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A wall heat transfer model for subcooled boiling flow

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Abstract

The development of modern cooling systems aims in many cases at a most compact, space and weight saving design. The consequential demand for highest possible heat transfer rates has lead to the very promising concept of providing for a controlled transition from pure single-phase convection to subcooled boiling flow in thermally highly loaded regions. The application of this approach in modern engineering design requires a realistic modeling of the complex phenomena associated with the two-phase flow heat transfer. The present work proposes for the computation of the specific wall heat transfer rate a modified superposition model, where the total heat flux is assumed to be additively composed of a forced convective and a nucleate boiling component, respectively. Since the present model requires only local input quantities, it is well suited to CFD of geometrically very complex coolant flows, where the definition of global length or velocity scales would be very impractical. The wall heat fluxes predicted by the present model were compared to experimental data for the most part obtained by in-house measurements with water being the working fluid. The overall agreement is very good, particularly, in the partially nucleate boiling regime, where the effect of the bulk flow rate is significant. Deviations are primarily observed at higher wall superheats, where a strong two-way coupling between the motion of the liquid and the motion of the bubbles as well as considerable bubble-bubble interactions typically occur.

Introduction

During the last decade the investigation of subcooled boiling flow has been gaining attention. Aiming at highly efficient liquid cooling systems, be it for internal combustion engines, or, for microprocessors, inherently involves the optimization of the local heat transfer. Thus, whereever a high coolant power with limitations on the available surface area for the heat transfer, the mass flow rate of the liquid coolant as well as the acceptable wall temperatures is to be achieved, a controlled transition to the nucleate boiling regime offers an attractive solution. The concept of exploiting the markedly enhanced heat transfer rates associated with the highly complex phenomenom of evaporation is also a big challenge to the CFD of coolant flows. The method of direct numerical simulation, which attempts to resolve all physically relevant scales, is applicable only to strongly simplified cases like single bubble configurations, see [1]-[3]. For engineering flow configurations, where the technique of direct numerical simulation has to be ruled out for its exceeding computational cost, computationally feasible and at the same time accurate boiling models have to be provided to obtain reliable numerical results.

The location of subcooled boiling occurring in a channel heated from beneath is sketched for in Fig.1. In the subcooled region extending between position B and C, respectively, the wall superheat $T_w - T_s$ is sufficient to initiate and sustain nucleate boiling, while the temperature of the bulk liquid T_b remains below the local saturation temperature T_s. The corresponding subcooled boiling segment in the flow boiling curve is schematically shown in Fig.2. The lower boundary at point B marks the onset temperature T_{onb} of the partially developed boiling (PDB), where the boiling curve starts to deviate from the dashed-dotted extension of the almost straight single-phase line.

Nomenclature

Latin symbols

b	model constant [−]
c	specific heat [J/kgK]
C_s	constant [−]
D_{hyd}	hydraulic diameter [m]
F	force [N]
g	gravitational acceleration [m^2/s]
G_s	$= (du/dy)(y/u)$ shear rate [−]
h	heat transfer coefficient [W/m^{2}K]
h_{lg}	latent heat [J/kg]
k	thermal conductivity [W/mK]
K, n	constants [−]
p	pressure [N/m^2]
q	specific heat transfer rate [W/m^2]
R	radius [m]
S	suppression factor [−]
T	temperature [^{o}C]
t	time [s]
u	velocity [m/s]
u_τ	$= \sqrt{\tau_w/\rho}$ wall friction velocity [m/s]
V	volume [m^3]
x	axial coordinate [m]
y	wall normal coordinate [m]
y^+	$= \rho_l u_\tau y/\mu_l$ non-dimensional distance [−]

Greek symbols

α	thermal diffusivity [m/s^2]
κ, χ	constants [−]
μ	kinetic viscoity [kgm/s]
ρ	mass density [kg/m^3]
Φ, Ψ_{tp}	enhancement factors [−]
σ	surface tension [kg/s^2]
τ	shear stress [N/m^2]
Θ	angle [rad]

subscripts

b	bulk
bcy	buoyancy
d	drag
du	bubble growth
D	departure
fc	forced convection
$flow$	flow-induced
g	vapour phase
l	liquid pahse
L	lift-off
nb	nucleate boiling
onb	onset of boiling
p	constant pressure
s	saturation
sl	shear lift
sub	subcooling
$trans$	transition
w	wall

