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Film boiling heat transfer for liquid flowing with high velocity†

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Abstract—The state-of-the-art for research on flow film boiling heat transfer is discussed briefly in this paper. Emphasis is placed on reviewing our current studies on this field during the last 10 years, for subcooled or saturated liquid flowing with higher velocity either along a flat plate or through a circular tube or flat duct.

1. INTRODUCTION

FILM BOILING is known as an instance of the so-called 'Leidenfrost phenomenon', discovered in 1756 [1]. That is, liquid droplets dance on the surface of super-heated vapor will exist between the heating surface and the bulk of liquid; only part of the heat supplied by the heating surface penetrates to the liquid, as if the vapor film serves as a thermal insulation layer, and hence, the surface temperature rises much more than that for nucleate boiling. Film boiling occurs when the surface temperature exceeds the 'Leidenfrost temperature', which is, for example, about 275°C for water boiling at atmospheric pressure, much higher than the corresponding saturation temperature of boiling liquid.

For a long time, the Leidenfrost phenomenon was referred to as exceedingly curious but not particularly important by modern physicists due to its non-equilibrium characteristics in nature, and film boiling was generally considered as a dangerous case to be avoided in engineering applications. This is untrue because film boiling is really of 'technical importance', as quoted by Jakob [2] from Drew and Mueller's excellent review paper in 1937. Metallurgists had to face film boiling in studying quenching first, and the early research was therefore focused on pool film boiling, mostly studying experimentally the film boiling around electrically heated wires submerged in a liquid pool, for which Bromley [3] had derived a theoretical equation well known in the literature.

In recent years, because of its important applications in such fields as cryogenic engineering, space technology, nuclear reactor safety research, and especially the urgent need for developing new efficient cooling control technology of high-temperature quenching, the studies on liquid flow film boiling, or forced-convective film boiling, have drawn more and more attention. Bromley *et al.* [4] made an early analytical study on heat transfer for forced-convective film boiling of a saturated liquid. As regards the laminar-flow film boiling of a saturated liquid on a flat plate, an approximate relation was reported by Cess and Sparrow [5] as

$$\frac{Nu}{\sqrt{Re_1}} = 0.5\left[\frac{Pr_1\, h_{fg}}{C_{p1}(t_w - t_s)}\right]^{1/2} \tag{1}$$

for the case $(\rho_2/\rho_1) \gg 1$, i.e. liquid deviating greatly from its critical state, and with $t_{w,x} = \text{const.}$ Here $Nu = \alpha x/\lambda_1$, $\alpha = q_w/(t_w - t_s)$, t_w and t_s are wall and saturated liquid temperature respectively, while subscripts '1' and '2' refer to vapor and liquid respectively. Cess and Sparrow also analyzed the laminar-flow film boiling of subcooled liquid on a horizontal surface [6], and for boundary condition $q_{w,x} = \text{const.}$ [7]. Later, Jordan [8] and Kalinin [9] reviewed the works done in this field, and recommended equation (1) for predicting laminar-flow film boiling on a horizontal plate surface. However, laminar flow will change to turbulent flow at a certain critical distance from the leading edge along the flow passage, which is decreased as the free-stream velocity increases. As shown by Suryanarayana and Merte [10], laminar-flow film boiling, with a steady vapor–liquid interface, is limited to a comparatively small distance from the leading edge of the plate; in most cases, there will exist fluctuations at the two-phase interface, which direct the characteristics of turbulent film boiling from the start. Most of the research reported in the literature was concerned with turbulent-flow film boiling of liquid, saturated or subcooled slightly, and flowing with lower velocity. Fung [11] pointed out that we still knew very little about the characteristics of such a complicated two-phase flow heat transfer process, especially for subcooled liquid flowing at higher velocity. As a result, it is usual to calculate the heat transfer rate for turbulent film boiling of subcooled liquid flowing at high velocity with the well known empirical equation correlating heat transfer data for single-phase turbulent flow.

Recently, we have researched flow film boiling for

† Dedicated to Professor Dr.-Ing. Dr.-Ing.e.h. Ulrich Grigull.

