A generalized correlation for two-phase forced flow heat transfer

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Abstract—A generalized correlation for forced convection vaporization heat transfer is suggested which is valid for both vertical and horizontal channels with a fully wetted perimeter. It represents, with a mean absolute deviation of 12.9%, experimental data for nine different fluids (water, freons, cryogens) in the following main parameter ranges: pressure 0.61–30.4 bar, mass flow rate 50–2690 kg m^{-2} s^{-1}, vapour quality 0.017–1.00, channel diameters 1.63–41.3 mm. The vaporization heat transfer rate is found to depend on the thermal conductivity of channel wall material. A universal criterion for convective boiling is suggested which determines the dominant mode of heat transfer of the two main ones: nucleate boiling or evaporation. Recommendations for calculating heat transfer to a two-phase flow in the presence of both heat transfer mechanisms are given.

INTRODUCTION

IT MAY be considered as an established fact at present that two main mechanisms participate in the transmission of heat in two-phase forced flow in a channel, one of which, nucleate boiling, is characterized by the presence of active nucleation sites on the heated wall and by a strong dependence of α on q, much the same as in pool boiling. The other mode of heat transfer, in which active sites are fully suppressed and heat is transferred by evaporation from the boundary between a liquid film and a vapour core, is called forced convection vaporization (convective boiling, two-phase forced convection). In the vaporization region, heat transfer does not depend on q and is governed in the main by the mass flow rate and vapour quality. This paper deals predominantly with the investigation of the latter heat transfer mechanism and determination of its boundaries.

Convective boiling is of great interest for those fields of engineering where it is required to remove high heat fluxes at extremely small wall–liquid temperature differences. Heat transfer coefficients in this region can exceed several times the values of α attainable in such a high-intensity process as nucleate boiling. Therefore, it is of great practical significance to have well-founded design recommendations which would allow this process to be calculated with a high degree of accuracy.

PREVIOUS STUDIES

More than 30 years have passed since the first mention was made about the existence of a purely convective mode of heat transfer in a high-speed two-phase flow [1, 2], and all this time the relevant literature was not short of various kinds of attempts at experimental data correlation. Excellent surveys of these correlations, pertaining mainly to steam–water flows, can be found in refs. [3, 4]. In this paper only some of these correlations [5–9] will be presented which, in principle, allow calculation of heat transfer of any evaporating liquid and which have been confirmed by a fairly large number of experimental data. The correlation proposed by Chen [5] and repeatedly modified by him [10, 11] ranks among the most suitable suggestions. It is written in the form

$$\alpha = \alpha_{FC}F + \alpha_b S. \tag{1}$$

It is assumed that the convection term can be found from the Dittus–Boelter equation

$$\alpha_{FC} = 0.023 Re_f^{0.8} Pr_1^{0.4} \frac{\lambda_1}{d} \tag{2}$$

irrespective of the flow regime determined by the single-phase Reynolds number. The parameter F has a purely hydrodynamic nature; it takes into account the enhancement of heat transfer due to an increase in the vapour quality (mixture velocity). The boiling term was calculated from the Forster–Zuber pool boiling equation the dimensional form of which is

$$\alpha_b = 1.22 \times 10^{-3} \frac{\lambda_1^{0.79} c_{pl}^{0.45} \rho_1^{0.49}}{\sigma^{0.5} \mu_1^{0.29} r^{0.24} \rho_v^{0.24}} \Delta T^{0.24} \Delta p^{0.75} \tag{3}$$

where Δp is the difference in saturation pressures which corresponds to ΔT. The parameter S in equation (1) takes into account the nucleate boiling suppression by forced convection because of the reduction in the thermal boundary layer thickness and, correspondingly, worsening of the conditions for the growth of vapour bubbles. Equation (1) correlates selected data on heat transfer to steam–water flows, organic fluids and freons, thus allowing many authors to recommend it as an engineering correlation [3, 12].

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NOMENCLATURE

a thermal diffusivity

Ar Archimedes number, $(gb^3/\mu_l^2)\rho_l(\rho_l-\rho_v)$

b Laplace constant, $\sqrt{(\sigma/g(\rho_l-\rho_v))}$

Bo boiling number, q/rG

Bo_* modified boiling number, $(rG/q)[1+x(\rho_l/\rho_v-1)]$

c_p specific heat at constant pressure

d channel diameter

Fr_m Froude number, $w_m/\sqrt{(gd)}$

g body force acceleration

G total mass flow rate

K_p dimensionless parameter, $p/\sqrt{(\sigma g(\rho_l-\rho_v))}$

K_s dimensionless parameter, $r/c_{pl}T_s$

l channel length

N_{CB} convective boiling number defined by equation (34)

Nu Nusselt number, $\alpha b/\lambda_l$

p pressure

Pe_* modified Peclet number, $qb/r\rho_v a_l$

Pr Prandtl number, $\mu c_p/\lambda$

q heat flux density

r latent heat of vaporization

Re_* modified Reynolds number, $qb/r\rho_v v_l$

Re_f superficial Reynolds number, $G(1-x)d/\mu_l$

Re_m Reynolds number of mixture, $w_m b/v_l$

T temperature

ΔT temperature difference, T_w-T_s

w_m two-phase mixture velocity, $(G/\rho_l)[1+x(\rho_l/\rho_v-1)]$

x quality (relative enthalpy)

X_{tt}^{-1} reciprocal Martinelli parameter, $(x/(1-x))^{0.9}(\rho_l/\rho_v)^{0.5}(\mu_v/\mu_l)^{0.1}$.

Greek symbols

α heat transfer coefficient

λ thermal conductivity; Schicht's map parameter

μ dynamic viscosity

ν kinematic viscosity

ρ density

σ surface tension

ψ Schicht's map parameter.

