equation, the hydrodynamics of the condensate flow from horizontal tubes is in many respects governed by the surface tension of liquids. Analysis of the reported experimental results shows that they were obtained under conditions with the surface tension forces of slight effect, $We > 10$, where $We = D[g(\rho'-\rho'')/\sigma]^{1/2}$. The aim of the present series of experiments was a systematic experimental study of heat transfer in the condensation of Freon-21 on tubes of different diameters to determine the effect of surface tension. The experiments were performed at the saturation temperatures t'' = 40 and 60°C. The diameter of the test section varied from 1.5 to 43 mm and the Weber number was changed from 1.2 to 45. Simultaneously with heat transfer measurements, a systematic processing of the photographic pictures of the condensation process was made and the lengths of the waves of the falling films were measured. The wavelengths were measured for several liquids. Figure 3 shows the heat transfer data as the dependence of the heat transfer coefficient on the vapour–wall temperature difference [21]. It can be seen that this dependence is of the form $\alpha \sim \Delta t^{-0.25}$ for practically all the tubes studied, which is in agreement with [5]. A deviation from this dependence was observed for a 45 mm dia. tube, when the film Reynolds number exceeded a certain value above which there was a noticeable wave formation in the condensate film.

Figure 4(a) gives the heat transfer data for cylinders of different diameters processed in the coordinates

$$Nu = f(We) \quad \text{at } Re = \text{const.} \qquad (4)$$

where $Re = \pi Dq/\mu r$, and Fig. 4(b) shows the measurements of the wavelengths in the coordinates

$$\frac{\lambda}{\lambda_0} = f(We) \qquad (5)$$

where $\lambda_0 = 2\pi[3\sigma/g(\rho'-\rho'')]^{1/2}$ is the wavelength on a flat plate [22]. As follows from these plots, heat transfer and the relative value of the wavelength have a weak maximum at We = 2. At We < 2, both the wavelength in the film and the size of the separating drops decrease. Thus the analysis of the experimental data shows that the surface tension has an influence on the heat transfer and hydrodynamics of the film flow, but this effect is insignificant.

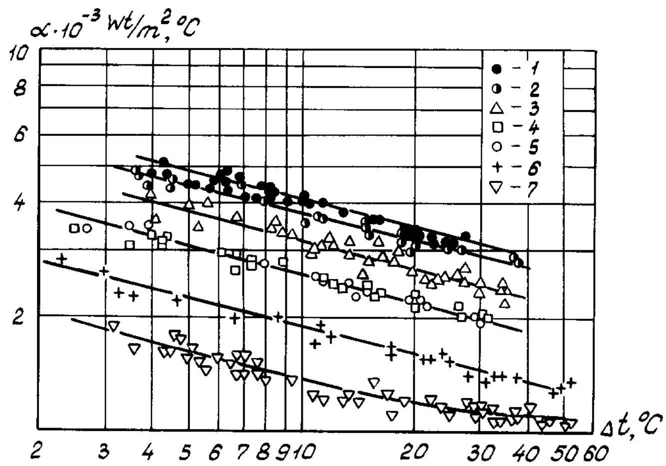

FIG. 3. Heat transfer in F-21 condensation on horizontal cylinders of different diameters at $t'' = 40$ and 60°C. $D = 1, 1.5; 2, 2.5; 3, 3.6; 4, 6; 5, 7; 6, 17$ and 7, 45 mm.

HEAT TRANSFER IN FLOW CONDENSATION ON A PACKET OF SMOOTH HORIZONTAL TUBES

The experiments with Freon-21 and Freon-12 were performed on a 10-row packet of horizontal tubes. Some of the results are published in [24]. It is noted in this reference that the body of the reported data on vapour condensation on tube packets is very limited.

The test sections in the present experiments were smooth 16 × 14 mm nickel tubes or 8 finned tubes on top and 2 smooth tubes on bottom, on which the measurements were made. The latter arrangement was used for the experiments with large film Reynolds number. The sections were arranged in a row with the relative spacing $S/D = 1.87$.

The experimental values of the heat transfer coefficient for the first tube are 9% for F-21, and 10% for F-12, higher than those calculated according to [5]. With increase in the number of tubes in the packet the heat transfer coefficient decreases, with the dependence of α on Δt being different for different tubes of the packet.

