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Analysis of pool boiling heat transfer: effect of bubbles sliding on the heating surface

G. Sateesh [a], Sarit K. Das [a], A.R. Balakrishnan [b,*]

[a] *Department of Mechanical Engineering, Indian Institute of Technology—Madras, Chennai 600 036, India*
[b] *Department of Chemical Engineering, Indian Institute of Technology—Madras, Chennai 600 036, India*

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Abstract

Pool boiling on surfaces where sliding bubble mechanism plays an important role has been studied. The heat transfer phenomenon for such cases has been analysed. The model considers different mechanisms such as latent heat transfer due to microlayer evaporation, transient conduction due to thermal boundary layer reformation, natural convection and heat transfer due to the sliding bubbles. Both microlayer evaporation and transient conduction take place during the sliding of bubbles, which occurs in geometries such as inclined surfaces and horizontal tubes. The model has been validated against experimental results from literature for water, refrigerant R134a and propane. The model was found to agree well for these fluids over a wide range of pressures. The model shows the importance of the contributions of the different mechanisms for different fluids, wall superheats and pressures.

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1. Introduction

Although investigated extensively for nearly half a century, pool boiling continues to be an area of active research interest even today. From the pioneering investigations by Jakob [1,2] and Rohsenow [3,4], research on pool boiling has undergone a metamorphosis from an empirical approach to physics based comprehension of the phenomenon. A recent review by Dhir [5] shows that even after much concentrated research efforts in this area, a satisfactory understanding of the physics of the boiling phenomenon is still elusive. The review shows that there is a lot of interest in simulating bubble hydrodynamics and heat transfer from a purely theoretical approach in recent years. This is also evident from the works of Dhir et al. [6,7], Genske and Stephan [8], and Stephan et al. [9]. However, it must be understood that numerical simulations alone cannot achieve an understanding of the mechanism of heat transfer, as the modelling itself is based on assumptions which come from earlier work, both theoretical and experimental. Thus, to successfully predict pool boiling heat transfer and to achieve an understanding of the different mechanisms contributing to it, one has to have a knowledge, a priori, of the different mechanisms as well as their relative importance under different operating conditions.

This task is best accomplished by modelling the boiling heat transfer phenomenon based on different observed mechanisms to predict, within acceptable limits, the heat flux for a wide range of operating parameters. Efforts at modelling using such an approach have been

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[*] Corresponding author. Tel.: +91 44 2257 8209; fax: +91 44 2257 0509.

E-mail address: arbala@iitm.ac.in (A.R. Balakrishnan).

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Nomenclature

A	area, m^2
Ar	Archimedes number, $\left(\frac{g}{\nu_f^2}\right)\left(\frac{\sigma}{\rho_f g}\right)^{3/2}$
B	constant (in Eq. (7))
c_p	specific heat, J/kg K
C_d	drag coefficient
d	diameter of bubble, m
D	diameter of the tube, m
f	frequency of bubble departure, s^{-1}
F	force, N
g	acceleration due to gravity, m/s^2
Gr_D	Grashof number based on tube diameter, $\frac{g\beta\Delta T D^3}{\nu^2}$
Gr_L	Grashof number based on length of the vertical surface, $\frac{g\beta\Delta T L^3}{\nu^2}$
Ja	Jacob number, $\frac{c_p\Delta T}{\lambda}$
k	conductivity, W/m K
K	ratio of area of influence to projected area of bubble at departure (=1.8)
l_s	sliding length, m
L	length of the vertical surface, m
M, N	functions of θ_m as defined in Eq. (5)
N	number of nucleation sites
n_b	nucleation site density, (N/A), $1/m^2$
n_s	number of sliding bubbles at any instant per unit area, m^{-2}
Nu	Nusselt number, h L/k
Pr	Prandtl number, $c_p\mu/k$
P_R	reduced pressure, (P/P_c)
P_c	critical pressure, N/m^2
q	heat flux, W/m^2
r	radius of bubble, m
R	parameter as defined in Eq. (28)
ΔT	wall superheat, K
t	time, s
U	velocity, m/s
x	parameter as defined in Eq. (30)
Y	parameter as defined in Eq. (31)

Greek symbols

α	thermal diffusivity, m^2/s
λ	latent heat of vaporisation, J/kg
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3
σ	surface tension, N/m
θ	contact angle, rad

Subscripts

a	advancing
B	buoyancy
b	bubble
d	bubble departure
D	drag
f	fluid
g	vapour
i	different mechanisms (me, tc, mes, tcs and nc)
l	lift-off
m	mean
me	microlayer evaporation (due to stationary bubble)
mes	microlayer evaporation due to sliding bubble
nc	natural convection
r	receding
S	surface tension
s	sliding
st	stationary
tc	transient conduction (due to stationary bubble)
tot	total
tcs	transient conduction due to sliding bubble
w	waiting

Superscript

m	modified

made for decades. Mikic and Rohsenow [10] modelled the phenomenon assuming transient conduction to be the major contributor to the heat flux, while Moore and Mesler [11] gave importance to microlayer evaporation. Judd and Hwang [12] later on added the contributions of natural convection to transient conduction and microlayer evaporation. These models, while bringing out the importance of the contributions of natural convection, transient conduction or microlayer evaporation separately, emphasised one or the other contribution. On the contrary Benjamin and Balakrishnan [13] combined microlayer evaporation during bubble growth, transient conduction during thermal boundary layer reformation and natural convection to predict the heat flux, which worked for a range of organic fluids and water for different surface roughness values at near atmospheric pressures.