NOMENCLATURE

a	thermal diffusivity
C_p	specific heat at constant pressure
D	diameter
g	gravitational acceleration
h	cross-sectional height of flat duct
h_{fg}	latent heat of evaporation
Ja	Jakob number
Nu	Nusselt number
Pr	Prandtl number
q	local heat flux
Re	Reynolds number
t	temperature
u	flow velocity along the surface
w	cross-sectional width of flat duct.

Greek symbols

α	local heat transfer coefficient
δ	vapor film thickness
λ	thermal conductivity
μ	absolute viscosity
ν	kinematic viscosity
ρ	density.

Subscripts

i	at inlet
sat	saturated condition
sub	subcooled condition
w	at surface
1	vapor
2	liquid.

a subcooled liquid flowing with higher velocity along a flat plate, or through a smooth circular tube or flat duct. We will summarize our main results below.

2. PRESSURE GRADIENT CAUSED BY INCREASE IN VAPOR FILM THICKNESS

We found from equation (1) that Nu will approach zero if the free-stream velocity becomes smaller and smaller, which is obviously not true in accordance with the real situation of pool film boiling. We adopt an analytical model as shown in Fig. 1. Since the vapor film thickness, δ, is a function of x only, we get the pressure gradient due to vapor film thickness increasing along the flow path as

$$\frac{\partial p}{\partial x} = -(\rho_2-\rho_1)g\frac{\mathrm{d}\delta}{\mathrm{d}x} \tag{2}$$

and the momentum equation in the vapor film is therefore

$$u_1\frac{\partial u_1}{\partial x}+v_1\frac{\partial v_1}{\partial y} = \nu_1\frac{\partial^2 u_1}{\partial y_2}+\frac{\rho_2-\rho_1}{\rho_1}g\frac{\mathrm{d}\delta}{\mathrm{d}x}. \tag{3}$$

The last term was not included in Cess and Sparrow's analyses. It is doubtless a buoyancy force, yet differs from the ordinary form related to natural convection. For a liquid deviating greatly from its critical state, i.e. $(\rho_2-\rho_1)/\rho_1$ having a large value, for example water at atmospheric pressure, the magnitude is about 10^5; the last term will be of the same order of magnitude as the rest of the terms and cannot be neglected in equations (3). Shi and I [12] presented, for the first time, the effect of such a pressure gradient on flow film boiling

$$Nu^5 = \frac{\sqrt{Re}}{\sqrt{\pi}}\left(\frac{\rho_2\mu_2}{\rho_1\mu_1}\right)^{1/2}\left[\frac{C_{p2}(t_s-t_f)Pr_1}{C_{p1}(t_w-t_s)Pr_2}\right]Nu^4$$

$$+\frac{1}{4}\frac{Pr_1\,h_{fg}}{C_{p1}(t_w-t_s)}Re_1\,Nu^3+\frac{1}{48}\frac{Pr_1\,h_{fg}}{C_{p1}(t_w-t_s)}Ar \tag{4}$$

where $Ar = g(\rho_2-\rho_1)x^3/(\rho_1\nu_1^2)$ is the Archimedes number, ν_1 is the kinematic viscosity of vapor, $Nu = \alpha x/\lambda_1$ and $\alpha = q_w/(t_w-t_s)$.

For saturation film boiling, the liquid temperature away from the surface, $t_0 = t_s$, equation (4) can then be simplified to

$$\left(\frac{Nu}{\sqrt{Re}}\right)^5 = \frac{Pr_1\,h_{fg}}{C_{p1}(t_w-t_s)}\left[\frac{1}{4}\left(\frac{Nu}{\sqrt{Re}}\right)^3+\frac{1}{48}\frac{Ar}{Re^{5/2}}\right]. \tag{5}$$

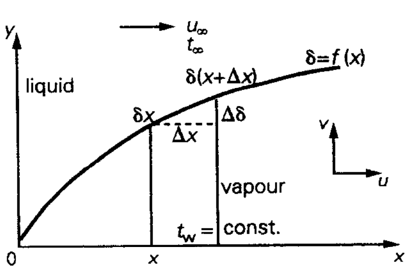

FIG. 1. Analytical model.