The heat transfer in the PDB regime is basically dominated by two effects, the macroconvection due to the motion of the bulk liquid, and the latent heat transport associated with the evaporation of the liquid microlayer between the bubble and the heater wall. The contribution of the macroconvection is very important in the PDB region. As it is schematically shown in Fig. 2, the individual paths of the boiling curves associated with different flow rates differ significantly in the PDB region. With increasing bulk velocity the onset of nucleate boiling is typically shifted

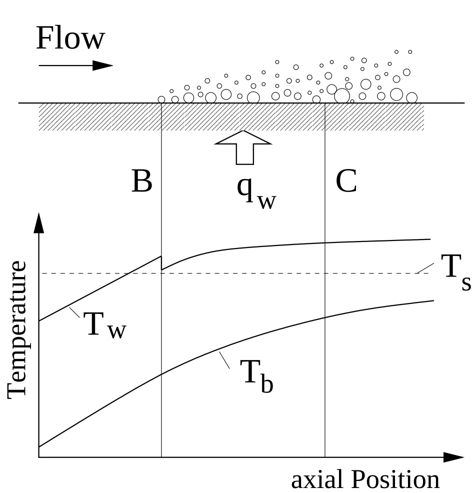

Figure 1: Subcooled flow boiling domain in a heated channel

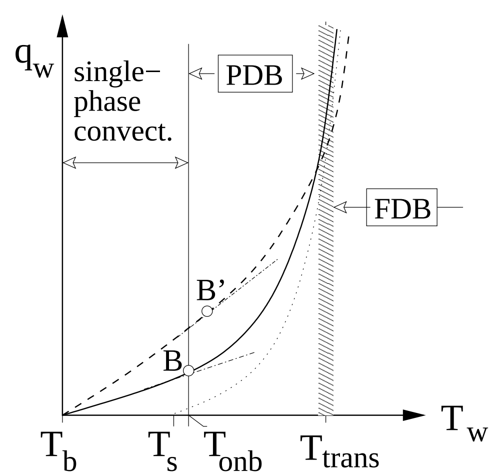

Figure 2: Flow boiling curves for two bulk velocities denoted by the full and the dashed line, respectively, where the dashes represent the higher velocity case; dotted line denotes the pool boiling curve.

to higher wall superheats, such that the nucleate boiling sets in at point B' instead of B in case of the higher flow rate. At higher wall superheats the influence of the macroconvection becomes less pronounced, and the evaporating effect prevails. Accordingly, the flow boiling curves approach the pool boiling curve. The transition to the fully developed boiling (FDB) regime can then be localized at the temperature T_{trans} beyond which the different flow boiling curves follow very closely almost parallel paths independently of their actual flow rate.

Many models suggested for subcooled flow boiling assume the total wall heat flux q_w to be superimposed of two additive contributions, which can be written as

$$q_w = q_{fc} + q_{nb}\,. \tag{1}$$

The first term q_{fc} is due to forced convection, the latter q_{nb} is due to nucleate boiling. This concept of additive contributions was first introduced by Rohsenow [4]. In this approach Rohsenow subtracted from the experimentally measured values for the total wall heat flux a convective single phase contribution and attributed the residual term to nucleate boiling.

Rather than assuming the additive composition (1) an alternative group of models suggests a geometrical combination of the basic contributions. Models of this type formulate the effective heat transfer coefficient as some some product function which can be generally written as

$$h_{tp} = h_{fc}\Psi_{tp}, \tag{2}$$

where the Ψ_{tp} represents a correction function due to nucleate boiling. Correlations of this type as suggested by Kandlikar[5], or, by Shah[6], mostly distinguish between the partially and the fully developed boiling regime. Accordingly, they propose different correlations for each regime, which in turn requires the prior determination of the point of transition from the PDB to the FDB regime, respectively.
Among the superposition models of type (1) the proposal due to Chen [9] is widely used today especially in engineering applications in the automotive industry. In his approach Chen advanced his predecessors' superposition models by accounting explicitely for the interaction between the liquid and the vapour phase. In particular, he distinguished two counterplaying effects on the outcome of total wall heat flux, the enhanced convective transport due to bubble agitation, and the flow-induced suppression of the nucleate boiling. The first effect was assumed to be negligible in the subcooled boiling regime. The latter flow-induced suppression effect is modeled by Chen in terms of a supression factor depending on the Reynold number of the bulk flow as the only correlation parameter. The Bubble Departure Lift-off (BDL) model, which is proposed in the present work, was devised to improve Chen's superposition approach, in that it attempts to model the flow-induced supression on a physically sounder base. Rather than using the bulk flow Reynolds number, which can be hardly defined in geometrically complex flows, the present approach models the flow-induced suppression using local flow quantities provided by the numerical solution of the single phase wall shear layer. The impact of the subcooling, which is very pronounced at small flow rates combined with low superheats, is accounted for as well through an additional parameter. The predictive capability of the BDL model is evaluated by comparing its predictions with experimental data. These data have been obtained in-house in boiling water flow experiments. A detailled description of the experimental apparatus used for these measurements as well as the mathematical formulation for BDL model are presented in the following two sections, respectively.