Figure 5 shows the primary data on condensation of F-21. When for the first and the second tube $\alpha \sim \Delta t^{-0.25}$ (according to the theory), then for the subsequent tubes this dependence becomes weaker with increase in Δt, and for the bottom tubes the heat transfer coefficient is practically independent of both the temperature difference and the number of tubes in

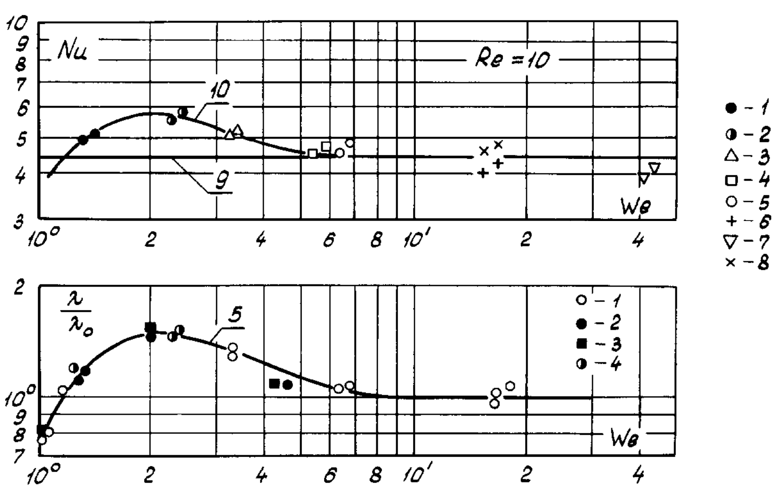

FIG. 4(a). Data on heat transfer at cylinders of different diameters processed in the coordinates $Nu = f(We)$ at $Re = \text{const}$. $D = 1, 1.5; 2, 2.5; 3, 3.6; 4, 6; 5, 7; 6, 17; 7, 45$ and 8, 16 mm; 9, calculation according to [5]; 10 is the line averaging the experiment.

FIG. 4(b). Relative wavelengths in liquid condensation on cylinders of different diameters. 1, F-21; 2, hexane; 3, ethanol; 4, water; 5, line averaging the experiment.

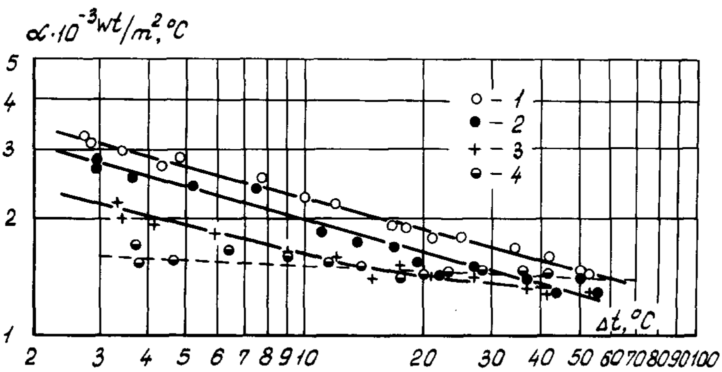

FIG. 5. Heat transfer coefficient, $\alpha \times 10^{-3}$ W m^{-2} °C^{-1}, as a function of temperature difference Δt, F-21, $t'' = 40$ and 90°C. Packet tubes: 1—No. 1; 2—No. 2; 3—No. 3; 4—No. 9.

the packet. A change in the character of this dependence is possibly caused by a change in the film condensate flow.

Figure 6 shows the data on condensation of Freon on a packet of horizontal tubes, obtained in the present work and processed in the coordinates

$$Nu = f(Re) \tag{6}$$

where $Re = \pi D \sum_{1}^{i} q_i/\mu r$.

In these experiments, the ranges of the Reynolds numbers for Freon-21 and Freon-12 were 3–840 and 7–2200, respectively. Figure 6 also contains the data on condensation of water vapour in the ranges $20 < Re < 80$ [26] and $13 < Re < 240$ [27] and of Freon-12 in the range $15 < Re < 190$ [25]. We were unable to use the results in the remainder of familiar publications, since they are lacking data on distribution of the heat transfer coefficient over the depth of the bundle. As is seen from Fig. 6, the data processed in coordinates (6) agree rather well amongst themselves at the constant Prandtl number. For water vapour $Pr = 1.75$ and for Freons $Pr \approx 3.5$. Depending on the value of Re, the condensate falls from the top tubes onto the lower ones in drops or streams. The experiments were performed with liquids which differed substantially in physical properties. Moreover, the ratio of the spacing between the tubes to their diameters, S/D, varied significantly, while in [26] this ratio was twice as large. The correlation obtained in Fig. 6 makes it possible to assert that the Nu number (mean for a given tube) is governed only by the Re and Pr numbers of the liquid and is practically independent of a change in the wavelength of the falling condensate film and the ratio S/D (with tube spacing being larger than the size of a separating drop).