Subscripts

b   boiling

c   two-phase forced convection (vaporization)

cal   calculated

cr   critical

eq   equivalent

exp   experimental

FC   single-phase forced convection

l   liquid at saturation

m   mixture

s   saturation

TP   two-phase

v   vapour at saturation

w   wall.

Some authors believe [3] that the impressive accuracy (mean deviation of measurements from theory is less than 12%) achieved with the aid of Chen's equation can be attributed to the adequacy of the physical model. However, there are good reasons to offer more prosaic explanations, i.e. (1) the scarcity of experimental data used; (2) the application of the graphical method of correlation, which, from the viewpoint of the accuracy, is undoubtedly superior to analytical methods and especially to the common means of correlation based on simple power-law relations; (3) the fortunate choice of the relationship for nucleate boiling. Here, it is implied that the Forster–Zuber equation predicts the function $\alpha_b \sim q^{0.5}$ which is markedly weaker than those predicted by the majority of other pool boiling relations, but, as shown in refs. [9, 13] it fits precisely the case of forced convection boiling.

In 1976 Shah suggested [6] a graphical correlation which could successfully represent experimental data on water, freons, ammonia, hydrocarbons from 18 independent literature sources—more than 90% of 780 experimental points were found to be in the $\pm 30\%$ range. It should be noted, however, that an attempt to apply this correlation to the data on cryogenic liquids were not successful [14].

In ref. [7], an extended version of Rohsenow's well-known interpolation formula is given which, in the author's opinion, is valid for both nucleate and convective boiling regions. When $x > 0.05$, this formula has the following form:

$$q = q_c + q_b\left[1-\left(\frac{\Delta T_{ib}}{\Delta T}\right)^3\right] \qquad (4)$$

where ΔT_{ib} is the temperature difference at the start of boiling; the convective component can be found from

$$q_c = \frac{Re_f^{0.9}\, Pr_l\, F_1(X_{tt})\lambda_l}{F_2 d}\Delta T. \qquad (5)$$

Here F_1 and F_2 are the functions of the Martinelli parameter and of the Prandtl and Reynolds numbers. The quantity q_b can be calculated from Mikic–Rohsenow's equation [15]

$$\frac{q_b b}{r\mu_l} = B_M \frac{\lambda_l^{1/2}\rho_l^{17/8}c_{pl}^{19/8}\rho_v^{1/8}}{\mu_l r^{7/8}(\rho_l-\rho_v)^{9/8}\sigma^{5/8}T_s^{1/8}}\Delta T^3 \qquad (6)$$

where B_M is a dimensional constant being individual for every liquid. From comparison with a limited body of data on water Bjorge et al. [7] inferred that their correlation was more accurate than that of Chen because it gave a mean deviation of 15%, as against 17.4% in Chen's correlation. It is felt that this inference is not quite reasonable in a situation when the quantity of works on steam–water flow heat transfer considerably exceeds the number of publications (six) which were used by Bjorge et al. [7] for correlation. Moreover, equation (6) for q_b contains the dimensional constant B_M the value of which is known only for water. This restricts the applicability of the correlation only to steam–water flows.

Styushin [8] suggested a system of criteria which is mainly made up of similarity numbers that characterize the process of liquid pool boiling and which is augmented with Bo based on the mixture velocity. He believes that heat transfer in the region of developed nucleate boiling can be found from

$$St(K'_p)^{-1/3} = 1.25 Bo'(Pe'_*)^{-1/3} K_s^{0.5} \qquad (7)$$

where

$$St = \frac{\alpha}{c_{pl}\rho_1 w_m}; \quad K'_p = \frac{p}{\sigma}\left(\sqrt{\left(\frac{\sigma}{g\rho_v}\right)}\right);$$

$$Bo' = \frac{q}{r\rho_1 w_m}; \quad Pe'_* = \frac{q}{r\rho_1 a_1}\sqrt{\left(\frac{\sigma}{g\rho_v}\right)}.$$

Equation (7) is valid when

$$M = Bo'(Pe'_*)^{-1/3} K_s^{0.5} \geqslant 0.3 \times 10^{-5}$$

i.e. when heat transfer is governed entirely by the mechanism of nucleate boiling. With a decrease of M below the given value above there occurs a transition to the vaporization region, where the mass flow rate and the quality begin to be important and then

$$St(K'_p)^{-1/3} = 2 \times 10^{-3}[Bo'(Pe'_*)^{-1/3} K_s^{0.5}]^{0.5}. \qquad (8)$$

Equation (8) is based on the data of only two publications [16, 17] and has a very essential shortcoming, because it predicts rather a strong dependence of the heat transfer coefficient on heat flux in the region of convective boiling $\alpha \sim q^{1/3}$ in disagreement with the available experimental data according to which $\alpha \approx \text{const.}(q)$ [18–20].

Finally, in ref. [9] the following correlation was suggested which was based solely on the data for cryogenic fluids

$$\frac{Nu}{Nu_b} = \begin{cases} 1 & \text{for } Bo_* < 7 \times 10^4 \\ 0.0038 Bo_*^{0.5} & \text{for } Bo_* > 7 \times 10^4 \end{cases} \qquad (9)$$

where

$$Nu_b = 7.4 \times 10^{-3} Pe_*^{0.6} K_p^{0.5} Pr_1^{-1/3}(\lambda_w/\lambda_1)^{0.15}. \qquad (10)$$

In the purely convective boiling region $(Bo_* > 7 \times 10^4)$ this equation is based on the data taken from four

independent sources on heat transfer with neon [19] and nitrogen [21–23] two-phase flows. Its distinguishing feature is that it predicts an increase in the heat transfer rate with the thermal conductivity of the channel wall material. The existence of the effect was confirmed by special experiments for both nucleate boiling [24] and forced convection vaporization [25].

A more detailed analysis of the available predicting techniques was not the aim of the present investigation. Instead of it, in what follows comparison will be made between the available data on forced convection vaporization of different liquids and correlations will be given.