It is important to note that the above models concentrated mainly on horizontal surfaces. In practice, surfaces are often non-horizontal or even horizontal surfaces such as horizontal tubes have got non-horizontal (curved) walls. Nishikawa et al. [14] studied the effect of surface inclination on the heat flux. Their experimental results show that the boiling heat transfer coefficient for downward facing inclined surfaces is higher than that for upward facing horizontal surfaces. This heat transfer coefficient increases with the increase of angle of inclination under low heat flux conditions or in partial nucleate boiling regime. However, the heat transfer coefficient is independent of surface inclination under high heat flux conditions or in fully developed nucleate boiling regime. This was attributed to the presence of sliding bubbles which enhance heat transfer by cyclic disruption and reformation of the thermal boundary layer. The heat transfer coefficient for surfaces with 90° angle of inclination is about two to three times higher than that for horizontal surfaces (0° angle of inclination) in the partial nucleate boiling regime. Cornwell [15] studied the effect of sliding bubbles on horizontal tubes. However, Cornwell and Schüller [16] confined their studies to just microlayer evaporation during sliding, neglecting the transient conduction due to sliding which can be as important as the former. Hence, any modelling of non-horizontal surfaces require the incorporation of all the mechanisms involved. This was the approach taken to some extent by Luke and Gorenflo [17] and Das and Roetzel [18]. While Luke and Gorenflo [17] took the sliding bubble mechanism into account, they considered transient conduction to be continuous and natural convection to be intermittent. The studies of Benjamin and Balakrishnan [13] show that the nature of these contributions are just the opposite. The work of Das and Roetzel [18] was based on a number of assumptions and was applied to only R134a, which gave good predictions in the low pressure region only. Another recent work by Yu and Cheng [19] uses a fractal model for a vertical surface and combines transient conduction with microlayer evaporation and natural convection. However, this model also does not consider the sliding of bubbles and has not been validated over a wide range of fluids and pressures.

Thus, from the above survey, it is seen that a heat transfer model for pool boiling on non-horizontal surfaces, which takes into account all the known mechanisms of heat transfer including the sliding of bubbles is still required. The aim of this study is to develop such a model which is capable of predicting the heat flux over a wide range of pressures and wall superheats and can be used with refrigerants, organic liquids and water with reasonable accuracy.

2. The model

Enhancement in heat transfer during pool boiling on non-horizontal surfaces can be attributed to both sliding bubbles and to stationary bubbles. There are primarily two mechanisms associated with the bubbles—firstly, the latent heat transfer due to microlayer evaporation at the bubble base during bubble growth and secondly, transient conduction as the disrupted thermal boundary layer reforms during the waiting period (i.e., before the incipience of the next bubble at the same nucleation site). These two mechanisms are intermittent and constitute one cycle. As the bubble departs from the nucleation site, it tends to slide for some distance before lift-off from the surface. The experimental results of Thorncroft et al. [20] indicate that there is bubble growth even during sliding, which means that there must be microlayer evaporation as the bubbles slide and transient conduction during the reformation of the thermal boundary layer disrupted by the sliding bubbles. Furthermore, there is natural convection in the area uninfluenced by the stationary and sliding bubbles.

Figs. 1 and 2 show the sliding bubble on a vertical surface and on a horizontal tube respectively. Fig. 3 shows the various forces acting on the bubble, namely buoyancy, drag and surface tension. Using the expressions for surface tensional force and buoyancy force as derived by Cornwell and Schüller [16], the expression

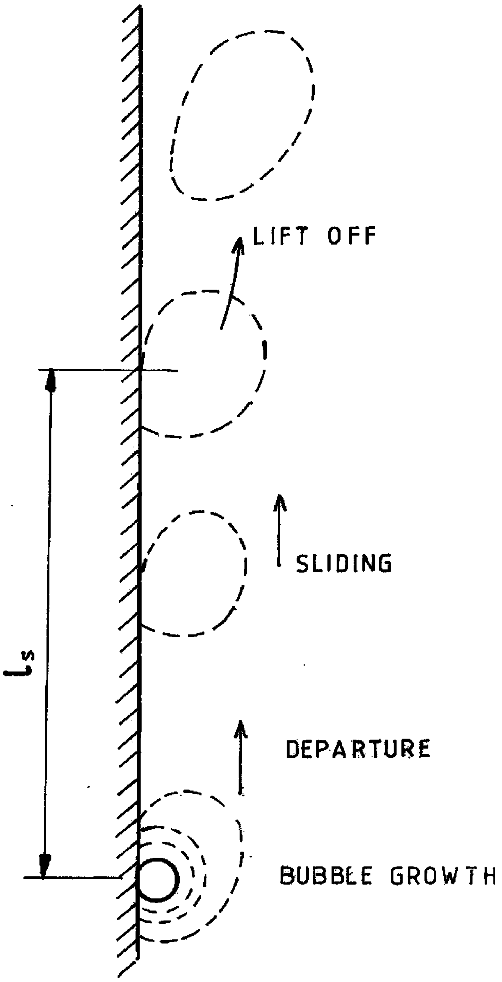

Fig. 1. Sliding of a bubble on a vertical surface.

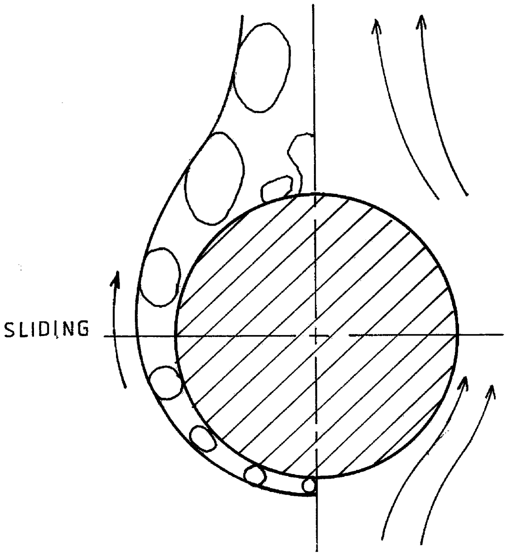

Fig. 2. Sliding of a bubble on a horizontal tube.