At $Re \leqslant 10$, the experimental data are described by the relation corresponding to the laminar mode of the condensate flow [5]. With an increase in the Reynolds number, the dependence of the heat transfer coefficient on this criterion becomes less significant. At $100 < Re < 1000$, the heat transfer is practically constant. In studies on hydrodynamics of the vertical film flow [28–30], it is shown that the residual thickness of the film, which appears to present the main thermal resistance, changes but slightly in a wide range of Re numbers. That the residual film thickness remains conservative with a change in Re may also be deduced

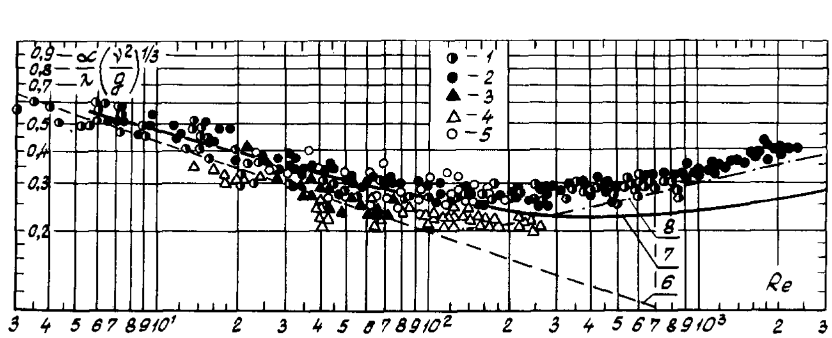

FIG. 6. Generalization of experimental data in coordinates (6) in condensation on a horizontal packet of tubes. 1, F-21 [24], Re = 3–840; 2, F-12, Re = 7–2200; 3, water vapour [26], Re = 20–80; 4, [27], Re = 13—240; 5, F-12 [25], Re = 15–190; 6, calculation according to [5]; 7, line averaging experimental data on condensation on vertical tubes (Fig. 2); 8, calculation of local Nu numbers in turbulent regime by equation (3).

from some wave characteristics, e.g. from the self-similarity of the phase velocity of small waves on the residual film thickness [31].

In the developed turbulent regime of the condensate film flow ($Re > 1000$), the heat transfer rate increases. Figure 6 also contains line 6, which is a theoretical Nusselt line for vapour condensation on a horizontal tube, and line 7, which averages the experimental data on condensation on the vertical tubes from Fig. 2 of the present paper.

As is seen from Fig. 6, the heat transfer correlations for horizontal and vertical packets have one feature in common, which is an insignificant change in heat transfer over a wide range of the film Reynolds number, $100 < Re < 1000$.

It should be noted that in the laminar range of the Re numbers, $Re < 10$, the heat transfer for these cases is practically described by one relation

$$Nu = CRe^{-1/3} \tag{7}$$

where $C = 0.925$ and 0.95 for vertical and horizontal tubes, respectively.

The local values of the heat transfer coefficients for the vertical turbulent film flow calculated by (3) are shown as line 8.

As is seen, the data for the packet of horizontal tubes, obtained with the film Reynolds number being based on the total condensate flow rate (6), and the data on local heat transfer for vertical walls, determined by (3), are nearly equal.

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TRANSFERT THERMIQUE LORS DE LA CONDENSATION EN FILM D'UNE VAPEUR EN MOUVEMENT LENT

Résumé — A partir d'une analyse d'expériences, la loi (courbe) est proposée pour la condensation en film d'une vapeur, en l'absence d'un frottement appréciable à l'interface gaz–liquide. La courbe révèle clairement la région du mouvement à onde laminaire, une région de transfert thermique quasi établi et la région de transfert thermique turbulent pleinement établie. La loi est qualitativement la même et quantitativement très semblable, pour différentes géométries (mur vertical, faisceau de tubes horizontaux).

WÄRMEÜBERGANG BEI FILMKONDENSATION VON LANGSAM STRÖMENDEM DAMPF

Zusammenfassung—Gestützt auf zahlreiche Versuchsergebnisse, wird eine Formel (Kurve) für die Filmkondensation von Dampf ohne nennenswerte Reibung an der Gas–Flüssigkeits-Grenzfläche vorgeschlagen. Die Kurve zeigt klar das Gebiet der laminaren Wellen, ein ausgedehntes Gebiet mit einem quasiähnlichen Wärmeübergang und die Zone des Wärmeübergangs mit vollständig entwickelter turbulenter Strömung. Es wird gezeigt, daß die Formel qualitativ dieselben und quantitativ sehr ähnliche Ergebnisse für verschiedene Geometrien liefert (vertikale Wand, horizontales Rohrbündel).

ТЕПЛООБМЕН ПРИ ПЛЕНОЧНОЙ КОНДЕНСАЦИИ МЕДЛЕННО ДВИЖУЩЕГОСЯ ПАРА

Аннотация — На основании большого экспериментального материала даётся закон (кривая) плёночной конденсации пара при отсутствии заметного внешнего трения на границе раздела фаз. Отчётливо выделяются области ламинарно-волнового движения, обширная область квазиавтомодельного теплообмена и область развитого турбулентного теплообмена. Показано, что этот закон качественно един и количественно близок для различных геометрий (вертикальная стенка, пакет горизонтальных труб).