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Turbulent film boiling heat transfer for liquid flowing with high velocity through a horizontal flat duct

X. F. PENG and B. X. WANG
Thermal Engineering Department, Tsinghua University, Beijing 100084, China

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Abstract—A unique semi-empirical expression is derived for turbulent film boiling heat transfer of saturated or subcooled liquid, flowing with high velocity through a horizontal flat duct. The investigation on the effect of liquid subcooling is advanced, and the influence of subcooling is divided into three zones with a distinct physical model and mechanism according to the characteristic parameter Ja_2. The theoretical and analytical results are verified by experimental data of liquid R11.

1. INTRODUCTION

IN RECENT years, to meet the need of development for improving and expanding productive technology, more and more attention is paid to the study on liquid flowing film boiling. Especially, to suit the needs of developing the efficient cooling-control technology of a high-temperature quenching process, which is regarded as one of the most important technical duties in modern metallurgical industries. Shi [1] and Peng [2] reviewed the early and recent research in this field. Wang and co-workers [3–13] reported the new achievements for film boiling of liquid flowing along solid surfaces. The important effect of pressure gradient, due to the increasing thickness of the vapour film layer and caused by a large difference between the density of liquid and that of the vapour phase, was investigated and theoretically analysed on laminar-flow film boiling [3, 4]. The physical model and analytical method have been proposed for forced-flow turbulent film boiling, and systemized semi-empirical heat transfer theories have been set up [5–13], which were verified by the experiments of both deionized water [1] and liquid R11 [2]. The analyses and experiments show that the forced-flow film boiling is in close relationship with liquid velocity and subcooling, etc. Buoyancy dominant for the low velocity case, inertia dominant for the high velocity case, and both dominant for the transition case. However, we did not know how the subcooling affects the film boiling. The research achievements reported previously [3–13] were limited to the two extremities, i.e. heat flux $q_w = q_g$ for the saturated condition, and $q_w = q_l$ for the high subcooled condition. The transition from low to high velocity is much more complicated and generally accepted experimental data are much less.

The numerical results reported previously [14, 15] show that liquid subcooling also plays an important role in flow film boiling as liquid velocity. Srinivasan and Rao [15] pointed out explicitly that, the heat flux $q_{w,x}$ transferred from the heated wall is provided partly to heat the subcooled liquid, $q_{l,x}$, and partly to evaporate liquid and to heat vapour in the film, $q_{g,x}$. The ratio of $q_{l,x}/q_{g,x}$ develops on liquid subcooling. Recently, a semi-empirical method was proposed to investigate the turbulent film boiling of subcooled liquid with high velocity flowing along a horizontal flat plate, a unique semi-empirical theory and a characteristic parameter Ja_2 were presented [16] to describe the effect of liquid subcooling from saturation to high subcooling on film boiling heat transfer, and it was pointed out that there is a characteristic value Ja_2^* [16] to distinguish the film boiling with high subcooling (i.e. $q_{w,x} = q_{l,x}$) from that with subcooling (i.e. both $q_{l,x}$ and $q_{g,x}$ dominant and $q_{w,x} = q_{l,x}+q_{g,x}$).

In this paper, the new investigation and analyses will be further advanced for turbulent film boiling of subcooled liquid flowing with high velocity through a horizontal flat duct. Also, we shall investigate the influences of liquid subcooling and the features of the characteristic parameter Ja_2 in much more detail.

2. FUNDAMENTAL CONSIDERATIONS AND MATHEMATICAL DESCRIPTION

The physical model adapted is kept the same as in refs. [5, 6, 9], shown in Fig. 1.

Supposing the width of the duct (in the z-direction), w, is so large compared with the height, b, that the influence of the side walls of the duct can be neglected, the analytical model will be thus reduced to the two-dimensional one.

The velocity distribution within the liquid region should become very complex in nature due to the existence of both the vapour–liquid interface and the insulated top cover plate. Fortunately, for the case of subcooled liquid flowing turbulently with high velocity, the time-mean velocity distribution within

NOMENCLATURE

a	thermal diffusivity
b	height of horizontal duct
c_p	specific heat at constant pressure
h_{fg}	latent heat of evaporation
Ja	Jacob number
Nu	Nusselt number
Pr	Prandtl number
q	local heat transfer rate per unit area
Re	Reynolds number
t	temperature
u	velocity
w	width of horizontal duct.