Experimental setup

The present experimental apparatus was specially designed to investigate subcooled convective boiling flow at moderate pressures. The test section being the key part of the forced convective loop is schematically shown in Fig. 3. The bulk velocity and the bulk temperature of the liquid feed as well as the operating pressure are controlled at the inlet of the section. In the present configuration the bulk velocity can be varied within the range of $0.05 \leq u_b \leq 3.0\,[\mathrm{m/s}]$. The absolute operating pressure can be set within the range $1.0 \leq p \leq 2.0\,[\mathrm{bar}]$.

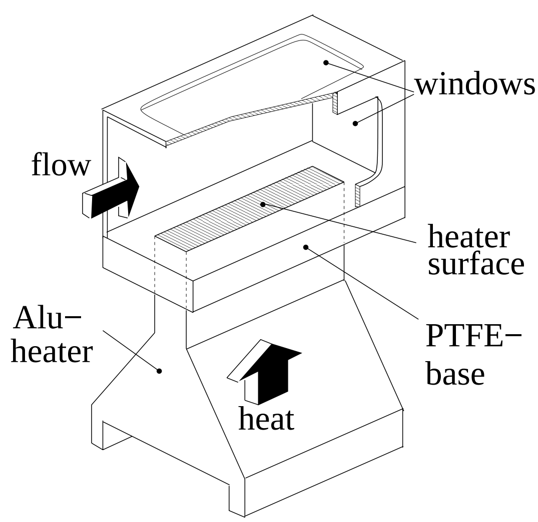

Figure 3: Test section of the experimental facility

The square cross section of the test channel has a height of 40 mm and a width of 30 mm. The heat flux into the channel is generated by heating coils placed at the bottom of the aluminium heater from where the heat is conducted to the top of the heater. At the upper surface (grey shaded in the sketch), whose length is 60 mm and width is 10 mm, respectively, the heat is transferred to the working fluid flowing through the channel. The wall temperature as well as the wall heat flux are determined based on measurements of the temperature using several K-type thermocouples appropriately distributed in the solid heater. The base plate of the test section, where the top of the aluminium heater is built in, is made of PTFE, whose outstandingly low thermal conductivity ($k_{PTFE} = 0.23\,[\mathrm{W/mK}]$) should guarantee the lowest possible heat loss of the heater to the surrounding structure. Ought to thermal durability restrictions of the PTFE base plate the maximum heater surface temperature was limited to $T_w = 160^o[\mathrm{C}]$. Windows made of glass are embedded in the top as well as the side walls of the test channel to make the heater surface optically accessible. Using the present configuration the total error in the experimentally obtained heat fluxes are mainly due to measurement and position errors of the thermocouples, as well as the uncertainties about the heat losses to the surroundings of the heater. A worst-case estimation turned out a total error for the heat flux ranging from $\pm 5\%$ in the convective regime to $\pm 2\%$ in the nucleate boiling regime referring to the value actually obtained from the measurements. The relative error in the measured surface temperatures amounts to $\pm 0.15^o[\mathrm{C}]$. The relative error of the measured bulk velocity is $\pm 0.5\%$.