for bubble departure diameter on a vertical surface can be obtained. The surface tensional force is given by

$$F_s = \pi\sigma r \sin\theta_m(1-\cos\theta_m). \tag{1}$$

The buoyancy force is given by

$$F_B = \frac{\pi r^3}{3}(1+\cos\theta_m)^2(2-\cos\theta_m)(\rho_f-\rho_g)g, \tag{2}$$

where, r is the instantaneous radius of the bubble. The relation between the advancing liquid contact angle, θ_a, and the mean liquid contact angle, θ_m [16] is

$$\cos\theta_m = \frac{1+\cos\theta_a}{2}. \tag{3}$$

At bubble departure, the buoyancy force will be equal to surface tensional force

$$F_B = F_s. \tag{4}$$

From Eq. (4), the bubble departure diameter, d_d, can be obtained as

$$d_d = \sqrt{\frac{12N\sigma}{Mg(\rho_f-\rho_g)}}, \tag{5}$$

where

$$N = \frac{\sin\theta_m(1-\cos\theta_m)}{\pi-\theta_m+\sin\theta_m\cos\theta_m}$$

and

$$M = \frac{(1+\cos\theta_m)^2(2-\cos\theta_m)}{\pi-\theta_m+\sin\theta_m\cos\theta_m}.$$

The velocity of the sliding bubble on a vertical surface can be obtained by a pseudo-static force balance between drag force, buoyancy force and the surface tensional force [16] as

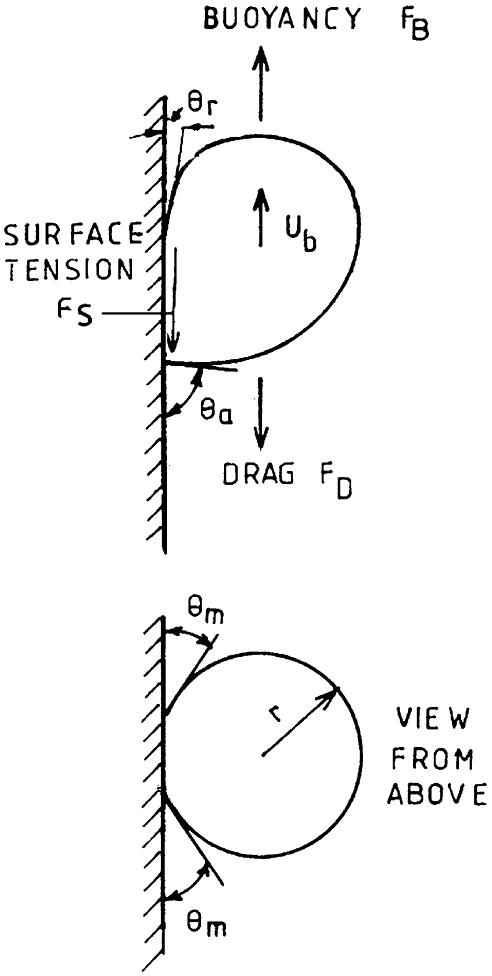

Fig. 3. Forces acting on a sliding bubble.

$$U_b = \sqrt{\frac{2\pi}{C_d}\left\{\frac{M(\rho_f-\rho_g)r}{3\rho_f} - \frac{N\sigma}{r\rho_f}\right\}}. \tag{6}$$

Photographic studies of Cornwell and Schüller [16] on R113 and also those of Williams and Mesler [21] in water indicate an angle θ_a of around 90°, implying θ_m is 60°. Further, Benardian et al. [22] reported a contact angle of 90° for water and that the contact angle was not pressure dependent. Hence, a value of 90° has been assumed for θ_a in the present analysis for all three fluids (water, R134a and propane) and at all pressures. The coefficient of drag, C_d for bubbles in the millimetre range in normal liquids as reported by Cornwell and Schüller [16] is 0.45 ± 0.08, and the same value is used in the present study. In the absence of any equation for bubble growth on vertical surfaces, the equation used by Benjamin and Balakrsishnan [13] on horizontal surfaces is used in the study

$$d(t) = BAr^{0.135}\left(\frac{k_{\mathrm{f}}Ja}{\rho_{\mathrm{f}}c_{\mathrm{pf}}}t\right)^{0.5}, \tag{7}$$

and the constant B is taken as 1.55. Ar and Ja are the Archimedes number and the Jacob number, defined as

$$Ar = \left(\frac{g}{v_{\mathrm{f}}^2}\right)\left(\frac{\sigma}{\rho_{\mathrm{f}}g}\right)^{\frac{3}{2}},$$

$$Ja = \left(\frac{\rho_{\mathrm{f}}c_{\mathrm{pf}}\Delta T}{\rho_v\lambda}\right).$$

Combining Eq. (7) with Eq. (5) gives the departure time, t_{d} as

$$t_{\mathrm{d}} = \frac{12N\sigma c_{\mathrm{pf}}}{MgJaB^2Ar^{0.27}k_{\mathrm{f}}}. \tag{8}$$

The departure time t_{d} was related to the waiting time, t_{w}, i.e., the time between the departure of one bubble and the incipience of the next bubble at the same nucleation site by van Stralen et al. [23] as

$$t_{\mathrm{w}} = 3t_{\mathrm{d}}, \tag{9}$$

and the bubble departure frequency

$$f = \frac{1}{t_{\mathrm{d}} + t_{\mathrm{w}}}. \tag{10}$$

2.1. Heat transfer to a stationary bubble

Heat transfer due to microlayer evaporation is assumed to be the latent heat required for generation of the vapour bubble (neglecting the condensation that may occur over the upper surface of the bubble) and can be written as

$$q_{\mathrm{me}} = \left(\frac{1}{6}\pi d_{\mathrm{d}}^3\rho_v\right)\lambda n_{\mathrm{b}}f. \tag{11}$$

The heat transfer due to transient heat conduction during the waiting period can be obtained as

$$\int_0^{t_{\mathrm{w}}}\frac{k_{\mathrm{f}}\Delta T}{\sqrt{\pi\alpha t}}\,\mathrm{d}t = 2\frac{k_{\mathrm{f}}\Delta T}{\sqrt{\pi\alpha t_{\mathrm{w}}}}t_{\mathrm{w}}. \tag{12}$$