Greek symbols

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

Subscripts

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

the liquid turbulent core region can be considered as being uniform, and kept constant as the mean inlet velocity, u_i, and the velocity within the vapour–liquid mixing region [8, 9]. The time-mean temperature within the vapour–liquid mixing region can be considered as being constant and equal to the saturated temperature t_s. The turbulent boundary layer would be thus limited to within the vapour film layer. The flow characteristics in this region will be the same as that of single-phase turbulent vapour flow. But the temperature distribution will exist both in the liquid and vapour film layer.

According to the physical model and assumptions [9], the energy equation in the liquid flow region should be

$$u_i \frac{\partial t_2}{\partial x} = \frac{\partial}{\partial y}\left(\varepsilon_{M,2} \frac{\partial t_2}{\partial y}\right). \qquad (1)$$

We take the turbulent viscosity, $\varepsilon_{M,2}$, for the liquid flow region as [8]

$$u_i \delta_x / \varepsilon_{M,2} = \text{constant} \quad \text{or} \quad \varepsilon_{M,2} = c'' u_i \delta_x \qquad (2)$$

where c'' is a constant determined by experiments and δ_x the vapour film thickness.

The corresponding mathematical description had been given as [2, 9]

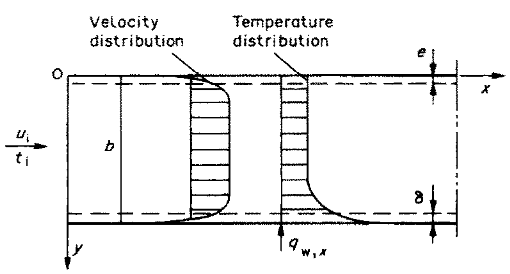

FIG. 1. Analytical model.

$$\frac{\partial t_2}{\partial x} = c'' \delta_x \frac{\partial^2 t_2}{\partial y^2} \qquad (3)$$

with boundary conditions

$$\left.\begin{aligned} &x = 0, \quad t_2 = t_i \\ &y = 0, \quad \partial t_2/\partial y = 0 \\ &y = (b-\delta_x) \approx b, \quad t_2 = t_s \end{aligned}\right\}$$

$$q_{w,x} = q_{l,x} + q_{g,x} \qquad (4)$$

$$q_{w,x} = 0.023 \rho_1 c_{p1} u_i (t_{w,x} - t_s)(u_i \delta_x / \nu_1)^{-1/4} \qquad (5)$$

$$q_{l,x} = -\rho_2 c_{p2} \varepsilon_{M,2} \left.\frac{\partial t_2}{\partial y}\right|_{y=(b-\delta_x)} \qquad (6)$$

$$q_{g,x} = \tfrac{7}{8} \rho_1 h_{fg} (\mathrm{d}\delta_x / \mathrm{d}x) \qquad (7)$$

with $x = 0$, $\delta_x = 0$.

3. APPROXIMATE ANALYSIS FOR FILM BOILING WITH HIGH VELOCITY

The turbulent-flow film boiling, especially for the case of both latent heat for evaporating and sensible heat for heating the subcooled liquid being equally dominant, is very complex, and the simplified mathematical equations are still difficult to solve analytically. Fortunately, our previous studies, including theoretical, numerical and experimental, have greatly achieved this, and some characteristics are recognized [5, 7]. So, it is possible and necessary for some advanced simplifications to be introduced to analyse the turbulent film boiling for some particular case.

3.1. *Approximate solution for saturated liquid flow film boiling*

When flowing liquid is in the saturated state, the heat would not be transferred into the stream, i.e.

$q_{\ell,x} = 0$, or from equation (6), $(\partial t_2/\partial y)_{y=(b-\delta_x)} = 0$. Thus equation (4) evolves to

$$q_{w,x} = q_{g,x} \quad (8)$$

and the mathematical descriptions could be given by equations (5), (7) and (8) for the film boiling of saturated liquid flowing with high velocity through a horizontal flat duct.