DEVELOPMENT OF CORRELATION

In a general case the heat transfer of a two-phase channel flow depends on a great number of different factors: heat flux, pressure, mass flow rate, quality, thermal properties of liquid and wall material, and channel geometry. This fact, together with a great number of different two-phase non-adiabatic flow patterns, hinders the construction of an adequate physical model of the process. However, by introducing reasonable physical abstractions it is possible to greatly simplify the problem and to obtain quite acceptable results.

Since the present analysis applies primarily to the region of evaporation, it will be assumed that the heat transfer coefficient is independent of the heat flux. This fact was observed by many researchers [1, 2, 18–20] and is hardly questionable. It will also be assumed that α is independent of the channel diameter. This assumption is more open to question, though there has also been experimental evidence of this [25–27]. The problem will be returned to at the end of the paper, but for the present it is sufficient to note that no appreciable influence of d has been observed in experiments, except for the very well-known work of Chawla [28] in which it was shown that an increase in the diameter of a horizontal channel resulted in the flow stratification and corresponding decrease of α.

The main regime parameters that determine the vaporization heat transfer are the mass flow rate and vapour quality. The effect of each of these was determined by applying the Dolphy technique consisting in the collection and generalization of closed expert assessments. These were taken to be the results obtained experimentally by eight groups of authors and approximated here in the form of power-law relations $\alpha \sim G^{m_1} x^{m_2}$. Then, it turned out that $m_1 = 0.61$ for all the data or $m_1 = 0.60$ when only the data for vertical channels were taken into account. Thus, the mean exponent at the quantity G turns out to be much smaller than for turbulent single-phase convection, being in disagreement with the majority of available correlations based on the Dittus–Boelter equation where $m_1 = 0.80$. Experimental data of the eight groups of authors gave $m_2 = 0.51$ and 0.58 for vertical channels only. This allows an inference that

G and x influence the heat transfer coefficient nearly to the same extent, thereby indicating that essential for the process is the product of these two quantities. To this conclusion there corresponds fairly well the model of homogeneous two-phase mixture flow in which the governing parameter is the mixture velocity

$$w_m = \frac{G}{\rho_1}[1 + x(\rho_1/\rho_v - 1)]. \tag{11}$$

The choice of the mixture velocity as the main parameter also accounts for the fact that the measured function $\alpha(x)$ is always somewhat weaker than $\alpha(G)$. The reason is that at small values of x and ρ_1/ρ_v the deviation of the function $\alpha(x)$ from a simple power law makes itself felt appreciably because of the commensurability of both terms in square brackets of equation (11). The above arguments allow the following heat transfer relation to be written within the framework of the homogeneous model:

$$Nu_c = c_1 Re_m^{0.6} Pr_1^{1/3}. \tag{12}$$

Since it has been assumed that $\alpha \neq \alpha(d)$, the centre of gravity of the solution is carried over to the selection of the characteristic linear dimension. Taking into consideration that the most probable two-phase mixture flow regime in convective boiling is a dispersed-annular one, two possible versions of this choice can be considered in the order of increasing x.

In subregime I there is a rather thick liquid film with intensive wave formation on the interface in response to the interaction with the flow vapour core [29]. In this case the critical length of the instability wave, and also the wave amplitude, at which the crest of the wave breaks up into droplets, turn out to be proportional to the Laplace constant b, which thus can be taken as the characteristic linear dimension.

As x increases, there occurs the transition to subregime II characterized by a thin film with a smooth interface. There occurs a sudden decrease in friction losses also, but the heat transfer coefficient continues to increase with vapour quality. This clearly attests to the fact that, in contrast to a single-phase flow, the coupling between the flow and heat-transfer modes in a two-phase flow is not a unique one and, consequently, the boundaries of these modes can be displaced relative to each other. In this subregime the smooth film thickness is determined by the interaction of gravity and viscosity forces which is characterized by the following linear scale:

$$b_* = \left(\frac{v_1^2}{g}\frac{\rho_1}{\rho_1 - \rho_v}\right)^{1/3}. \tag{13}$$

Thus, remaining within the framework of the homogeneous model, it is possible to assume that heat transfer will be described by the following relation:

for subregime I

$$Nu_c = c_1 Re_m^{0.6} Pr_1^{1/3}; \tag{14}$$

for subregime II

$$Nu_{c*} = c_2 Re_{m*}^{0.6} Pr_1^{1/3} \tag{15}$$

(here the asterisk means that b_* is taken to be the characteristic dimension). For practical purposes it is undoubtedly more convenient to use a single equation, the more so that the experiment does not detect any changes in heat transfer trends during the transition from one subregime to the other. Equation (15) can be reduced to the form

$$Nu_c = Nu_c(Re_m, Pr_1).$$

For this, multiply both sides by $b/b_* = Ar^{1/3}$ to obtain

$$Nu_c = c_2 Re_m^{0.6} Pr_1^{1/3} Ar^{2/15}. \tag{16}$$

Now, using Cooper's invariant equations [30] in the form suggested in ref. [31]

$$Ar \sim Pr_1^{-2.66}\left(\frac{\rho_1}{\rho_1 - \rho_v}\right)^{2.11}\left(\frac{\rho_v}{\rho_1}\right)^{-0.383} \tag{17}$$

$$K_p \sim \left(\frac{\rho_v}{\rho_1}\right)^{1.16}\left(\frac{\rho_1}{\rho_1 - \rho_v}\right)^{0.91} Pr_1^{0.181}. \tag{18}$$