Figure 2 shows the variation of Nu as a function of Re_1 and $\varepsilon = Ar/(Re^{5/2})$. Ar is large enough in general; for example it reaches 10^{10}–10^{12}. It is clear that a small free-stream velocity, coinciding with a greater ε, could lead to an important effect on the heat transfer rate.

It can also be noted that: (i) as $\varepsilon \to 0$, i.e. the effect of pressure gradient on buoyancy force can be neglected, equation (5) reduces to equation (1), which was derived by Cess and Sparrow [5]; (ii) as the free-stream velocity decreases to zero, $Re \to 0$, i.e. for pool

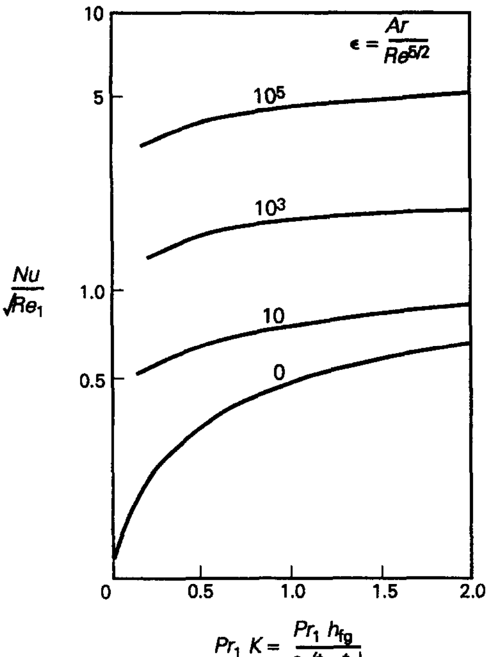

FIG. 2. Plot of equation (5).

film boiling, equation (5) reduces to

$$Nu^5 = \frac{1}{48} \frac{Pr_1 h_{fg}}{C_{p1}(t_w - t_s)} Ar. \tag{6}$$

For highly subcooled film boiling, the last two terms on the RHS of equation (4) can be neglected. Then

$$Nu = \frac{\sqrt{Re}}{\sqrt{\pi}} \left[\frac{(\rho\mu)_2}{(\rho\mu)_1}\right]^{1/2} \frac{C_{p2}(t_s - t_0) Pr_1}{C_{p1}(t_w - t_s) Pr_2}. \tag{7}$$

We have also analyzed successfully the case of $q_w = \text{const.}$ in a similar manner [13]. We may thus conclude that such analyses have effectively extended the classical results of Cess and Sparrow [5–7].

3. A PHENOMENAL MODEL PROPOSED FOR TURBULENT-FLOW FILM BOILING HEAT TRANSFER

It was found [14] that, for the laminar-flow film boiling of a subcooled liquid deviating greatly from its critical state, the velocity profile approaches uniformity within the liquid region, and the velocity gradient occurs mostly in the vapor film. This coincides qualitatively with numerical results reported by Ito and Nishikawa [15]. The velocity distribution within the liquid region will be more even for turbulent-flow subcooled film boiling, owing to turbulent mixing combined with evaporation of liquid and condensation of vapor at the liquid–vapor interface, which is in waves and fluctuates inherently. Hence, as shown in Fig. 3, it would be reasonable to divide the whole flow pattern into three regions, i.e. the vapor layer near the wall surface, the liquid flow region and the 'intermediate vapor–liquid mixing region' [14].

The 'vapor–liquid mixing region' is the region composed of bubble flow and fluctuating liquid–vapor interface, of which the thickness depends on free-stream velocity, u_0, and liquid subcooling degree, $(t_s - t_f)$. The vapor–liquid mixing process can be expected to be so strong that not only will the time–mean velocity distribution within this mixing region be uniform, but also the time–mean temperature distribution within this region can be considered as being constant and equal to saturated temperature, t_s.