Mathematical formulation of the subcooled boiling flow model

The BDL model invokes an additive ansatz as suggested by Chen[9] for the total wall heat flux

$$q_w = q_{fc}\,\Phi + q_{nb}\,S\,, \tag{3}$$

where the two correction parameters Φ and S modify the forced convection heat flux q_{fc} and the nucleate boiling heat flux q_{nb}, respectively. Both heat fluxes are computed following Chen's proposal. Accordingly, the first is written as

$$q_{fc} = h_{fc}\,(T_w - T_b)\ , \tag{4}$$

where the transfer coefficient h_{fc} is calculated using the Dittus-Boelter equation for the convective Nusselt number

$$Nu_{fc} = \frac{h_{fc} D_{hyd}}{k_l} = 0.023 Re_l^{0.8} Pr_l^{0.4} \tag{5}$$

involving the bulk flow Reynolds number and the Prandtl number of the liquid phase

$$Re_l = \frac{\rho_l\,u_b\,D_{hyd}}{\mu_l}, \quad Pr_l = \frac{\mu_l\,c_{p,l}}{k_l},$$

respectively. The nucleate boiling heat flux

$$q_{nb} = h_{nb}\,(T_w - T_s) \tag{6}$$

is obtained using a correlation due to Forster and Zuber [10]

$$h_{nb} = 0.00122\,\frac{k_l^{0.79} c_{p,l}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29}\, h_{lg}^{0.24} \rho_g^{0.24}} \Delta T_s^{0.25} \Delta p_s^{0.75}, \tag{7}$$

where the saturation pressure difference corresponding to the superheat temperature $\Delta T_s = T_w - T_s$ is written as

$$\Delta p_s = p_s(T_w) - p_s(T_s).$$

The factor Φ occurring in Eq.(3) represents the enhancement of the convective component due to bubble agitation. Since the vapour mass fractions in subcooled boiling flow are typically small, this effect was assumed by Chen[9] as negligibly small, such that Φ can be set to unity. The essential difference between Chen's approach and the present BDL concept consists in the modeling of the suppression factor S in Eq.(3). Chen correlated this flow-induced suppression factor as an empirical function of the bulk flow Reynolds number Re_l, which was later fitted by Butterworth[11] with the expression

$$S_{Chen} = \frac{1}{1 + 2.53 \cdot 10^{-6}\,Re_l^{1.17}}\,. \tag{8}$$

In contrast to Chen's bulk-flow dependent, hence, basically global, correlation (8) the BDL model attempts to model the flow-induced suppression based on the local dynamics of a vapour bubble subject to the surrounding flow field near the heater surface. Thereby, the BDL model utilizes a concept which was originally proposed by Zeng *et al.* [12] to compute the size of a bubble at the point of detachment from the heater surface. According to the hypothesis of Zeng and his coworkers the whole process of the bubble detachment basically evolves in three different stages, as schematically shown in Fig.4.

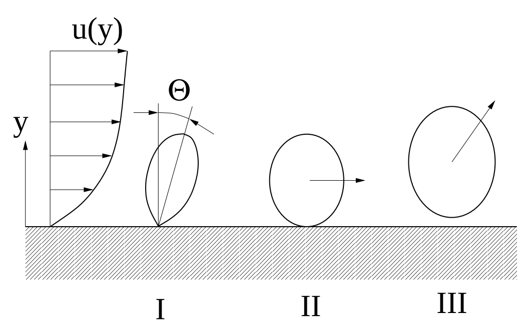

Figure 4: Three stages of a vapor bubble departing from the heater surface

At the first stage the bubble is attached to its nucleation site, and it is inclined by the angle Θ ought to the hydrodynamic flow forces. The attached bubble is growing until it reaches a critical departure volume V_D, where the bubble is dragged off its nucleation site.

After the departure stage II begins, and the bubble slides in upright posture ($\Theta = 0$) along the heater surface. It keeps growing in size until it reaches a bubble volume V_L, where the buoyancy force is sufficiently high to make the bubble lift-off from the surface, which marks the beginning of the stage III. The volume equivalent departure and lift-off radii are defined as

$$R_D = \left(\frac{3\,V_D}{4\,\pi}\right)^{\frac{1}{3}} \quad \text{and} \quad R_L = \left(\frac{3\,V_L}{4\,\pi}\right)^{\frac{1}{3}},$$

respectively. Fig.5 shows schematically all the relevant forces at the point of departure, being the drag force F_d, the shear-lift force F_{sl}, the buoyancy force F_{bcy}, and the bubble growth force F_{du}. Ought to the small density ratio $\frac{\rho_g}{\rho_l} \ll 1$ the inertial forces were neglected. The surface tension force was assumed to be negligibly small at the moment of departure and omitted as well. Solving for the balance equations of the considered forces in the $x-$ and $y-$direction