It should also be taken into account that the two-phase forced convection regime occurs only at low reduced pressures for which

$$\frac{\rho_1}{\rho_1 - \rho_v} \sim K_p^{0.03}. \tag{19}$$

Solving the system of equations (17)–(19) for Ar and eliminating K_p and $\rho_1/\rho_1 - \rho_v$, it is possible to obtain that

$$Ar \sim Pr_1^{-2.65}\left(\frac{\rho_v}{\rho_1}\right)^{-0.307}. \tag{20}$$

The substitution of equation (20) into equation (16) yields the following relation for subregime II:

$$Nu_c \sim Re_m^{0.6} Pr_1^{-0.02}(\rho_v/\rho_1)^{-0.04} \tag{21}$$

or, neglecting the factors that have small powers

$$Nu_c = c_3 Re_m^{0.6}. \tag{22}$$

A very interesting result was obtained: heat transfer in the smooth thin film subregime is independent of Prandtl number. Moreover, comparison of equations (14) and (22) demonstrates that with transition from a wavy to a smooth film the structure of solution changes little, the change consisting in a decrease of the exponent of Pr_1 from 1/3 to 0. It may be presumed that, based on the mean exponent, the correlation will fairly well represent the entire region of convective boiling which includes both subregimes, i.e.

$$Nu_c = c_4 Re_m^{0.6} Pr_1^{1/6}. \tag{23}$$

It is clear that for heat transfer to be more accurately calculated, it is necessary to take into account, at least, two other important factors. First, it is the difference

between the real two-phase flow and its rather rough homogeneous model. This implies that the constant in equation (23) should be the function of the parameter which takes into account the interaction of both phases. Such a parameter may be represented, for instance, by the density ratio ρ_v/ρ_l. Second, as has been noted earlier [9, 13], the intensity of heat transfer during evaporation, just as during nucleate boiling, also depends on the thermal properties of the heated wall. This effect can be taken into account by means of the λ_w/λ_l ratio.

By representing the correlating equation in the traditional form of the product of power functions, the task is reduced to the determination of n_1 and n_2 in the relation

$$Nu_c = c_5\, Re_m^{0.6}\, Pr_l^{1/6}(\rho_v/\rho_l)^{n_1}\,(\lambda_w/\lambda_l)^{n_2} \qquad (24)$$

on the basis of comparison with experimental data given in Table 1. The use of the conventional procedure will eventually yield

$$Nu_c = 0.087Re_m^{0.6}\, Pr_l^{1/6}(\rho_v/\rho_l)^{0.2}\,(\lambda_w/\lambda_l)^{0.09}. \quad (25)$$

COMPARISON WITH EXPERIMENTAL DATA

Correlation (25) was constructed on the basis of forced convection vaporization data for nine different fluids (water, freons, cryogens) in different flow regimes (exclusive of stratification) in vertical and horizontal channels. In the latter case the flow patterns were identified with the aid of Schicht's map [51] modified by the present author on the basis of the data of Steiner and Schlünder [52] (Fig. 1). However, whenever the data of visual observations or measurements of the local (circumferential) heat transfer coefficient were available, which allowed the inference about the flow pattern, these were given preference. It should be emphasized that comparison of adiabatic flow pattern maps with the results of observation of diabatic flows reveals that the transition to the latter causes a noticeable expansion of the stratified flow pattern boundaries. In the author's opinion, it is quite natural, since the presence of active vaporization sites on the wall of the heated channel and an increased instability of the liquid–vapour interface increase the probability for the liquid film break-down and direct contact of vapour with the wall. Therefore, there seems to be no basis for Shah's recommendations [14] to use Baker's map [53] for identifying the diabatic flow patterns since the region of stratified flows on it is even narrower than on Schicht's map. All in all, 552 experimental points were processed for the vaporization region, with only one point deviating from correlation (25) by more than 35%. It should be pointed out that specifically excluded from consideration were rather a large number of data on the evaporation of freons in long tubes, in which lengthwise *averaged* heat transfer coefficients were measured, because then important information on the change of heat transfer regimes becomes lost. The only exception is ref. [36], taken as an example, all the rest data refer to lengthwise *local* heat transfer coefficients. The information processing involved those data which had been obtained both in the presence and in the absence of an adiabatic section upstream of the test section. In spite of the fact that entrainment in both cases can be quite different, the heat transfer coefficient is represented by a single correlation, equation (25), within $\pm 35\%$.

The mean absolute deviation given by

$$\bar{D} = \frac{1}{N}\sum_{i=1}^{N}\frac{|\alpha_{exp} - \alpha_{cal}|}{\alpha_{cal}} \qquad (26)$$

where N is the number of experimental points amounted to 12.9%, much in excess of the corresponding values given by other correlations (see Table 2).

Equation (25) is very special in that it predicts an increase in the rate of heat transfer with the thermal conductivity of the tube wall material. This is confirmed by Fig. 2 in which all the data from Table 1 are presented. The wall material effect is relatively small at the ambient temperature: the predicted heat transfer coefficients for water or refrigerants evaporating in a copper tube are about 30% higher than in a stainless steel channel; however, this discrepancy increases with a decreasing temperature and attains 70% for neon and 100% for helium. It should also be stressed that, unless precautions are taken in working with copper tubes, their surface becomes quickly covered with an oxide film, the thermal conductivity of which is very close to the thermal conductivity of steel. Naturally, the effect of the material then disappears. It is obvious that, to judge for certain about the wall material effect, it is necessary to have experimental information for tubes made of different materials but with identical geometrical characteristics within the same range of operational parameters. These results are presented in ref. [25]. They demonstrate that heat transfer coefficients for nitrogen evaporating in a copper tube ($\lambda_w = 248$ W $m^{-1}K^{-1}$) are 30–35% higher than in a stainless steel tube ($\lambda_w = 8.5$ W $m^{-1}K^{-1}$).