Since the time–mean velocity distribution within the liquid flow region is considered as being uniform and kept constant as free-stream velocity, the turbulent boundary layer would then coincide with the vapor-film layer, within which the flow characteristic will be the same as that of a single-phase turbulent vapor flow along the solid surface. Meanwhile, owing to the subcooled liquid, liquid temperature will drop from saturated, t_s, at the liquid–vapor interface, to the temperature t_0 in the liquid flow region, such that the thermal boundary layer should be thicker than the boundary layer.

This proposed phenomenal model is simple in form and convenient to use. The higher the turbulent flow velocity and the greater the degree of liquid subcooling, the more realistic the model is. By this quite unique model, we have deduced semi-empirical relations to predict the heat transfer rate for film boiling of a subcooled liquid flowing with high velocity along a flat plate [14, 16] and through a smooth circular tube [17, 18] or horizontal flat duct [19–21] with the results agreeing well with experimental data.

4. TURBULENT VISCOSITY IN THE LIQUID REGION FOR FLOW FILM BOILING

According to the physical model suggested above, the energy equation in the liquid region for turbulent-flow film boiling of a subcooled liquid flowing along a flat plate should be

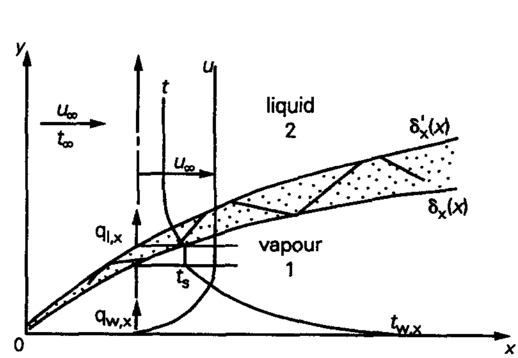

FIG. 3. Analytical model.

$$u_0 \frac{\partial t_2}{\partial y} = \frac{\partial}{\partial y}\left[(a_2+\varepsilon_{t,2})\frac{\partial t_2}{\partial y}\right] \tag{8}$$

where $\varepsilon_{t,2}$ is the turbulent thermal diffusivity of liquid. Noting that $\varepsilon_{t,2} \gg a_2$, and taking conventionally $Pr_t \equiv \varepsilon_m/\varepsilon_t \approx 1$, where ε_m is the turbulent viscosity for momentum exchange, equation (8) can be simplified to

$$u_0 \frac{\partial t_2}{\partial y} = \frac{\partial}{\partial y}\left[\varepsilon_{m,2}\frac{\partial t_2}{\partial y}\right]. \tag{9}$$

From the classical Karman theory, it can be written that

$$\varepsilon_m/\nu = f(y^+) \tag{10}$$

where $y^+ = (y/\nu)\sqrt{(S_{w,x}/\rho)}$ is a modified Reynolds number with shear-stress velocity $v_f = \sqrt{(S_{w,x}/\rho)}$. For forced-turbulent film boiling, the increase in vapor-film thickness along the flow direction will intensify the phase-transition process within the liquid–vapor mixing region; meanwhile, fluctuations at the vapor–liquid interface as well as vortex movement of liquid will spread rapidly into the liquid region, so that the turbulent viscosity of liquid becomes very large and approximately uniform at any given x-section, i.e. the turbulent viscosity may be independent of y at $y \gg \delta_x$. However, δ_x should depend on x, and $S_{w,x}/\rho$ may be a function of Re_x. Hence, for the liquid region, equation (10) can be expressed as

$$\varepsilon_{m,2}/\nu_2 = k\,Re_x^m. \tag{11}$$

The dimensionless coefficient k may be a weak function of physical properties of the specified liquid, liquid temperature subcooling and wall-surface temperature superheated. k can be taken as an empirical constant, to be determined experimentally.