Substituting equations (5) and (7) into equation (8), we have

$$\left.\begin{aligned} \left(\frac{\delta_x}{b}\right)_{\text{sat}} &= 0.065\left[\frac{c_{p1}(t_{w,x}-t_s)}{h_{fg}}\right]^{4/5} Re_{b,1}^{-1/5}(x/b)^{4/5} \\ \text{or} \\ \left(\frac{\delta_x}{b}\right)_{\text{sat}} &= 0.065 Ja_1^{4/5} Re_{b,1}^{-1/5}(x/b)^{4/5} \end{aligned}\right\} \quad (9)$$

where subscript 'sat' expresses the saturated state. The Jacob number of vapour is given as

$$Ja_1 = c_{p1}(t_{w,x}-t_s)/h_{fg}.$$

Combining equations (9) and (5), we obtain

$$q_{w,x} = 0.0456\rho_1 c_{p1} u_i\, Ja_1^{-1/5} Re_{x,1}^{-1/5}. \quad (10)$$

Defining the Nusselt number as

$$Nu_{b,w} = \frac{q_{w,x}}{t_{w,x}-t_s}\frac{b}{\lambda_1} \quad (11)$$

the heat transfer relation for saturated film boiling would be

$$Nu_{b,w} = 0.0456 Ja_1^{-1/5}(b/x)^{1/5} Re_{b,1}^{4/5} Pr_1. \quad (12)$$

3.2. *Approximate solution for film boiling of subcooled liquid flow*

The film boiling heat transfer had been analysed previously [8] for high subcooling liquid flowing with high velocity through a flat duct, the wall heat flux is assumed transferred completely to the liquid region, i.e. $q_{g,x} = 0$, and the vapour film thickness is much thinner than that for saturated liquid flow film boiling. The transition for liquid being saturated to being highly subcooled would be much more complicated, for which the heat balance equation is given as equation (4), neither $q_{g,x} = 0$ nor $q_{\ell,x} = 0$. The vapour film thickness would be thinner than that of the saturated case and thicker than that of the highly subcooled case, and decreases from $(\delta_x/b)_{\text{sat}}$ to $(\delta_x/b)_{\text{sub}}$ with an increase in liquid subcooling. The relative vapour film thickness, (δ_x/b), will change with liquid subcooling as shown in Fig. 2.

For the extreme case, $q_{w,x} = q_{\ell,x}$, i.e. the flow film boiling for highly subcooled liquid, we obtain

$$\left(\frac{\delta_x}{b}\right)_{\text{sub}} = \left(\frac{0.023}{2c''c''_x}\right)^{4/5}\left(\frac{Ja_1}{Ja_2}\right)^{4/5}\left(\frac{\rho_1}{\rho_2}\right)^{4/5} Re_{b,1}^{-1/5} \quad (13)$$

where

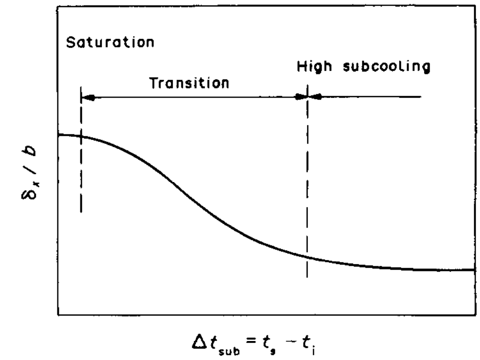

FIG. 2. The change of (δ_x/b) with liquid subcooling.

$$c''_x = \sum_{j=0}^{\infty} \exp\left[-k'' Re_{b,2}^{-1/5}\left(\frac{x}{b}\right)\left(\frac{2j+1}{2}\pi\right)^2\right] \quad (14)$$

$$k'' = c''\left(\frac{0.023}{2c''c''_x}\right)^{4/5}\left[\left(\frac{Ja_1}{Ja_2}\right)\left(\frac{\rho_1}{\rho_2}\right)\right]^{4/5}\left(\frac{\nu_1}{\nu_2}\right)^{1/5}. \quad (15)$$