Equation (25) predicts a monotonous increase of the heat transfer rate with vapour quality up to the onset of the crisis (liquid film dryout). Conversely, Chen's and Shah's correlations predict the existence of the maximum heat transfer coefficient at $x = 0.80$ and a decrease of α with a further increase of x. It is difficult to imagine the reason for the heat transfer decrease at a fixed x regardless of the operational parameters and kind of fluid. Moreover, it is necessary to emphasize that a decrease of α with an increase of x nearly up to 1 has not been detected experimentally either in vertical [26, 27] or in horizontal channels [22, 32, 35], provided the perimeter was entirely wetted with liquid. Only a decrease in the circumferentially *averaged* heat transfer coefficient was recorded in hori-

Table 1. Experimental investigations analysed to develop the correlation suggested

Ordinal No.	Authors	Fluid	Test section; dimensions (mm); orientation	Material	Pressure (bar)	Mass flow rate ($kg\,m^{-2}\,s^{-1}$)	Vapour quality	Number of experimental points		Symbols
								Vaporization $N_{CB} > 1.6 \times 10^4$	Nucleate boiling $N_{CB} < 1.6 \times 10^4$	
1	Dengler and Addoms [1]	Water	Tube; $l = 6096$, $d = 25.4$; vertical	Copper covered by oxide film	2.2†	724	0.017–0.14	5	0	▲
2	Isbin et al. [32]	Water	Tube; $l = 235$, $d = 9.55$; horizontal	Brass	1.5–5.7	248	0.055–0.91	26	0	◑
3	Bennett et al. [33]	Water	Annular channel; $l = 737$, $d_{eq} = 4.47$ and 6.12; vertical	Stainless steel	1.5–1.8†	152–290	0.15–0.50	16	1	■
4	Collier et al. [34]	Water	Annular channel; $l = 1219$, $d_{eq} = 6.12$; vertical	Stainless steel	1.6†	136–258	0.025–0.60	10	7	◈
5	Rounthwaite [35]	Water	Tube; $l = 6248$, $d = 41.3$; horizontal	Mild steel	14.8	235	0.12–1.0	14	0	◬
6	Andreevsky and co-workers [26, 27]	Water	Tube; $l = 800$, $d = 8.12$ and 18; vertical	Stainless steel	4.1–30.4	183 2690	0.025–0.92	75	58	⊗
7	Aounallah et al. [20]	Water	Tube; $l = 457$, $d = 9.6$; vertical	Copper–nickel alloy	1.0–1.4	105 203	0.05–0.42	16	0	⊡
8	Bryan and Quaint [36]	Freon 11	Tube; $l = 3048$, $d = 8$; horizontal	Copper	1.2–1.6	137–587	0.085–0.50	17	0	⊕
9	Chawla [28]	Freon 11	Tube; $l = 119$, $d = 6$; horizontal	Copper	0.61	78.7–251	0.10–0.80	43	24	▶
10	Uchida and Yamaguchi [17]	Freon 12	Tube; $l = 1000$, $d = 6.4$; horizontal	Stainless steel	3.92	259–518	0.00–0.97	65	7	▽

11	Bandel and Schlünder [18]	Freon 12	Tube; l = (not given), d = 14; horizontal	Copper	3.13	104–441	0.50	17	10	×
12	Malyshev et al. [37]	Freon 12	Tube; l = 90, d = 10; horizontal	Stainless steel	1.63	224	0–0.72	14	3	*
13	Kirin et al. [38]	Freon 12	Tube; l = 400, d = 8; vertical	Copper	5.67	50 / 240	0.11–0.80	11	2	◑
14	Pujol and Stenning [39]	Freon 13	Tube; l = 1295, d = 15.8; vertical	Stainless steel	2.0	586	0.01–0.64	16	16	+
15	Altman et al. [40]	Freon 22	Tube; l = 1219, d = 8.7; horizontal	Copper	5.7 and 10.5†	146–284	0.31–0.65	3	3	◈
16	Troshin and Epishov [21]	Nitrogen	Tube; l = 900, d = 3.87; horizontal	Stainless steel	0.9–1.0	94.3–199	0.20–0.91	19	10	⊙
17	Klein [22]	Nitrogen	Tube; l = 1000, d = 12; horizontal	Copper	3.0	206–257	0.10–0.90	28	5	△
18	Klimenko and Sudarchikov [23]	Nitrogen	Tube; l = 1850, d = 10; vertical	Stainless steel	1.8–6.8	260–340	0.031–0.50	4	128	⊕
19	Grigoriev [41]	Nitrogen	Tube; l = 1630, d = 10; vertical	Copper	1.75–4.6	242–517	0.055–0.49	57	40	◕
20	Klimenko et al. [25]	Nitrogen	Tube; l = 500, d = 4, 9 and 15.8; vertical	Stainless steel	1.5–6.8	200–700	0.02–0.80	43	27	⊠
21	Muller-Steinhagen [50]	Argon	Tube; l = 175, d = 14; horizontal	Copper	1.67	447	0.05	4	0	▽
22	Mohr and Runge [19]	Neon	Tube; l = 100, d = 4; horizontal	Copper	1.5	77–131	0.31–0.85	42	27	●
23	Solodovnikov [42]	Helium	Tube; l = 180, d = 1.63; vertical	Stainless steel	1.02	87.4	0.32–0.68	8	2	>
	Total number of points for every region							552	370	
	Total number of points								922	

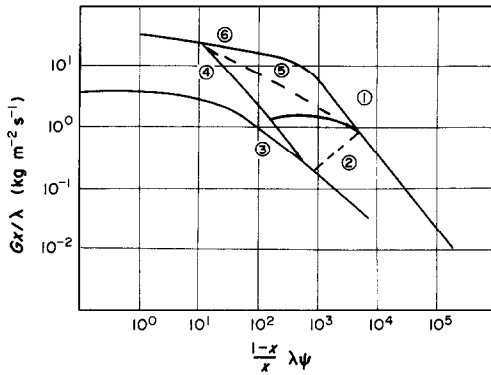† Mean value with the pressure drop over the test section taken into account.

FIG. 1. Modified map of flow patterns in a horizontal channel. Figures denote flow patterns: 1, bubbly; 2, plug; 3, stratified; 4, wavy; 5, slug; 6, annular. ——, Boundaries of flow patterns according to Schicht. ———, Boundaries of stratified and wavy flow patterns according to Steiner and Schlünder.

zontal tubes where this phenomenon can be attributed to the two-phase flow stratification.