The dimensionless exponent in equation (11), m, depends on the intensity of the liquid–vapor interface fluctuations, and for a specified plate or flow passage, may be a weak function of the vapor-film thickness, i.e. $m = f(\delta_x)$. For violently turbulent flow, m can approach a constant. As an example, Eckert and Drake [22] suggested a simple form, $\varepsilon_m \sim (u_0, x)$, for free-shear-layer flow, which is just a special case for $m = 1$ in equation (11). For turbulent film boiling of a liquid subcooled greatly, the increase of vapor-film thickness will be suppressed strongly, especially for the case of subcooled liquid flowing with high velocity along the wall surface, the vapor-film thickness, δ_x, itself would be small. Hence, m becomes a constant. It is predicted analytically that $m = 2/3$ for different subcooled liquids flowing with high velocity along a flat plate [16], and $m = 4/5$ or 0.8 for film boiling of a subcooled liquid flowing with high velocity through a circular tube [18] or flat duct [20]. The values determined from experiments were $m = 0.68$ for subcooled water [13, 14] and $m = 0.65$ for subcooled liquid R11 [23] flowing with high velocity along a flat plate, the average value being 2/3 and the deviation only $\pm 2.5\%$

5. PREDICTION OF FILM BOILING HEAT TRANSFER FOR LIQUID SUBCOOLED GREATLY AND FLOWING WITH HIGH VELOCITY

As shown in Fig. 3, there exists a balance relation of heat flux according to conservation of energy

$$q_{w,x} = q_{l,x} + q_{g,x} \tag{12}$$

where $q_{g,x}$ is the heat flux needed for evaporating the liquid, and $q_{l,x}$ is the heat flux transfer to liquid for supplying the sensible heat due to the temperature rise. For the case of film boiling of liquid subcooled greatly and flowing with high velocity along the wall surface, the vapor-film thickness δ_x would be small, and the increase of vapor-film thickness along the flow path is strongly suppressed, so that $q_{g,l} \to 0$, or $q_{w,l} \approx q_{l,x}$.

We deduced [16] an approximate solution for turbulent flow film boiling heat transfer of subcooled liquid along a flat plate in the form

$$Nu'_{x,2} = k'\,Re_{x,2}^{m'}\,Pr_2 \tag{13}$$

with

$$m' = (m+1)/2 = 5/6, \quad \text{or} \quad m = 2/3 \tag{14}$$

$$k' = \sqrt{[k(m+1)/\pi]} \tag{15}$$

and

$$k = c\left(\frac{\mu_1}{\mu_2}\right)^{1/3}\left[\frac{\rho_1 C_{p1}(t_{w,x}-t_s)}{\rho_2 C_{p2}(t_s-t_0)}\right]^{4/3}. \tag{16}$$

Here, c is an empirical constant to be determined experimentally, and $Nu'_{x,2}$ is a local Nusselt number defined as

$$Nu'_{x,2} = \frac{q_{w,x}}{(t_s-t_0)}\frac{x}{\lambda_2} \tag{17}$$

i.e. the heat transfer rate, α_x, is predicted on the basis of temperature difference (t_s-t_0). Equation (13) has been used quite successfully to correlate the experimental data of deionized water at 1 atm, with flow velocity in the range of 1–4.5 m s^{-1}, a temperature subcooled range of 22–72°C and wall-surface temperature range 300–1000°C as [14]

$$Nu'_{x,2} = 0.054Re_{x,2}^{0.84}\,Pr_2 \tag{18}$$

for $Re_{x,2} \geqslant 10^5$. As compared with equation (13), this measured value for an exponent of $Re_{x,2}$, $m' = 0.84$, is really the same as the analytically predicted value, $m' = 5/6$ or 0.833 in equation (14).

We also deduced the semi-empirical relations for film boiling of subcooled liquid flowing with high velocity through a circular tube [17, 18] or through a flat plate [19, 20] with the same form as equation (13) as

$$Nu'_{D,2} = b_1\,Re_{D,2}^{0.8}\,Pr_2 \quad \text{for a circular tube} \tag{19a}$$

or

$$Nu'_{h,2} = b_1\, Re_{h,2}^{0.8}\, Pr_2 \quad \text{for a flat duct} \tag{19b}$$

with

$$b_1 = c_1 \left(\frac{\nu_1}{\nu_2}\right)^{1/5} \left[\frac{\rho_1 C_{p1}(t_{w,x}-t_s)}{\rho_2 C_{p2}(t_s-t_1)}\right]^{4/5} \tag{20}$$

and

$$c_1 = f(\exp[-Re_D^{-1/5}\, x/D]) \quad \text{for a circular tube} \tag{21a}$$

or

$$c_1 = f(\exp[-Re_h^{-1/5}\, x/h]) \quad \text{for a flat duct} \tag{21b}$$

where D is the diameter of the circular tube, h is the height of the flat duct, and t_1 is the inlet temperature of subcooled liquid. c_1 will be a weak function of x/D or x/h for a specified subcooled liquid flowing with high velocity, and thus in turn, b_1 may be a constant for correlating experimental data [21, 23].