Subscript 'sub' refers to subcooled liquid film boiling. As in ref. [16], we can express the relative thickness of the vapour film δ_x/b as the following weighted sum of equations (9) and (13):

$$\delta_x/b = (\delta_x/b)_{\text{sat}} \exp(-B\,Ja_2) + (\delta_x/b)_{\text{sub}}(1-\exp(-B\,Ja_2)) \quad (16)$$

where $Ja_2 = c_{p2}(t_s - t_i)/h_{fg}$ is the Jacob number of the liquid. The empirical coefficient, B, introduced here will be determined by experiments. Obviously, if $Ja_2 \to 0$, equation (16) evolves as $\delta_x/b \to (\delta_x/b)_{\text{sat}}$, i.e. saturated liquid flow film boiling equation (9); if $Ja_2 \to \infty$, equation (16) evolves as $\delta_x/b \to (\delta_x/b)_{\text{sub}}$, i.e. highly subcooled liquid flow film boiling equation (13).

Having introduced the semi-empirical relation equation (16), the heat flux from the flat wall can be determined directly from equation (5), or

$$q_{w,x} = 0.023\rho_1 c_{p1} u_i (t_{w,x}-t_s) Re_{b,1}^{-1/5} \times\left[0.065 Ja_2^{4/5}\left(\frac{x}{b}\right)^{4/5} e^{-B Ja_2} + \left(\frac{0.023}{2c''c''_2}\right)^{4/5}\left(\frac{Ja_1}{Ja_2}\right)^{4/5}(1-e^{-B Ja_2})\right]^{-1/4}. \quad (17)$$

According to equation (11), we have

$$Nu_{b,w} = 0.023 A_x^{-1/4} Re_{b,1}^{4/5} Pr_1 \quad (18)$$

where

$$A_x = 0.065 Ja_1^{4/5}\left(\frac{x}{b}\right)^{4/5} e^{-B Ja_2} + \left(\frac{0.023}{2c''c''_x}\right)^{4/5}\left(\frac{Ja_1}{Ja_2}\right)^{4/5}(1-e^{-B Ja_2}). \quad (19)$$

Substituting equations (9), (13) and (16) into equation (7), we obtain

$$q_{g,x} = \frac{7}{8}\rho_1 h_{fg} u_1 \frac{dA_x}{dx} Re_{b,1}^{-1/5} \tag{20}$$

or introducing

$$Nu_{b,g} = \frac{q_{g,x}}{t_{w,x}-t_s}\frac{b}{\lambda_1}$$

and

$$F_x = \frac{dA_x}{dx}$$

it would be clear that

$$Nu_{b,g} = 0.875 F_x Ja_1^{-1} Re_{b,1}^{4/5} Pr_1. \tag{21}$$

Rewriting equation (4) as

$$\frac{q_{l,x}}{t_{w,x}-t_s}\frac{b}{\lambda_1} = \frac{q_{w,x}}{t_{w,x}-t_s}\frac{b}{\lambda_1} - \frac{q_{g,x}}{t_{w,x}-t_s}\frac{b}{\lambda_1}$$

$$\text{or } Nu_{b,l} = Nu_{b,w} - Nu_{b,g} \tag{22}$$

then

$$Nu_{b,l} = (0.023A_x^{-1/4} - 0.875 Ja_1^{-1} F_x) Re_{b,1}^{4/5} Pr_1. \tag{23}$$

Equation (21) is the heat transfer relation for supplying both the latent heat of liquid evaporation and the sensible heat of vapour superheated, while equation (23) is the relation for the heat transferred into subcooled liquid bulk flow.

4. DISCUSSION

It is obvious from equations (18), (21) and (23) that $q_{w,x}$ and the distributive proportion of $q_{l,x}$ and $q_{g,x}$ will change with Ja_2 and would be different for actual liquid subcooling. That is, for the saturated liquid flow, $e^{-Ja_2} = 1$, equation (18) evolves to equation (12). On the other hand, for the extreme case of liquid film boiling with high subcooling, $e^{-Ja_2} \to 0$, or $q_{g,x} \to 0$ and $q_{w,x} = q_{l,x}$, equation (18) evolves to

$$Nu_{b,w} = 0.023\left[\left(\frac{0.023}{2c''c''_x}\right)\left(\frac{Ja_1}{Ja_2}\right)\left(\frac{\rho_1}{\rho_2}\right)\right]^{4/5} Re_{b,1}^{4/5} Pr_1 \tag{24}$$

or with defined 'local revised Nusselt number' [5, 6]

$$\widetilde{Nu}_b = \frac{q_{l,x}}{t_s - t_i}\frac{b}{\lambda_1} \tag{25}$$

equation (18) evolves to

$$\widetilde{Nu}_b = 0.0562(c''c''_x)^{-1/5}\left[\left(\frac{Ja_1}{Ja_2}\right)\left(\frac{\rho_1}{\rho_2}\right)\right]^{4/5} Re_{b,2}^{4/5} Pr_2. \tag{26}$$