$$0 = F_d + F_{du}\, sin\Theta\,, \quad (9)$$

$$0 = F_{bcy} + F_{du}\, cos\Theta + F_{sl}, \quad (10)$$

yields the radius R_D. The lift-off radius R_L is obtained from Eqs.(9)-(10) assuming zero inclination angle, $\Theta = 0$, and zero slip between the sliding bubble and the liquid phase, which implies $F_d = F_{sl} = 0$. The drag force and the shear-lift force are given by

$$F_d = 6\pi\mu_l u R \left\{ \frac{2}{3} + \left[\left(\frac{12}{Re_{nb}} \right)^n + 0.796^n \right]^{-\frac{1}{n}} \right\}, \quad (11)$$

with $n = 0.65$ and

$$F_{sl} = \frac{3.877}{2}\rho_l u^2 \pi R^2 G_s^{\frac{1}{2}} \left(\frac{1}{Re_{nb}^2} + 0.014\, G_s^2 \right)^{\frac{1}{4}}, \quad (12)$$

respectively. Therein, the bubble Reynolds number as well as the shear rate,

$$Re_{nb} = \frac{\rho_l\, u\, 2\, R}{\mu_l} \quad \text{and} \quad G_s = \left| \frac{du}{dy} \right|_{y=R} \frac{R}{u}$$

are obtained using Reichardt's near wall velocity formula in wall coordinates

$$\frac{u}{u_\tau} = \frac{1}{\kappa}\, ln\left(1 + \kappa\, y^+\right) + K \left[1 - exp\left(-\frac{y^+}{\chi}\right) - \frac{y^+}{\chi}\, exp\left(-\frac{y^+}{3}\right) \right] \quad (13)$$

evaluated at the location $y = R$ with the constants set to $\kappa = 0.41$, $\chi = 11$, and $K = 7.4$, respectively. The buoyancy force is given by

$$F_{bcy} = \frac{4}{3} R^3\, \pi\, g\, (\rho_l - \rho_g)\,. \quad (14)$$

The bubble growth force F_{du} is modeled following Zeng *et al.*[13], who considered a hemispherical bubble expanding in an inviscid liquid. They proposed the equation

$$F_{du} = -\rho_l \pi R^2 \left(\frac{3}{2} C_s \dot{R}^2 + R\ddot{R} \right), \quad (15)$$

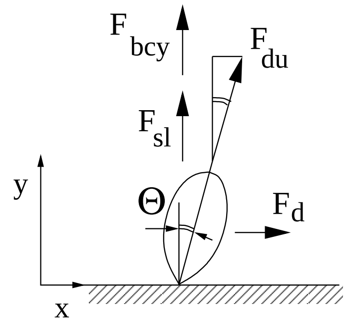

Figure 5: Force balance at a vapor bubble at the instant of departure from its nucleation site

where the empirical constant C_s was introduced to account primarily for the presence of the wall. Based on a comparison with experimental data available to them the authors suggested to set $C_s = \frac{20}{3}$. The temporal evolution of the bubble radius $R(t)$, as well as its temporal derivatives $\dot{R}$ and $\ddot{R}$ needed in Eq.(15), are obtained assuming a diffusion controlled bubble growth model due to Zuber[14], which reads

$$R(t) = \frac{2\,b}{\sqrt{\pi}}\, Ja\, \sqrt{\alpha_l\, t}\,, \quad (16)$$

involving the Jakob number

$$Ja = \frac{\rho_l c_{p,l}(T_w - T_s)}{\rho_g h_{lg}},$$

the thermal diffusivity of the liquid phase α_l and an empirical constant b of the order of unity, respectively.

The predicted bubble radii with the growth rate parameter occurring in Eq.(16) set to $b = 0.21$ were compared to experimental data measured at a given wall superheat $\Delta T_s = 29^o$ for four different velocities $u_b = 0.05, 0.39, 0.77, 1.17$ [m/s], respectively. The measurements were carried out with the experimental facility decribed above. Thereby, the size of the bubbles was optically measured using video records. The wall friction velocities u_τ needed by the model in Eq.(13) were obtained from LDA measurements of the axial velocity profiles. As shown in Fig. 6