Han and Griffith [24] postulated that the bubble on its departure takes with it liquid from an area, called the area of influence, that was four times the projected area of the bubble at departure. This must be the area on which transient heat conduction takes place during the waiting period. But Judd and Hwang [12] could match their predicted heat fluxes with experimental data only if the area of influence was 1.8 times the projected area of the bubble. In this study, a value of 1.8 is assumed and is denoted by 'K'. The heat flux due to transient conduction, q_{tc}, is therefore

$$q_{\mathrm{tc}} = 2\sqrt{\frac{k_{\mathrm{f}}\rho_{\mathrm{f}}c_{\mathrm{pf}}}{\pi t_{\mathrm{w}}}}\Delta T\left(n_{\mathrm{b}}K\frac{\pi d_{\mathrm{d}}^2}{4}\right)t_{\mathrm{w}}f. \tag{13}$$

2.2. Heat transfer to a sliding bubble

The vapour bubble, after departure, tends to slide along the surface for some distance before lift-off from the surface. Thorncroft et al. [20] reported frequent lift-offs with lift-off diameter about 1.6 times the departure diameter in pool boiling on vertical surfaces. However, sliding distances have not been reported. So as an approximation, the average sliding distance for pool boiling on vertical surfaces is assumed to be equal to half the distance between two nearest nucleation sites in an area under consideration. This can be written quantitatively as $0.5\sqrt{A/N}$. Assuming the stationary bubble growth equation of Benjamin and Balakrishnan [13] (Eq. (7)) to be valid for sliding bubbles and using Eq. (6) for bubble velocity, the time for lift-off (t_{l}) and hence lift-off diameter (d_{l}) can be estimated as

$$\int_{t_{\mathrm{d}}}^{t_{\mathrm{l}}}U_{\mathrm{b}}(t)\,\mathrm{d}t = 0.5\sqrt{\frac{A}{N}}. \tag{14}$$

Numerical integration can be performed to determine the limit t_{l} which satisfies the above equation. The maximum lift-off diameter calculated as above comes out to be 1.6 times the departure diameter, which is in agreement with the experimental results of Thorncroft et al. [20]. If the sliding distance were assumed to be $\sqrt{A/N}$, then the ratio (maximum) of lift-off diameter to departure diameter would be 2.0. However, for horizontal tubes, the sliding distance is assumed to be $\sqrt{A/N}$. It may be noted from Eq. (6), that the bubble velocity is a function of bubble radius, which in turn is a function of time. Eqs. (5) and (6) are valid even for horizontal tubes as the overall accuracy does not warrant consideration of the velocity distribution or the tube curvature [16], and therefore for horizontal tubes

$$\int_{t_{\mathrm{d}}}^{t_{\mathrm{l}}}U_{\mathrm{b}}(t)\,\mathrm{d}t = \sqrt{\frac{A}{N}}. \tag{15}$$

An expression for the number of sliding bubbles per unit area at any instant of time, n_{s}, can be obtained as

$$n_{\mathrm{s}} = n_{\mathrm{b}}ft_{\mathrm{s}}, \tag{16}$$

where t_{s} is average time of sliding of the bubble and is given by

$$t_{\mathrm{s}} = t_{\mathrm{l}} - t_{\mathrm{d}}. \tag{17}$$

The increase in the volume of the vapour bubble during sliding is due to microlayer evaporation. If this is assumed to be equal to the latent heat that is transferred, then the heat transfer due to microlayer evaporation, q_{mes}, can be written as

$$q_{\mathrm{mes}} = \frac{1}{6}\pi(d_1^3 - d_{\mathrm{d}}^3)\rho_v\lambda n_{\mathrm{b}} f. \tag{18}$$

During sliding, the area traversed by the bubble in an infinitesimally small time, dt is $d(t)U_{\mathrm{b}}(t)\,\mathrm{d}t$. Therefore, the area over which transient heat conduction takes place will be equal to $Kd(t)U_{\mathrm{b}}(t)\,\mathrm{d}t$. Integrating this elemental area from departure time, t_{d} to lift-off time, t_1, the heat transfer due to transient conduction during sliding, q_{tcs}, is given by

$$q_{\mathrm{tcs}} = 2\sqrt{\frac{k_{\mathrm{f}}\rho_{\mathrm{f}}c_{\mathrm{pf}}}{\pi t_{\mathrm{w}}}}\Delta T n_{\mathrm{b}} t_{\mathrm{w}} f \int_{t_{\mathrm{d}}}^{t_1} K d(t) U_{\mathrm{b}}(t)\,\mathrm{d}t. \tag{19}$$

It may be noted here that the time required for the thermal boundary layer reformation is assumed to be equal to the waiting time.

2.3. Natural convection

Natural convection has to be considered in the area that is not influenced by the bubbles (either stationary or sliding). At any instant of time, three fourth of the nucleation sites will be undergoing transient conduction and the remaining fourth will be in the growth state. This is because the waiting period is three times the growth period. The area occupied by these bubbles per unit heater surface can be written as

$$A_1 = \frac{3}{4}n_{\mathrm{b}}\left(K\frac{\pi d_{\mathrm{d}}^2}{4}\right) + \frac{1}{4}n_{\mathrm{b}}\left(\frac{\pi d_{\mathrm{d}}^2}{4}\right). \tag{20}$$

At any instant of time, the area occupied by the sliding bubbles per unit heater surface area can be written as

$$A_2 = n_{\mathrm{s}}\left(\frac{\pi\left(\frac{d_{\mathrm{d}}+d_1}{2}\right)^2}{4}\right). \tag{21}$$