Equation (24) or equation (26) is the heat transfer expression for the turbulent-flow film boiling of subcooled liquid flowing with high velocity along a horizontal flat duct reported in ref. [9], which has been verified by experimental results of both deionized water [1, 6] and liquid R11 [2, 9]. The empirical coefficient B in equation (16) has been estimated as around 60–70 [16] from the experimental data of deionized water and liquid R11. However, we know little about the performance and mechanism of Ja_2 affecting film boiling. Hence, theoretical investigations and experimental verifications are still to be advanced.

5. EXPERIMENTAL RESULTS WITH DISCUSSIONS

Steady-state experiments have been conducted on the turbulent film boiling of subcooled liquid R11 flowing through a horizontal flat duct. The testing installation is shown schematically in Fig. 3. A heat flux controller specially developed is put into use in the heating system to accomplish the steady heating and to prevent the test section from burning out [17]. The experimental system and apparatus have been described in refs. [9, 12] in more detail.

The test section, 19.2 mm high, 11.1 mm wide and 200 mm long, is installed horizontally. The bottom of the duct is heated by a heater with constant heat flux. The flow velocity of the subcooled liquid R11 in experiments ranges from 0.6 to 4.0 m s^{-1}, or $Re_{b,2} = 2.8 \times 10^4$ to 2.2×10^5, and Ja_2 ranges from 0.062 to 0.153.

In order to investigate the performance of Ja_2, introducing equation (25), we can rewrite equation (18) as

$$\left.\begin{aligned} &\widetilde{Nu}_{b,w} = 0.023\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{Ja_1}{Ja_2}\right)A_x^{-1/4} Re_{b,2}^{4/5} Pr_2 \\ &\text{or} \\ &\widetilde{Nu}_{b,w}/(Re_{b,2}^{4/5} Pr_2) = 0.023\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{Ja_1}{Ja_2}\right)A_x^{-1/4}. \end{aligned}\right\} \tag{27}$$

Clearly, for a highly subcooled liquid, $e^{-Ja_2} \to 0$, and

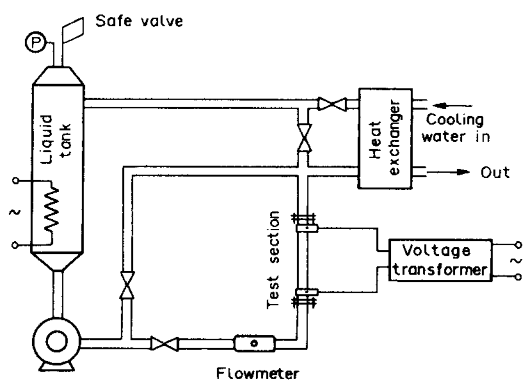

FIG. 3. The schematic diagram of the experimental unit.

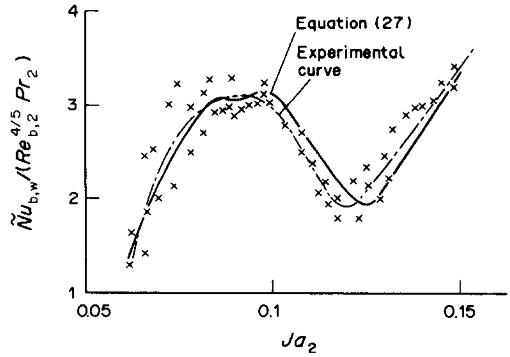

FIG. 4. Comparison of experimental and calculated results.