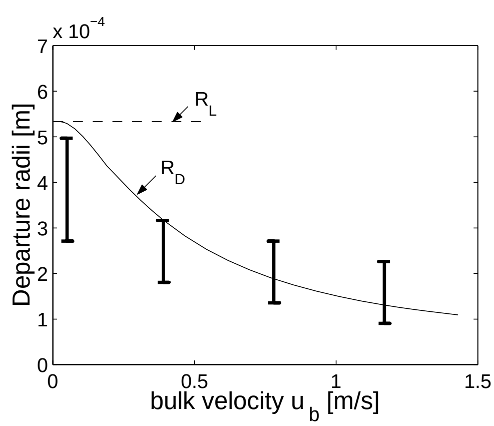

Figure 6: Predicted departure radii R_D in comparison with experimental data denoted by the error bars.

the agreement is good for the higher velocities, while deviations occur at the lowest velocity considered. Both radii become equal, i.e., $R_D = R_L$, in the limit of zero bulk velocity. With increasing flow rate the departure radius R_D decreases markedly relative to the lift-off radius R_L reflecting the influence of the local velocity field on the bubble detachment process from the surface. Based on this consideration the BDL model proposes the ratio

$$S_{flow} = \frac{R_D}{R_L}. \tag{17}$$

for the flow-induced suppression factor. According to its definition S_{flow} evidently represents the flow-induced deviation of the bubble departure radius from the corresponding pool boiling limit, which is associated with zero bulk velocity, where $R_D = R_L$ and, hence, $S_{flow} = 1$. However, since no subcooling is explicitely accounted for in the formulation so far, the obtained lift-off radius R_L represents the lift-off radius in the saturated boiling regime and not in the subcooled boiling regime as it is considered in the present configuration. This also explains the overprediction for the bubble sizes in the very low velocity range shown in Fig. 6. It is reasonable to assume that even at very small flow rates the advection of subcooled bulk liquid is sufficient to maintain a subcooled thermal boundary layer, where the departing bubbles are typically smaller than in the saturated case. Accordingly, the experimentally measured departure radii tend towards a limit notably below the predicted lift-off radius R_L representing the saturated case. The BDL model accounts for the effect of subcooling by introducing an additional suppression factor

$$S_{sub} = \frac{T_w - T_s}{T_w - T_b}. \tag{18}$$

Alike S_{flow} the factor S_{sub} is always less or equal unity. Representing a measure for the subcooling S_{sub} decreases for increasing $\Delta T_{sub} = T_s - T_b$. Substituting the two suppression factors defined in the Eqs.(17) and (18) as total suppression into Eq.(3) the total wall heat flux obtained with the BDL model reads

$$q_w = q_{fc} + q_{nb} S_{flow} S_{sub}. \tag{19}$$

Comparison of the model predictions with experiments

Using the experimental setup described above several flow boiling curves for a given absolute pressure und a given bulk velocity were measured. The thereby considered individual pressure-velocity combinations are summarized in table 1. In all considered cases the working fluid was water, whose bulk inlet temperature was kept constant at $T_b = 95^o$C, which implies a subcooling of $\Delta T_{sub} = 16^o$C in the case $p = 1.5$ bar and $\Delta T_{sub} = 25^o$C in the case $p = 2.0$ bar, respectively.

p [bar]	u_b[m/s]		
1.5	0.05	0.39	1.17
2.0	0.20	0.39	1.17

Table 1: Pressures (absolute) and velocities of the bulk liquid considered in the boiling flow experiments

The heater wall temperature was varied within the interval $95^o \leq T_w \leq 150^o$C. The slowest velocity in the lower pressure case ($p = 1.5$ bar) was chosen deliberately small, i.e., $u_b = 0.05$ [m/s], to come as closest to the pool boiling limit as it was possible in the present experimental facility. As already noted in the former section the individual wall friction velocities u_τ associated with each bulk velocity u_b and required as model input in Eq.(13) were obtained using LDA measurements of the axial velocity profiles. Figs. 7 and 8 show the comparison between the total wall heat fluxes predicted by the BDL model with the corresponding experimental data at the two considered levels of the operating pressure, respectively. In all diagrams the saturation temperature T_s is marked by a thin vertical line. In addition to the results of the present BDL approach the heat fluxes predicted by