As seen from Table 1, the available experimental data cover the range of channel diameters from 1.63 to 41.3 mm, including the range from 4.47 to 41.3 mm for the same liquid (water). A successful correlation of all these data with the aid of equation (25), which does not contain d, confirms the assumption made in the theoretical analysis that two-phase flow vaporization heat transfer does not depend on the channel diameter.

BOUNDARY BETWEEN NUCLEATE BOILING AND FORCED CONVECTION VAPORIZATION REGIMES

In this section a criterion will be presented with the aid of which it is possible to determine which of the two main types of heat transfer—nucleate boiling or vaporization—is dominating for a given fluid with a known set of operational parameters. This will be based on the natural assumption that the relative role of various heat transfer mechanisms can be determined by the ratio of corresponding heat transfer coefficients. The reference equation for the vaporization region is taken to be augmented with factors that take into account the interface friction and the influence of the thermal properties of the wall, i.e.

$$Nu_c \sim Re_m^{0.6} \, Pr_l^{1/3} (\rho_v/\rho_l)^{0.2} \, (\lambda_w/\lambda_l)^{0.09}. \quad (27)$$

The choice of equation (14), which describes heat transfer in subregime I, rather than a more general equation (25), is explained by the fact that the transition from nucleate boiling to evaporation occurs in this very subregime when, on the one hand, the liquid film is thick enough for bubble inception in it and, on the other hand, the two-phase mixture velocities can be sufficiently high to completely suppress nucleate boiling.

The nucleate boiling heat transfer rate will be estimated from equation (10). Taking into consideration that $Pe_* = Re_* \, Pr$ and omitting the numerical coefficient, this equation can be rewritten as

$$Nu_b \sim Re_*^{0.6} \, Pr_l^{0.267} \, K_p^{0.5} (\lambda_w/\lambda_l)^{0.15}. \quad (28)$$

Divide equation (27) by equation (28); the resulting ratio of heat transfer coefficients is essentially the sought-after criterion

$$\frac{Nu_c}{Nu_b} \sim \left(\frac{Re_m}{Re_*}\right)^{0.6} Pr_l^{0.066} \left(\frac{\lambda_w}{\lambda_l}\right)^{-0.06} \left(\frac{\rho_v}{\rho_l}\right)^{0.2} K_p^{-0.5}. \quad (29)$$

Neglecting the dependence on the thermal properties of the wall yields

$$\frac{Nu_c}{Nu_b} \sim \left(\frac{Re_m}{Re_*}\right)^{0.6} Pr_l^{0.066} \left(\frac{\rho_v}{\rho_l}\right)^{0.2} K_p^{-0.5}. \quad (30)$$

Table 2. Comparison of forced convection vaporization heat transfer correlations with experimental data

| Deviation parameter	Fluid	Correlation			
		Present paper	Chen	Shah	Styushin
Number of points outside	Water	0.6	29.0	29.6	27.8
±35% range, percentage to	Freons	0.0	22.6	32.1	61.8
number of data points	Cryogens	0.0	23.0	47.0	46.5
	All fluids	0.2	24.6	36.8	46.2
Mean absolute deviation (%)	Water	11.8	23.9	27.4	35.9
	Freons	13.7	23.2	31.4	54.9
	Cryogens	13.0	25.1	42.3	85.1
	All fluids	12.9	24.1	34.2	58.7
Maximum deviation,	Water	41	77	101	190
$(\alpha_{exp} - \alpha_{cal})/\alpha_{cal}(\%)$		−25	−59	−60	−23
	Freons	33	135	170	185
		−34	−48	−26	−27
	Cryogens	35	80	145	46
		−35	−61	−48	−43
	All fluids	41	135	170	460
		−35	−61	−60	−43
	Total number of experimental points	552			
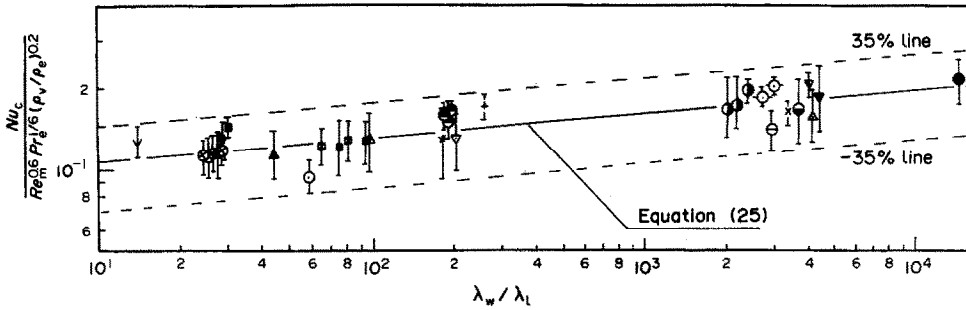
FIG. 2. Forced convection vaporization heat transfer rate vs the thermal conductivity of tube wall material. For symbols see Table 1.