The forced-convective turbulent-flow film boiling heat transfer analyses we have so far reported in the literature are not confined to a horizontal plate, tube or duct only, but are also true for flow along an inclined and vertical plate or through an inclined (vertical) tube and duct in either direction, respectively. For example, we [23] have reported the results of experimental study on the steady turbulent film boiling of subcooled liquid R11 flowing upward in a vertical circular tube so as to determine the empirical coefficient. The pressure drop should be different, but due to the static head of the fluid column only.

It should be noted here, however, that equations (19a) and (19b) were derived for the case of D or h being far larger than δ_x. Also, there may exist a side-wall effect for flow film boiling heat transfer in a flat duct, so that c_1 may depend on the ratio of width and height, w/h, as shown in experiments [24], and the process will thus be three-dimensional, which needs to be further studied.

6. EFFECT OF LIQUID SUBCOOLING ON HIGH-VELOCITY FLOW FILM BOILING HEAT TRANSFER

The above-mentioned analyses are confined to the case of liquid being subcooled greatly, so that $q_{g,x} \to 0$ or $q_{w,x} \approx q_{1,x}$ from equation (12). If the liquid subcooling, $(t_s - t_0)$ or $(t_s - t_1)$, is down to zero, i.e. for flow film boiling of saturated liquid, $q_{1,x} \to 0$ or $q_{w,x} \approx q_{g,x}$ from equation (12). Hence, it is very clear that the ratio $q_{1,x}/q_{g,x}$ depends on liquid subcooling, which would therefore also play an important role in flow film boiling, like liquid velocity. This conclusion has been revealed by the numerical results of Ito and Nishikawa [15] and Srinivasan and Rao [25] for laminar flow film boiling. We analyzed the approximate solutions for film boiling of saturated liquid flowing

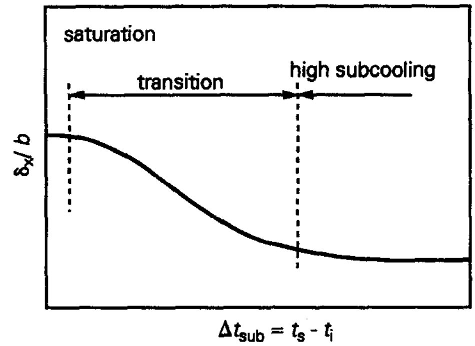

FIG. 4. The change of (δ_x, b) with liquid subcooling.

with high velocity along a flat plate [26] or through a circular tube [27] and in a flat duct [21].

We [26] found that the Jakob number of the liquid region, $Ja_2 = C_{p2}(t_s - t_0)/h_{fg}$, can serve as a characteristic criterion for reflecting the possible effect of liquid subcooling on flow film boiling heat transfer, and so $Ja_2 = 0$ for saturated liquid, while $Ja_2 \to \infty$ for infinite subcooling. We introduce a hypothetical relation, taking the case of a flat duct for illustration

$$\delta_x/h = (\delta_x/h)_{sat} \exp(-B\, Ja_2) + (\delta_x/h)_{sub}[1 - \exp(-B\, Ja_2)] \tag{22}$$

where B is an empirical coefficient to be determined experimentally, subscripts 'sat' and 'sub' denote liquid being saturated and highly subcooled respectively. Obviously, if $Ja_2 \to 0$, equation (22) evolves into $\delta_x = \delta_{x,sat}$, i.e. saturated liquid flow film boiling; if $Ja_2 \to \infty$, $\delta_x = \delta_{x,sub}$, i.e. highly subcooled liquid flow film boiling. However, there exists a critical value Ja_2^*, such that

$$\exp(-B\, Ja_2^*) = 0.01 \tag{23}$$

$\delta_x \to \delta_{x,sub}$ if $Ja_2 \geqslant Ja_2^*$. To this end, as shown in Fig. 4, the influence of liquid subcooling can be divided into 'transition' and 'high subcooling region' with Ja_2^* as a criterion. The theoretical and analytical results are verified by experimental data using liquid R11 with an empirical value of B around 65 [21].