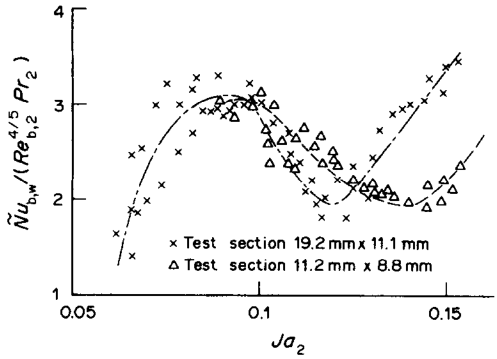

FIG. 5. Comparison of results for different experiments.

by equation (19), equation (27) changes to equation (26).

The experimental data and the calculated results from equation (27) are plotted in Fig. 4. Evidently, the experimental data verify the regularity theoretically predicted by equation (27).

We had pointed out in ref. [16] that, there exists a characteristic value, Ja_2^*, to distinguish 'highly subcooled' film boiling from subcooled film boiling. As a result, we can divide the subcooled film boiling further into three different zones with a distinguished physical model and mechanism: (1) $Ja_2 < 0.1$, 'convective vaporizing film boiling'; (2) $0.1 < Ja_2 < 0.12$, 'transition film boiling' relating to subcooling; (3) $Ja_2 > 0.12$, 'high subcooled film boiling'. In the convective vaporizing zone, the wall heat flux ($q_{w,x}$) partly provides the vaporizing latent heat of liquid and sensible heat for heating vapour, $q_{g,x}$, and partly provides the sensible heat for heating subcooled liquid, $q_{\ell,x}$, as described by equation (4). Of course, $q_{\ell,x}$ increases with increasing Ja_2, therefore changes the ratio of heat flux $q_{\ell,x}$ and $q_{g,x}$ accordingly. In the 'transition' zone, the subcooling of liquid is large enough to restrain the evaporation of liquid, and $q_{g,x}$ evidently decreases with increasing Ja_2, which $q_{\ell,x}$ increases in some degree, and so, the total heat flux, $q_{w,x}$, will decrease with increasing Ja_2. The high subcooled film boiling is formed when Ja_2 increases to a characteristic value Ja_2^*, and thereafter, the wall heat flux will be almost completely transferred to the liquid stream. In such a case, the total heat flux, $q_{w,x}$, depends on the liquid subcooling of Ja_2, and $q_{\ell,x}$ increases with increasing Ja_2. The characteristic value, Ja_2^*, suggested in ref. [16], may be used to distinguish the film boiling of 'highly subcooled' from that of subcooling liquid. As shown in Fig. 4, the experimental results make clear $Ja_2^* = 0.12$. Compared with the predicted value $Ja_2^* = 0.126$, the deviation is 5% only. The new experiments verify also that the empirical coefficient B ranges from 60 to 70.

Experiments have been conducted also on a test section, 11.2 mm high and 8.8 mm wide, to clarify the effects depending on the geometry and ratio b/w. The data obtained are shown in Fig. 5. We find the tendency is similar to that described above, but the range of transition film boiling and so the characteristic value of Ja_2^* is considerably different. The smaller the height and width of the test section and the lower the ratio b/w, the larger the value of Ja_2^* would be to form the 'highly' subcooled film boiling. The experiments reported previously [9] with a test section 19.2 mm high and 11.1 mm wide have clearly shown the general regularity similar to that in Fig. 4, the theoretical curve approaching the experimental one. It may be concluded that, the deviation of experimental data points from the predicted curve cannot be simply imputed to the accuracy of experiments or reasonableness of theoretical analysis. However, we still do not make it clear how the geometry and the ratio of b/w affects the transition film boiling quantitatively.

For the film boiling of liquid with high subcooling, equation (26) evolves to

$$\widetilde{Nu}_b/(Re_{b,2}^{4/5}\ Pr_2) = 0.0562(c''c''_x)^{-1/5}\left(\frac{Ja_1\ \rho_1}{Ja_2\ \rho_2}\right)^{4/5}. \tag{28}$$

In fact, to maintain high liquid subcooling it is required that the wall superheating $\Delta t_{sup} = (t_{w,x} - t_s)$ is much higher while Ja_2 increases, i.e. Ja_1 increases much more quickly than Ja_2, or Ja_1/Ja_2 increases with increasing Ja_2, and heat transfer is enhanced according to equation (28). After the film boiling of high subcooled liquid has been attained, with high flow velocity, wall temperature rises violently with a slight increase in heat flux, which is different from the 'critical heat flux' phenomenon.