The area over which transient heat conduction during sliding takes place per unit heater surface area can be written as

$$A_3 = n_{\mathrm{b}} l_{\mathrm{s}} K\left(\frac{d_{\mathrm{d}}+d_1}{2}\right), \tag{22}$$

where, l_{s} is the sliding length. Therefore, at a given instant of time, the area over which transient heat conduction takes place per unit heater surface area is

$$A_4 = \frac{3}{4}(A_3 - A_2). \tag{23}$$

Therefore, the fraction of the area over which natural convection takes place is

$$A_{\mathrm{nc}} = 1 - A_1 - A_2 - A_4. \tag{24}$$

Using the correlation due to Churchill and Chu [25] for natural convection on horizontal tubes

$$Nu = \left[0.6 + 0.387(Gr_D Pr C)^{1/6}\right]^2, \tag{25}$$

where,

$$C = \left[1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right]^{-16/9}$$

and

$$h_{\mathrm{nc}} = \frac{Nu\,k_{\mathrm{f}}}{D}.$$

In the case of vertical surfaces, another correlation due to Churchill and Chu [25] can be used:

$$Nu = \left[0.825 + \frac{0.387(Gr_L Pr)^{0.167}}{C}\right]^2, \tag{26}$$

where,

$$C = \left[1 + \left(\frac{0.437}{Pr}\right)^{0.5625}\right]^{0.296}$$

and

$$h_{\mathrm{nc}} = \frac{Nu\,k_{\mathrm{f}}}{L}.$$

The heat transfer due to natural convection then is

$$q_{\mathrm{nc}} = h_{\mathrm{nc}} A_{\mathrm{nc}} \Delta T. \tag{27}$$

2.4. Total heat flux

Eqs. (11) and (13) are valid when there is no bubble interaction. That is, heat flux is linearly dependent on nucleation site density as long as there is no bubble interaction. With the increase of nucleation site density (N/A), the total heat flux increases and the area available per nucleation site (A/N) decreases. Bubble interaction is assumed to take place if (A/N) is less than the area of the bubble at departure. Literature suggests that the heat flux and bubble departure diameter decreases and bubble departure frequency increases because of bubble interaction. However, a correlation or expression relating the nucleation site density, bubble departure diameter and heat flux in bubble interaction regime is not available to the best of the authors' knowledge. So, in order to quantify this effect, it is assumed that the bubble departure diameter remains unaffected by the bubble interaction, but the nucleation sites under the bubble get suppressed temporarily. Therefore, an area ratio parameter 'R' is defined as the ratio of area available per nucleation site (A/N) to the projected area of the bubble at departure (A_{d}), assuming no interaction:

$$R = \frac{A/N}{A_{\mathrm{d}}}. \tag{28}$$

Hence, at any instant of time, the number of active nucleation sites will be equal to Rn_{b},

$$Rn_b = \frac{(A/N)}{A_d}\left(\frac{N}{A}\right) = \frac{1}{A_d}.$$

This means that in the bubble interaction regime or high heat flux regime, the heat flux becomes independent of nucleation site density, and a continuous but discrete layer of bubbles may be visualized. But, as will be seen later, for R134a and propane, a correction in terms of reduced pressure is required.

2.5. Heat transfer due to sliding bubble

Sliding bubbles are assumed to be present as long as there is no bubble interaction. It is assumed to be absent once bubble interaction starts, i.e., when $R \leqslant 1$.

2.6. Heat transfer due to natural convection

The area available for natural convection decreases with the increase of nucleation site density. When the area available for natural convection (Eq. (24)) becomes less than or equal to zero, then natural convection is absent.

2.7. Total heat flux

The total heat flux can now be written as,

$$q_{tot} = (q_{me} + q_{tc})x_{st} + (q_{mes} + q_{tcs})x_s + q_{nc}, \qquad (29)$$

where, when $R > 1$

$$\begin{aligned} x_{st} &= 1, \\ x_s &= 1; \end{aligned} \qquad (30)$$

when $R \leqslant 1$

$$\begin{aligned} x_{st} &= R \quad \text{(for water at 1 bar)} \\ &= \left(\frac{R}{Y}\right)^{\frac{1}{Y}} \quad \text{(for R134a and propane at different} \\ &\qquad \text{reduced pressures),} \\ x_s &= 0 \quad \text{where,} \;\; Y = 3.7(P_R)^{0.19}. \end{aligned} \qquad (31)$$

While Eq. (30) can be obtained by simple reasoning, Eq. (31) was obtained when the model was compared with literature data and the values of x_{st} and x_s were adjusted. The total heat flux can now be written in terms of modified heat fluxes,

$$q_{tot} = (q^{m}_{mc} + q^{m}_{tc}) + (q^{m}_{mes} + q^{m}_{tcs}) + q_{nc}, \qquad (32)$$

where the superscript 'm' refers to modified or corrected values. Hereafter, the modified heat fluxes are referred to as just heat fluxes.

3. Results and discussion

The model has been compared with experimental heat flux data of Wang and Dhir [26] obtained with water on vertical surface, Barthau and Hahne [27] obtained with R134a (a refrigerant) on a horizontal tube of 15 mm diameter, and Luke and Gorenflo [17] obtained with propane (an organic fluid) on a horizontal tube of 8 mm diameter. This comparison was greatly facilitated by the fact that the experimental works of these authors also contain the nucleation site density data, which have been used in the present heat transfer model.

The predicted heat fluxes for water closely match the experimental data (Fig. 4) within ±25% for 80% of the data. Fig. 5 shows the fractional contributions of different mechanisms for water on vertical surface at 1 bar. At low superheats, the heat transfer due to sliding bubbles mechanism is quite considerable (about 50% at 5 K superheat). Transient conduction due to cyclic disruption and reformation of boundary layer as the bubbles slide along the surface is mainly responsible for the heat transfer enhancement. At relatively lower superheats, bubble interaction will be absent and bubbles tend to slide longer distances (which is assumed to be equal to $0.5\sqrt{A/N}$). Latent heat transfer due to microlayer evaporation also contributes to the heat transfer. The sliding bubble contribution first increases as the number of sliding bubbles increases and then it decreases as the sliding length decreases. Because of the relatively lower nucleation site density at lower superheats, the area available for natural convection will be high and hence its contribution is substantial (40% at 4 K superheat). The fractional contribution due to stationary bubble microlayer evaporation and transient conduction put together is lower (about 30% at 5 K superheat) due to lower nucleation site density.