7. EFFECT OF FLOW VELOCITY ON FILM BOILING OF SATURATED LIQUID

The phenomenal model quoted in Section 2 for $t_0 = t_s$ and methods proposed for subcooled liquid have been extended to analyze the turbulent film boiling of saturated liquid flowing with high velocity along a flat plate [13, 24] or in a circular tube [27]. The analytical results obtained for flow in a circular tube are as follows:

when $t_{w,x} = \text{const.}$

$$Nu_{D,1} = 0.0456 Re_{D,1}^{0.8}\, (x/D)^{-1/5}\, Pr_1\, Ja_1^{-1/5} \tag{24}$$

when $q_{w,x} = \text{const.}$

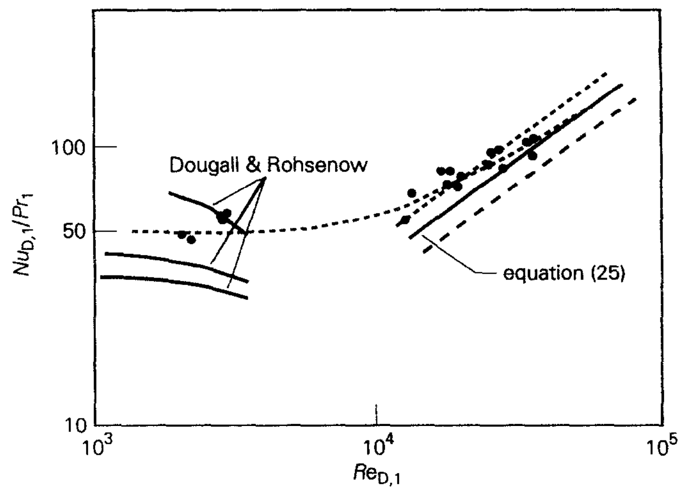

FIG. 5. Comparison of predictions and experimental data.

$$Nu_{D,l} = 0.0476 Re_{D,l}^{0.8} (x/D)^{-1/5} Pr_l \, Ja_l^{-1/5}. \quad (25)$$

Here, $Ja_l = C_{pl}(t_w - t_s)/h_{fg}$. Noting that $Nu_{D,l} \equiv Nu_{x,l}(D/x)$ and $Re_{D,l} \equiv Re_{x,l}(D/x)$, equations (23) and (24) can be transformed into equivalent ones for flow along a flat plate, i.e.

when $t_{w,x}$ = const.

$$Nu_{x,l} = 0.0456 Re_{x,l}^{0.8} Pr_l \, Ja_l^{-1/5} \quad (26)$$

when $q_{w,x}$ = const.

$$Nu_{x,l} = 0.0476 Re_{x,l}^{0.8} Pr_l \, Ja_l^{-1/5}. \quad (27)$$

This is rational, because the heat transfer relation for fluid flowing along a plate surface can be adapted for the tube flow near the entrance, whenever $\delta_x/D \ll 1$. It is also rational from comparing equation (24) with equation (25) that the heat transfer relations for the two extreme boundary conditions of uniform wall temperature and uniform wall heat flux are the same in form, and their coefficients differ from each other by only 4.4% due to highly turbulent flow. As a result, the effect of the actual wall thermal boundary condition is not prominent and can be ignored in engineering applications. This will also be true for the high-velocity flow film boiling heat transfer of subcooled liquid.

It is interesting to point out that the Nusselt number is proportional to $Re^{0.8}$, just like the convective heat transfer relation for single-phase turbulent flow. This supports the conventional empirical practices in correlating the high-velocity flow film boiling heat transfer data. However, as indicated by analytical solutions, the proportionality, i.e. $Nu/Re^{0.8}$ for flow film boiling, is related to Ja_l, being increased with an increase in $(t_w - t_s)$, and the corresponding heat transfer rate is obviously intensified.