On the right-hand side of equation (30) reconstruct the familiar parameter Bo_* which was successfully used [9, 27] for identifying the two-phase heat transfer regimes in water and cryogenic liquid flows. For this purpose, raise the right-hand side of equation (30) to the power 5/3 and take into consideration that $Bo_* = (Re_m/Re_*)(\rho_l/\rho_v)$; this yields

$$N_{CB} \sim Bo_* Pr_l^{0.11}(\rho_v/\rho_l)^{4/3} K_p^{-5/6} \qquad (31)$$

where N_{CB} is a universal parameter of convective boiling. Relation (31) can be considerably simplified. By combining equations (18) and (19) the following expression can be obtained:

$$K_p \sim (\rho_v/\rho_l)^{1.193} Pr_l^{0.186}. \qquad (32)$$

Substitution of this expression into equation (31) yields

$$N_{CB} \sim Bo_* Pr_l^{-0.045}(\rho_v/\rho_l)^{0.339}. \qquad (33)$$

By neglecting the inessential dependence on Prandtl number and rounding off the exponent at ρ_v/ρ_l to 1/3, it is possible to finally arrive at a rather simple expression

$$N_{CB} = Bo_*\left(\frac{\rho_v}{\rho_l}\right)^{1/3} = \frac{Re_m}{Re_*}\left(\frac{\rho_l}{\rho_v}\right)^{2/3}. \qquad (34)$$

The convective boiling number, defined by equation (34), allows the determination of the predominant mode of heat transfer in a two-phase flow; it is vaporization when $N_{CB} > (N_{CB})_{cr}$ and nucleate boiling when $Nu_{CB} < (N_{CB})_{cr}$. To verify this statement and to find the particular numerical value of $(N_{CB})_{cr}$, the experimental data from Table 1 (about 1000 points referring to both above-mentioned regions) were processed in the form of $Nu/Nu_c = f(N_{CB})$ presented in Fig. 3. It can be seen from this figure that at $N_{CB} \approx 1.5-2.0 \times 10^4$ there is a distinct change of one mode of heat transfer by another, with this transition being rather a 'sharp' one without any prolonged transitional region. This latter circumstance and the requirement imposed on the accuracy of the computational relation (at the level of $\pm 30-35\%$) make it possible to replace the transitional region by a point and to assign to it the value of the critical number N_{CB} equal to 1.6×10^4. This value is universal, since it is independent either of the kind of fluid, pressure or of

any other operational parameters varying within very wide ranges (see Table 1).

In the region $N_{CB} < 1.6 \times 10^4$, where evaporation is replaced by nucleate boiling, experimental points group rather densely around the line which corresponds to

$$Nu/Nu_c = 333 N_{CB}^{-0.6}. \qquad (35)$$

It is of interest that the empirical equation (35), which does not rely on any physical model of nucleate boiling at all, represents heat transfer in this region almost as accurately as equation (25) does in the evaporation region. In the author's opinion, this coincidence is not accidental. Really, let equation (35) be transformed into the system of dimensionless parameters that characterize the process of nucleate boiling. Combining equations (25), (34) and (35) gives, accurate to a coefficient, the following expression:

$$Nu_b \sim Re_*^{0.6} Pr_l^{1/6}(\rho_v/\rho_l)^{0.6}(\lambda_w/\lambda_l)^{0.09}. \qquad (36)$$

Expressing ρ_v/ρ_l through K_p and Pr_l from equation (32) and taking into account that $Re_* = Pe_*/Pr_l$, it is possible to obtain finally

$$Nu_b \sim Pe_*^{0.6} K_p^{0.5} Pr_l^{-0.39}(\lambda_w/\lambda_l)^{0.09}. \qquad (37)$$

Equation (37) virtually coincides with equation (10) obtained on the basis of theoretical analysis of forced flow nucleate boiling in tubes [31].

RECOMMENDATIONS FOR APPLICATION

To calculate heat transfer to a two-phase flow in vertical and horizontal channels in the case when the perimeter is fully wetted with liquid, the following correlations can be used:

$$Nu_{TP} = \begin{cases} Nu_b & \text{with } N_{CB} < 1.6 \times 10^4 \\ Nu_c & \text{with } N_{CB} > 1.6 \times 10^4 \end{cases} \qquad (38)$$

where Nu_b and Nu_c are found from equations (10) and (25), respectively.† Taking into consideration an

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† The condition $(N_{CB})_{cr} = 1.6 \times 10^4$ is satisfied with rather a high, but still limited accuracy. Therefore, to exclude the unjustifiable discontinuity in the function $\alpha(x)$ in the region where $\alpha_b \approx \alpha_c$, the higher heat transfer coefficient should be selected of the two.
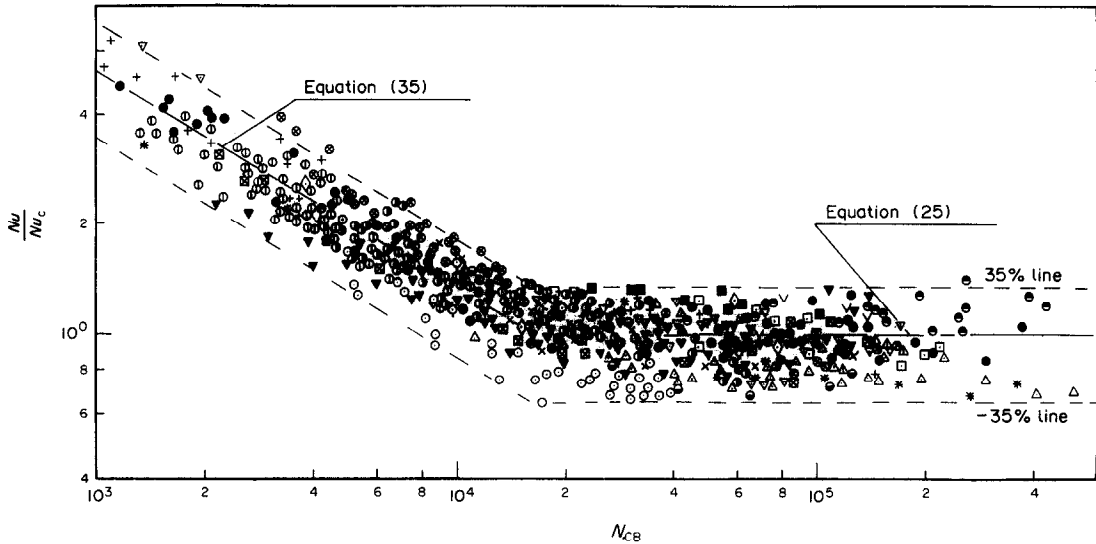
FIG. 3. Correlation of experimental data on two-phase forced flow heat transfer. Transition from nucleate boiling to forced convection vaporization. For symbols see Table 1.

extremely important role of water in modern engineering, the correlation suggested by Rassokhin et al. [43] for the water nucleate boiling region should be preferred, since, according to ref. [44], it represents the experimental data in the best possible way.