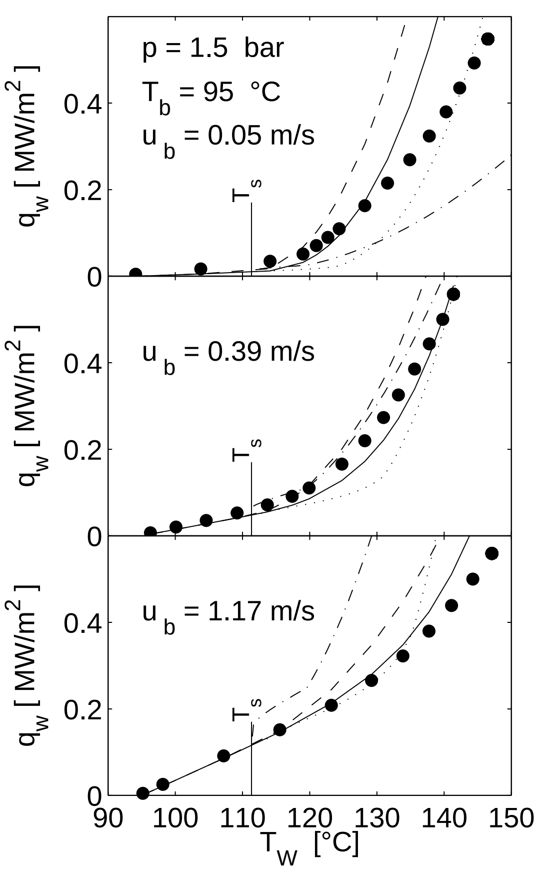

Figure 7: Predicted flow boiling curves at $p = 1.5$ [bar] and three different velocities of the bulk flow, respectively: full line,'——', BDL; dashed line,'– –', Chen; dashed dotted line ,'– · –', Shah; dotted line ,'· · ·', Kandlikar; symbols, '•', measurements

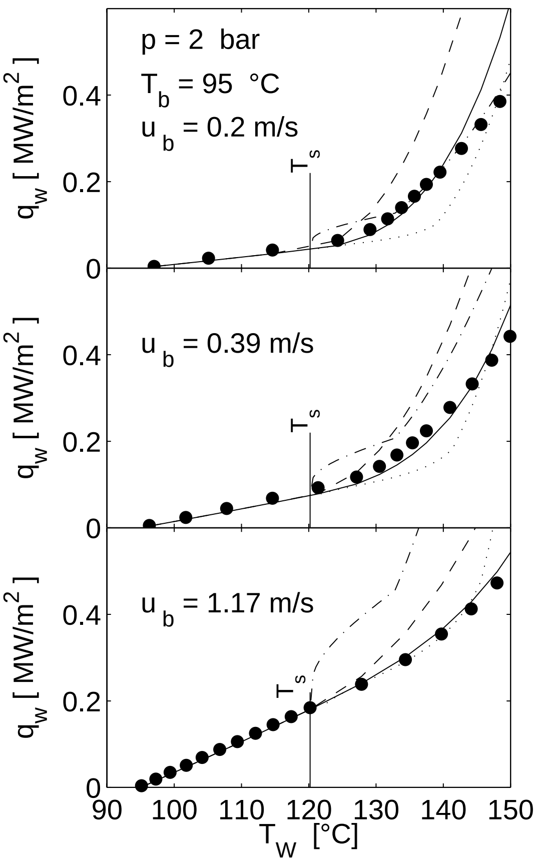

Figure 8: Predicted flow boiling curves at $p = 2.0$ [bar] and three different velocities of the bulk flow, respectively: full line,'——', BDL; dashed line,'− −', Chen; dashed dotted line ,'− · −', Shah; dotted line ,'· · ·', Kandlikar; symbols, '•', measurements

other widely used models, due to Chen[9], Shah[6] and Kandlikar[5], are depicted as well.

The overall agreement between the predictions of the BDL model and the experimental data is very good. Particularly, in the high velocity case ($u_b = 1.17$[m/s]) the model predicts the shift of the onset of nucleate boiling to higher wall superheats accompanied by a relativ reduction of the boiling component in the total heat flux very accurately. This indicates that the BDL model is capable to capture the strong flow induced supression of the nucleate boiling in the PDB regime very well. For higher wall superheats, however, approaching the FDB regime, the agreement becomes worse especially in case of the lowest velocity considered ($u_b = 0.05$[m/s]). This discrepancy can be explained by the fact that the present BDL model basically relies on the dynamics of a single bubble subject to a surrounding subcooled liquid flow field, which is assumed to remain unaffected by the presence of the vapour phase. Therefore, it is conceivable that the predictions of the model are less accurate once phenomena related to the very complex multi-bubble dynamics become important. Due to the high bubble number densities, which are typically found on the heater surface in the FDB regime, there is a strong bubble-bubble interaction as well as a notable two-way coupling between the motion of the bubbles and the liquid phase. In such a regime bubbles tend to coalesce forming larger structures on the surface. Moreover, the motion of the bubbles pronouncedly affects the surrounding flow field of the liquid phase and vice versa. Aiming at these highly complex multi-phase flow phenomena there is certainly scope for further development of the present model in order to improve its accuracy particularly in the FDB regime.