Because of the bubble interaction and decrease in sliding distances due to increase in nucleation site

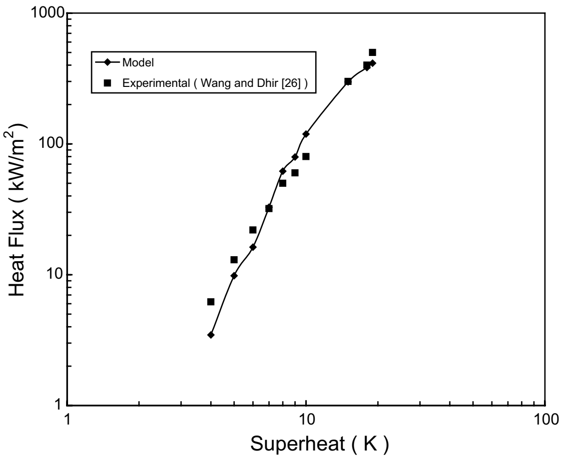

Fig. 4. Heat flux at various superheats for water boiling on a vertical wall at 1 bar.

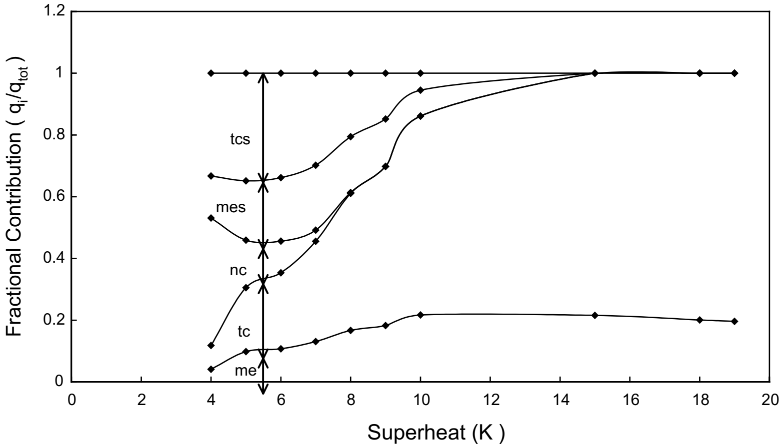

Fig. 5. Fractional contributions of different mechanisms at various superheats for water boiling on a vertical surface at 1 bar (data of Wang and Dhir [26]).

density with the increase of superheat, the contribution of sliding bubbles to the heat transfer decreases and becomes almost zero at about 14 K superheat in bubble interaction regime or fully developed nucleate boiling regime. These results are in conformity with the experimental results of Nishikawa et al. [14] for boiling on surfaces with different angles of inclination. These results show that sliding bubbles will be present at lower heat fluxes and cease to exist at higher heat fluxes because of bubble interaction.

At higher superheats (greater than 14 K), transient conduction due to stationary bubble is the dominant mechanism (about 75% contribution) for water and it increases with superheat. This result is in line with the confirmed general observation that transient conduction is the major mechanism in boiling of water at low pressures. However, latent heat transfer due to microlayer evaporation of a stationary bubble will also be considerable (about 25%). Natural convection ceases to exist at higher superheat values as there will be hardly any area available for natural convection to take place.

Fig. 6 shows the predicted heat fluxes versus superheat for R134a. The predicted heat fluxes are in good agreement with the experimental results. Fig. 7 shows the contributions of various heat transfer mechanisms for R134a on a horizontal tube of 15 mm diameter with a surface roughness of 0.4 μm at a reduced pressure of 0.15. It is evident from the figure that the heat transfer due to sliding of bubbles is about 20% at lower superheats and is absent at higher superheats. This is in accordance with the experimental observation of Cornwell [15]. Natural convection contribution, which is about 10% at lower superheats, gradually decreases and finally becomes zero. Heat transfer contribution due to stationary bubble microlayer evaporation and transient conduction increases with superheat. The former reaches a

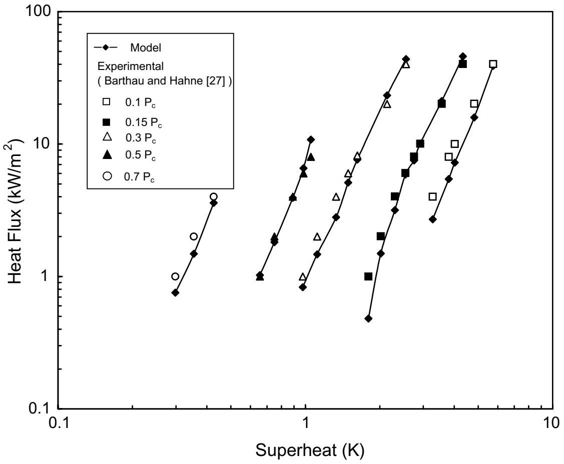

Fig. 6. Heat flux at various superheats for R134a on a horizontal tube.

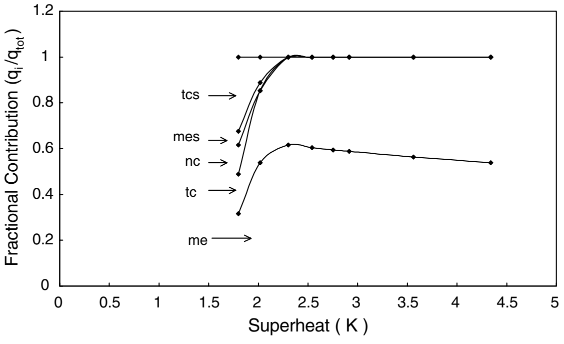

Fig. 7. Fractional contributions of different mechanisms at various superheats for R134a boiling on a horizontal tube at $0.15P_c$ (data of Barthau and Hahne [27]).

maximum of 60% and slightly decreases while the latter continues to increase. In this case, the contributions from microlayer evaporation and transient conduction are more or less equal. Fig. 8 shows the contributions for a reduced pressure of 0.5. About 80% heat transfer is due to microlayer evaporation or latent heat transfer and the rest due to transient conduction. Here, sliding bubbles and natural convection cease to exist. This is because the nucleation site density is so high that bubble interaction is very high. It must be noted that since the ratio of the nucleation site density to the superheat increases with pressure, natural convection and sliding bubbles are absent even for low superheats at higher pressures.