Expression (38) describes heat transfer in a 'normal' situation, when the growth of the two-phase mixture velocity ultimately results in a complete suppression of nucleate boiling. However, a quite different case is possible when a liquid film slows down its motion in a two-phase upward flow in large-diameter vertical channels. The film thickness increases in this case and there is no suppression at all of nucleate boiling. To this specific region, occurring, according to the author's observations, when $Fr_m < 10$, there corresponds the equation

$$Nu_{TP} = Nu_b \qquad (39)$$

for any values of N_{CB}, whereas the region of the applicability of equation (38) is restricted by the condition $Fr_m > 40$. When $10 < Fr_m < 40$, transitional modes of heat transfer take place.

All what has been said so far have related to the case when the single-phase convection contribution to the total heat transfer was relatively small. When the role of single-phase convection becomes appreciable, the total heat transfer is determined by a simple superposition correlation [45]

$$\alpha_\Sigma = (\alpha_{TP}^3 + \alpha_{FC}^3)^{1/3} \qquad (40)$$

where α_{FC} is the 'all-liquid' heat transfer coefficient based on the total flow rate G for the given flow regime corresponding to the Reynolds number, $Re = Gd/\mu_l$.

The correlations obtained in the present work include altogether eight thermal properties of fluid and channel wall material. The accuracy of calculations by these correlations depends, to a con-

siderable degree, on the reliability of information about the thermal properties; the use of outdated sources may result in considerable errors. This can be illustrated by an example: according to the data of ref. [46] the Prandtl number of liquid nitrogen at $p_s = 1$ bar is equal to 2.08, while the up-to-date publications [47–49] report the value 2.61! Yet, the data of ref. [46] are in use even in comparative investigations of various predictive correlations [14] and this can result in incorrect conclusions. Therefore, it is strongly recommended that the use of the data obtained prior to 1976 be avoided in responsible calculations. Preference should be given to the data certified by the State Service for Standard Reference Data of the U.S.S.R. and by the National Bureau of Standards of the U.S.A.

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UNE FORMULE GENERALE POUR LE TRANSFERT THERMIQUE DANS LES ECOULEMENTS FORCES DIPHASIQUES

Résumé—On propose une formule générale pour le transfert de chaleur avec vaporisation en convection forcée dans les canaux verticaux ou horizontaux, avec le périmètre complètement mouillé. Elle représente, avec un écart moyen absolu de 12,9%, des données expérimentales pour neuf fluides différents (eau, Freons, cryogéniques) dans les domaines de variation des paramètres suivants : pression 0,61–30,4 bar, débit-masse 50–2690 kg m^{-2} s^{-1}, qualité de la vapeur 0,017–1,00, diamètre du canal 1,63–41,3 mm. Le transfert thermique par vaporisation dépend de la conductivité thermique du matériau constituant la paroi du canal. On suggère un critère d'ébullition convective pour déterminer le mode dominant de transfert de chaleur : ébullition nucléée ou évaporation. On donne des recommandations pour le calcul du transfert thermique pour un écoulement diphasique en présence des deux mécanismes.

VERALLGEMEINERTE BEZIEHUNG FÜR DEN WÄRMEÜBERGANG BEI ERZWUNGENER ZWEIPHASENSTRÖMUNG

Zusammenfassung—Eine verallgemeinerte Beziehung für den Wärmeübergang bei der Verdampfung unter erzwungener Konvektion wird vorgeschlagen. Sie ist sowohl für vertikale als auch horizontale Kanäle mit vollständig benetztem Umfang gültig. Die Beziehung gibt mit einer mittleren Abweichung von ± 12,9% experimentelle Daten für neun verschiedene Fluide (Wasser, Freone, kryogene Fluide) in den folgenden Parameterbereichen wieder : Druck 0,61–30,4 bar ; Massenstromdichte 50–2690 kg m^{-2} s^{-1} ; Dampfgehalt 0,017–1,0 und Kanaldurchmesser 1,63–41,3 mm. Der Wärmeübergang bei der Verdampfung hängt vom Wärmeleitvermögen des Kanalwandmaterials ab. Ein allgemeingültiges Kriterium für das Sieden bei erzwungener Konvektion wird vorgeschlagen, welche die vorherrschende Art des Wärmeübergangs bestimmt : Blasensieden oder stilles Sieden. Es werden Empfehlungen für die Berechnung des Wärmeübergangs bei Zweiphasenströmungen für beide Wärmeübergangsmechanismen gegeben.

ОБОБЩЕННОЕ СООТНОШЕНИЕ ДЛЯ РАСЧЕТА ТЕПЛООТДАЧИ К ДВУХФАЗНОМУ ПОТОКУ ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ

Аннотация—Получено обобщенное соотношение для расчета теплоотдачи при испарении в условиях вынужденной конвекции, справедливое для вертикальных и горизонтальных каналов с полностью смоченным периметром. Соотношение со средним абсолютным отклонением 12,9% описывает экспериментальные данные для девяти различных жидкостей (вода, фреоны, криогены) в следующем диапазоне основных параметров: давление 0,61–30,4 бар, массовая скорость 50–2690 кг/м^2 с, паросодержание 0,017–1,00, диаметр канала 1,63–41,3 мм. Установлено, что интенсивность теплоотдачи при испарении зависит от теплопроводности материала стенки канала. Разработан универсальный критерий конвективного кипения, определяющий, какой из двух основных видов теплообмена: пузырьковое кипение или испарение является доминирующим. Даны рекомендации для расчета теплоотдачи к двухфазному потоку при наличии обоих указанных видов теплообмена.