The comparison with the results obtained with other models also shown in Figs. 7-8 reveals that Chen's approach generally overpredicts the heat fluxes. It turns out that the empirical function for the suppression factor S_{Chen} given in Eq.(8) used in Chen's model is obviously calibrated for saturated boiling, where the nucleate boiling heat transfer is typically higher than in subcooled boiling. In addition, since the factor S_{Chen} depends on the bulk flow Reynolds number only, it practically represents a pure bulk flow quantity. It is therefore insensitive to local quantities like the wall superheat or wall shear stress, which can be exspected to have a significant effect on a local phenomenom like the nucleate boiling heat transfer in a wall shear layer. As it was already pointed out in the former section, it was essentially these two shortcomings which motivated the authors to develop Chen's ansatz further to the present BDL proposal.

The results obtained using Shah's model [6] show a very good agreement in the FDB region in the case, at $p = 2.0$[bar] and $u_b = 0.2$ [m/s]. In the PDB regime the agreement is generally rather poor especially for the higher velocities, where considerable overpredictions are observed. The pronounced kink in the predicted boiling curves is ought to the switching from the formulation applied to the PDB to that applied to the FDB regime. In the approach due to Kandlikar[5] the formulation changes in a smooth transition from the PDB to the FDB correlations. The model gives accurate predictions for the PDB regime in the high velocity case. However, at the lower velocities the region which is dominated by single-phase convection extends too far into the nucleate boiling region, as the considerable underpredictions in the PDB region make evident.

In order to evaluate the performance of the BDL model in the case of stronger subcooling its predictions were also compared to experimental data obtained by Bibeau and Salcudean[15], who carried out their flow boiling measurements at a pressure of $p = 2$ [bar] and a bulk velocity of $u_b = 0.08$ [m/s]. The temperature of the bulk flow was $T_b = 75^o$[C], which implies a subcooling of $\Delta T_{sub} = 45^o$[C]. As it is shown in Fig. 9, the results applying the BDL model agree very well with the experiments. Due to the low flow rate the flow-induced suppression accounted for through S_{flow} is small. Thus, the total suppression S is mainly due

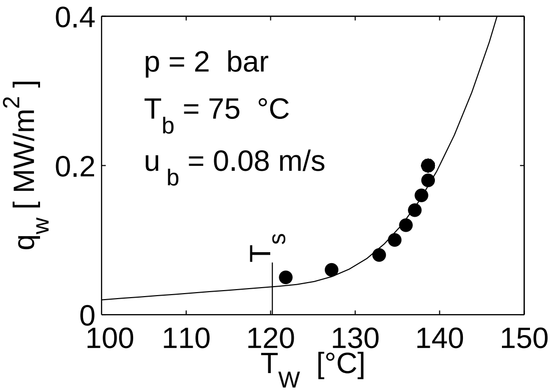

Figure 9: Comparison at stronger subcooling; full line,'——', BDL; symbols, '•', measurements by Bibeau and Salcudean[15]

to the effect of the subcooling. It turns out that the in the present case very dominant effect of subcooling is estimated very well in terms of the corresponding model parameter S_{sub} as given by Eq.(18).

Conclusion

In the present study a Chen-type superposition model is proposed to compute the effective wall heat flux in subcooled boiling flow. The proposed BDL model modifies the nucleate boiling contribution by introducing two suppression factors ought to the flow forces and to the subcooling of the thermal boundary layer, respectively. The comparison with exerimental data for water as working fluid shows good agreement especially in the partially developed boiling (PDB) regime. This good agreement in the PDB region, where the bulk flow rate exerts a significant effect on the nucleate boiling, indicates that the model captures the flow-induced suppression very well. Notable deviations occur primarily in the vicinity of the fully developed boiling (FDB) regime. These discrepancies clearly demonstrate the limits of the present model and give the scope for a further development. Thereby, the focus will have to be put on the consideration of the bubble-bubble as well as the bubble-liquid interactions, which are of great importance in the FDB regime.

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