Equation (25) is plotted in Fig. 5 with a bold line, and is compared with the reported experimental data of saturated liquid R113 flowing upwards in a vertical tube heated with uniform heat flux [28]. Considering the possible variations of the proportional coefficient including $Ja_l^{-1/5}$ along x due to the actual wall temperature distribution, the assessed extent of the theoretical prediction results by use of equation (25) is $\pm 10\%$ according to the Dougall and Rohsenow data. In the meantime, the Dougall and Rohsenow solution [28] is also plotted in Fig. 5 with fine lines. This makes clear that equation (25) tends to fit the experimental data well as $Re_{D,l}$ becomes higher than 5×10^4, but can be expanded downwards to 3×10^4, that is, the greater the value of $Re_{D,l}$, the better the coincidence.

The analytical results of Dougall and Rohsenow with buoyancy considered as the dominant effect [28] fit the experimental data only for the case of very low stream velocity, as shown in Fig. 5, and the deviation increases rapidly as the flow velocity becomes higher. We [12] have already pointed out that for liquid deviating greatly from its critical state and flowing with low velocity, the pressure gradient caused by a large ratio between the density of liquid and that of the vapor phase has an important effect on enhancing film boiling heat transfer, which can be expressed by use of the parameter $Ar/(Re^{5/2})$. Generally, Ar is of large magnitude, for example, it can reach 10^{10}–10^{11} for R113 at atmospheric pressure. So it may be a rational reason to explain why equation (25) deviates from the experimental data regularly as $Re_{D,l}$ decreases to below about 3×10^4. To this end, further work with experimental checks is still necessary to analyze this effect quantitatively.

8. CONCLUDING REMARKS

Flow film boiling heat transfer is not only of great importance for developing high technologies in engineering and industrial applications, but is also a very complicated transport phenomenon itself. We have here summarized briefly and recapped systematically our current research on this subject, especially film boiling heat transfer of subcooled liquid flowing with high velocity, in the last 10 years.

Although fruitful results have been obtained for predicting the heat transfer rate of subcooled (or saturated) high-velocity flow film boiling along a flat late or in a circular tube (or flat duct), further work is still necessary to clarify the problems as yet discussed.

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TRANSFERT THERMIQUE A L'EBULLITION DE LIQUIDE S'ECOULANT A GRANDE VITESSE

Résumé—On discute brièvement de l'état de la recherche sur l'ébullition en film avec écoulement. On met en relief les études sur ce domaine pendant les 10 dernières années pour un liquide sous-refroidi ou saturé s'écoulant avec une grande vitesse soit le long d'une plaque plane, soit à travers un tube circulaire ou une conduite aplatie.

WÄRMEÜBERGANG BEIM FILMSIEDEN SCHNELLSTRÖMENDER FLÜSSIGKEITEN

Zusammenfassung—In der vorliegenden Arbeit wird der Stand der Forschung auf dem Gebiet des Siedens strömender Filme kurz dargelegt. Den Schwerpunkt bildet ein Überblick über die eigenen Untersuchungen aus den letzten 10 Jahren auf diesem Gebeit. Diese Untersuchungen beschäftigten sich mit unterkühlten und gesättigten Flüssigkeiten, die mit hoher Geschwindigkeit entweder über ebene Platten, durch zylindrische Rohre oder ebene Kanäle strömen.

ТЕПЛОПЕРЕНОС В УСЛОВИЯХ ПЛЕНОЧНОГО КИПЕНИЯ ПРИ ИНТЕНСИВНОМ ТЕЧЕНИИ ЖИДКОСТИ

Аннотация—В настоящей работе кратко обсуждаются современные исследования по теплопереносу в условиях пленочного кипения при течении. Особое внимание уделятся обзору проведенных за последние 10 лет исследований в области интенсивного течения недогретой или насыщенной жидкости вдоль плоской пластины, а также по трубе круглого сечения или плоскому каналу.