Fig. 9 shows the model prediction for propane on a horizontal tube of 8 mm diameter with a surface roughness of 0.34 μm at different reduced pressures. Here also, the model agrees well with experimental data. Figs. 10 and 11 show the fractional contributions of the different mechanisms at different pressures. The trend observed here is the same as that for R134a. Hence, it can be inferred from Figs. 7–11 that latent heat transfer or microlayer evaporation contribution (due to stationary bubble) increases and transient conduction contribution decreases with increase in pressure. For water at atmospheric pressure, transient conduction is the major mechanism (about 75%) with appreciable latent heat transfer or microlayer evaporation (about 25%). But for R134a and propane, microlayer evaporation contribution is as much as transient conduction contribution at lower reduced pressures and the former increases with increase in pressure. In general, sliding bubble contribution is found to be higher for vertical surface (at 1 bar) as compared to horizontal tubes (at higher pressures).

Fig. 12 shows a parity plot for the performance of the model with the three sets of experimental data obtained from the literature. The agreement between the model and the data is seen to be within ±25%.

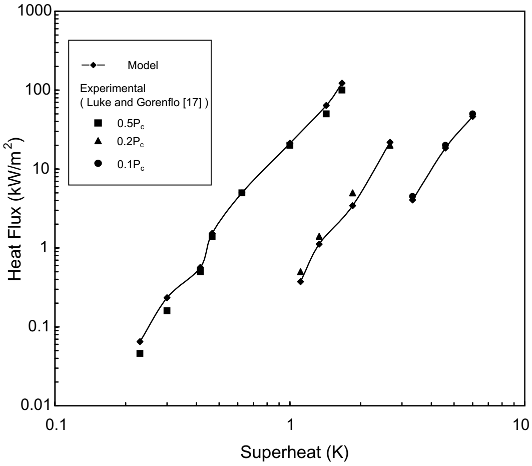

Fig. 9. Heat flux at various superheats for propane boiling on a horizontal tube.

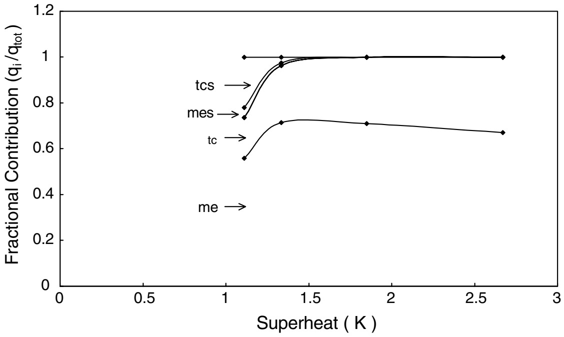

Fig. 10. Fractional contributions of different mechanisms at various superheats for propane boiling on a horizontal tube at $0.2P_c$ (data of Luke and Gorenflo [17]).

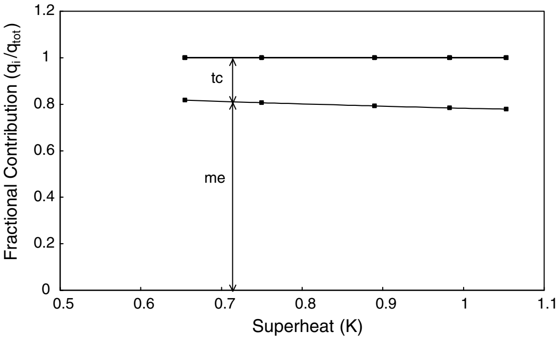

Fig. 8. Fractional contributions of different mechanisms at various superheats for R134a boiling on a horizontal tube at $0.5P_c$ (data of Barthau and Hahne [27]).

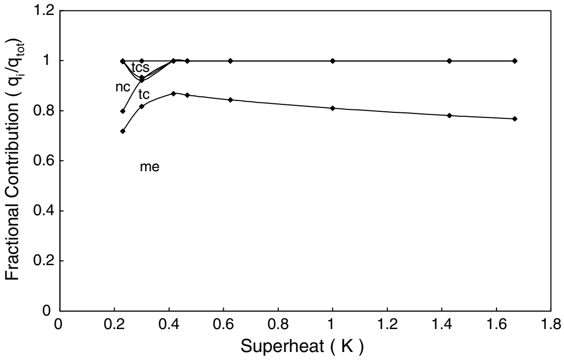

Fig. 11. Fractional contributions of different mechanisms at various superheats for propane boiling on a horizontal tube at $0.5P_c$ (data of Luke and Gorenflo [17]).

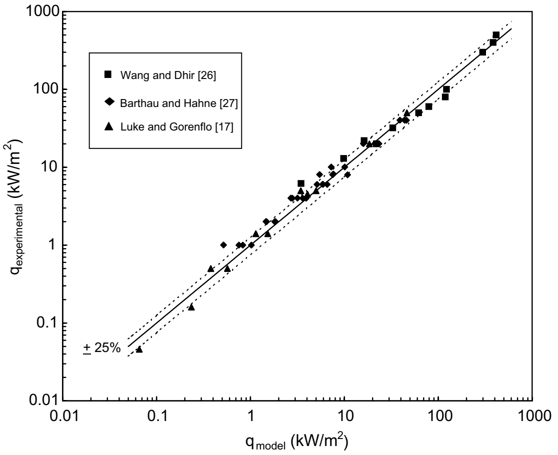

Fig. 12. Performance of the model with experimental data from the literature (parity plot).