Attention could be paid to the dispersion of experimental data and the deviation between experimental data and predicted results being considered concluded simply due to the uncertainty of experimental data as affected by several factors such as $Re_{b,2}$, x/b and the estimated value of B, etc., which are implicitly included in c''_x from equations (14) and (27). Besides, prior to the transition zone, as shown in Fig. 4, the theoretical curve emerges a temporary saddle-like change but increases generally with increasing Ja_2.

Perhaps, as the vaporizing latent heat and sensible vapour superheating heat, $q_{g,x}$, and the convective heat transferred to liquid, $q_{l,x}$, reach the same level, their interaction makes heat transfer undulate and results in some influences on wall temperature, etc., thus causing the heat transfer phenomena for flow film boiling of subcooled liquid to be further complicated.

6. CONCLUDING REMARKS

In this paper, the advanced analyses and experimental investigations have been carried out for turbulent film boiling of subcooled liquid flowing with high velocity through a horizontal flat duct. A unique semi-empirical expression, equation (18), including two special cases of saturation and of high subcooling, respectively, has been derived for predicting the effect of liquid subcooling to turbulent flow film boiling. This advances our previous works further. Especially, to clarify the regularity of the subcooling influence on turbulent flow film boiling would be of importance for application in engineering and industrial technology.

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TRANSFERT THERMIQUE PAR EBULLITION EN FILM TURBULENT POUR UN LIQUIDE S'ECOULANT A GRANDE VITESSE DANS UN CONDUIT PLAT HORIZONTAL

Résumé—Une expression unique semi-empirique est obtenue pour le transfert par ébullition en film turbulent avec un liquide saturant et sous-refroidi qui s'écoule à grande vitesse dans un conduit plat horizontal. L'étude de l'effet du sous-refroidissement est conduite et celui-ci est divisé en trois zones avec un modèle physique distinct et selon un mécanisme lié au paramètre caractéristique Ja_2. Les résultats théoriques et analytiques sont vérifiés par des expériences avec du R11.

WÄRMEÜBERGANG BEIM TURBULENTEN FILMSIEDEN EINER FLÜSSIGKEITSSTRÖMUNG IN EINEM WAAGERECHTEN FLACHKANAL BEI HOHER GESCHWINDIGKEIT

Zusammenfassung—Eine einzigartige halbempirische Beziehung für den Wärmeübergang beim turbulenten Filmsieden wird hergeleitet. Dabei wird der Fall einer gesättigten oder unterkühlten Flüssigkeit, die mit hoher Geschwindigkeit durch einen waagerechten Flachkanal strömt, betrachtet. Die Untersuchung des Einflusses der Flüssigkeitsunterkühlung wird weiter entwickelt. Der Einfluß der Unterkühlung wird in drei Zonen unterteilt, in denen jeweils ein unterschiedliches physikalisches Modell anhängig von der charakteristischen Kennzahl Ja_2 zu Grunde liegt. Die theoretischen und analytischen Ergebnisse werden mit Hilfe experimenteller Daten von flüssigem R11 bestätigt.

ТУРБУЛЕНТНЫЙ ТЕПЛОПЕРЕНОС В УСЛОВИЯХ ПЛЕНОЧНОГО КИПЕНИЯ ПРИ ТЕЧЕНИИ ЖИДКОСТИ С ВЫСОКОЙ СКОРОСТЬЮ В ГОРИЗОНТАЛЬНОМ ПЛОСКОМ КАНАЛЕ

Аннотация—Получено оригинальное полуэмпирическое выражение для турбулентного теплопереноса при пленочном кипении насыщенной или недогретой жидкости, текущей с высокой скоростью в горизонтальном канале. Исследуется эффект недогрева жидкости, при этом воздействие недогрева разделяется на три зоны с различными физическими моделями и механизмами в соответствии с характеристическим параметром Ja_2. Полученные теоретические и аналитические результаты подтверждаются экспериментальными данными для жидкости R11.