4. Conclusion

An analysis of pool boiling heat transfer on non-horizontal surfaces has been presented. The model incorporates the well-known contributions of microlayer evaporation, transient conduction and natural convection with special emphasis on the mechanism of sliding bubbles. It is argued that during sliding of bubbles along vertical surfaces or the curved surface of a horizontal tube, the mechanism of latent heat transfer or microlayer evaporation and transient conduction due to cyclic disruption and reformation of thermal boundary layer take place simultaneously. An effort has been made to build up the model from the observed physical features rather than resort to empiricism. The only adjustable parameter is the correction for bubble interaction.

The model has been validated against the available experimental data for water, R134a and propane at different pressures and geometries. The analysis indicates a high fraction of the heat transfer being due to transient conduction and a significant contribution of the sliding bubbles for water boiling on a vertical wall at atmospheric pressure. For organic liquids, the contribution of microlayer evaporation becomes comparable with transient conduction at lower pressures and its contribution dominates at higher pressures.

References

[1] M. Jakob, Heat Transfer, vol. 1, John Wiley, New York, 1949 (Chapter 29).

[2] M. Jakob, W. Linke, Heat transfer from a horizontal plate, Forsch. Geb. Ingenieurwes 4 (1933) 434.

[3] W.M. Rohsenow, A method of correlating heat transfer data for surface boiling of liquids, Trans. ASME 74 (1952) 969–976.

[4] W.M. Rohsenow, Heat transfer with evaporation, in: Heat Transfer, University of Michigan, Summer 1952, pp. 101–150.

[5] V.K. Dhir, Boiling heat transfer, Annu. Rev. Fluid Mech. 30 (1998) 365–401.

[6] V.K. Dhir, On the use of numerical simulations to augment our understanding of boiling heat transfer, in: Proceeding of National Heat Transfer Conference, Pittsburgh, 2000, pp. 1–21.

[7] G. Son, N. Ramanujapu, V.K. Dhir, Numerical simulation of bubble merger process on a single nucleation site density during pool nucleate boiling, J. Heat Transfer 124 (2002) 51–62.

[8] P. Genske, K. Stephan, Nucleate boiling-numerical simulation of heat transfer at moderate heat fluxes, in: Conference of the International Institute of Refrigeration Commission, October 3–5, 2001, Paderborn, Germany.

[9] P. Stephan, T. Gambaryan, Roisman, C. Hohmann, C. Brandt, J. Kern, Modelling and verification of thin film evaporative heat transfer, in: Conference of the International Institute of Refrigeration Commission, October 3–5, 2001, Paderborn, Germany.

[10] B.B. Mikic, W.M. Rohsenow, A new correlation of pool boiling data including the effect of heating surface characteristics, J. Heat Transfer 91 (1969) 245–250.

[11] F.D. Moore, R.B. Mesler, The measurement of rapid surface temperature fluctuations during nucleate boiling of water, AIChE J. 7 (1961) 620–624.

[12] R.L. Judd, K.S. Hwang, A comprehensive model for nucleate boiling heat transfer including microlayer evaporation, J. Heat Transfer 98 (1976) 623–629.

[13] R.J. Benjamin, A.R. Balakrishnan, Nucleate pool boiling heat transfer of pure liquids at low to moderate heat fluxes, Int. J. Heat Mass Transfer 39 (1996) 2495–2504.

[14] K. Nishikawa, Y. Fujitha, H. Ohta, Effect of surface configuration on nucleate boiling heat transfer, Int. J. Heat Mass Transfer 27 (1974) 1559–1571.

[15] K. Cornwell, The role of sliding bubbles in boiling on tube bundles, in: Proceeding of International Heat Transfer Conference, Jerusalem, 1990, pp. 455–460.

[16] K. Cornwell, R.B. Schüller, A study of boiling outside a tube bundle using high speed photography, Int. J. Heat Mass Transfer 25 (1982) 683–690.

[17] A. Luke, D. Gorenflo, Heat transfer and size distribution of active nucleation sites in boiling propane outside a tube, Int. J. Therm. Sci. 39 (2000) 919–930.

[18] S.K. Das, W. Roetzel, Heat transfer model for pool boiling on a horizontal tube, in: Proceedings of the International Heat Transfer Conference, Grenoble, 2002, pp. 671–676.

[19] B. Yu, P. Cheng, A fractal model for nucleate pool boiling heat transfer, J. Heat Transfer 124 (2002) 1117–1124.

[20] G.E. Thorncroft, J.F. Klausner, R. Mei, An experimental investigation of bubble growth and detachment in vertical upflow and downflow boiling, Int. J. Heat Mass Transfer 41 (1998) 3857–3871.

[21] D.D. Williams, R.B. Mesler, The effect of surface orientation on delay time of bubbles from artificial sites during nucleate boiling, AIChE J. 13 (1967) 1020–1024.

[22] J.D. Benardian, I. Mudawar, C.B. Walsh, E.I. Franses, Contact angle temperature dependence for water droplets

on practical aluminum surfaces, Int. J. Heat Mass Transfer 40 (1997) 1017–1033.

[23] S.J.D. Van Stralen, M.S. Sohal, R. Cole, W.M. Sluyter, Bubble growth rates in pure and binary systems: combined effect of evaporation and relaxation microlayer, Int. J. Heat Mass Transfer 18 (1975) 453–467.

[24] C.Y. Han, P. Griffith, The mechanism of heat transfer in nucleate pool boiling, bubble initiation, growth and departure, Int. J. Heat Mass Transfer 8 (1965) 887–904.

[25] W. Churchill, H.H.S. Chu, Correlating equations for laminar and turbulent free convection from a horizontal cylinder, Int. J. Heat Mass Transfer 18 (1975) 1049–1053.

[26] C.H. Wang, V.K. Dhir, Effect of surface wettability on active nucleation site density pool boiling of water on a vertical surface, J. Heat Transfer 115 (1993) 659–669.

[27] G. Barthau, E. Hahne, Nucleation site density and heat transfer in nucleate pool boiling of refrigerant R134a in a wide pressure range, in: Proceedings of the 3rd European Thermal Sciences Conference, vol. 2, 2000, pp. 731–736.