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A transient micro-convection model of nucleate pool boiling

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Abstract—This work advances the fundamental understanding of nucleate pool boiling on plain surfaces. A model is developed for prediction of the nucleate boiling coefficient as a function of the bubble dynamics characteristics (nucleation site density, bubble departure diameter and bubble frequency). This work extends that of Mikic and Rohsenow, who developed a model based on transient conduction to the superheated liquid layer. The present model includes the effects of transient convection to the liquid, as a result of convection in the wake of the departing bubbles. An asymptotic correlation is used to patch the transient conduction and the steady-state convection asymptotes to cover the full bubble cycle. The predicted heat transfer coefficient is compared using data for R-11 and R-123, and good agreement is found. The model shows that the contribution of transient conduction is small compared to transient micro-convection. It is believed that this is the first successful demonstration of a mechanistically based model for nucleate boiling on plain surfaces. © 1997 Elsevier Science Ltd.

INTRODUCTION

Several studies have focused on the mechanism of nucleate pool boiling on plain surfaces. The high heat flux in nucleate boiling is attributed to one or more of the following possible mechanisms:

(1) Transient conduction to, and subsequent replacement of, the superheated liquid layer in contact with the heating surface, as proposed by Han and Griffith [1]. This mechanism will be critically assessed later.
(2) Evaporation of a thin liquid microlayer beneath the growing bubble. This was first suggested by Moore and Mesler [2] based on their observation of rapid surface-temperature fluctuations in nucleate boiling. This is discussed in the following section.
(3) Circulation of liquid in the vicinity of a growing bubble due to thermocapillarity effects at the vapor–liquid bubble interface, as proposed by Brown [3]. There is no experimental evidence that this effect is significant. Tong *et al.* [4] theoretically showed that the contribution of Marangoni flow to the total heat flux can be ignored for all except very high subcooling cases.

Microlayer evaporation and latent heat transport

The existence of a thin evaporating microlayer beneath a growing bubble was experimentally confirmed by Hendricks and Sharp [5], and Cooper and Lloyd [6]. However, they established only the existence of the phenomenon, not its relative contribution to the nucleate boiling heat transfer. The following literature review on plain surfaces questions the significance of microlayer evaporation to the overall heat transport.

Judd and Hwang [7] conducted an experimental investigation of dichloromethane boiling on a smooth glass surface using laser interferometry and high-speed photography. Their data show that the microlayer evaporation accounts for 8–15% of 25–40 kW/m^2 total heat flux. Even at the greatest level of heat flux investigated (60 kW/m^2), the latent contribution was 30% of the total heat flux. In an experimental investigation of subcooled flow nucleate boiling of water on stainless steel at atmospheric pressure, Del Valle and Kenning [8] found that the latent contribution to the total heat flux from microlayer evaporation was only 2–3%. Paul *et al.* [9] took bubble dynamics and heat flux data for saturated nucleate boiling of water on an electrically heated platinum wire at atmospheric pressure, using a high-speed movie camera. Their data show that at a heat flux of 35 kW/m^2 (full-load chiller-operating condition), the

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NOMENCLATURE

- a proportionality constant defined by equation (12) [s^{-1}]
- c flow strength parameter defined in equation (12) [dimensionless]
- c_b empirical constant in equation (8) used by Nakayama *et al.* [20]
- c_p specific heat of the liquid [J/kg-K]
- c_q empirical constant in equation (9)
- d_b bubble departure diameter [m]
- f bubble formation and departure cycle frequency [1/s]
- g acceleration due to gravity [m/s^2]
- h heat transfer coefficient [W/m^2-K]
- i_{fg} latent heat of vaporization of the liquid [J/kg]
- k thermal conductivity of the liquid [W/m-K]
- n empirical constant in asymptotic matching
- n_s nucleation site density [$1/m^2$]
- Pr liquid Prandtl number [dimensionless]
- q total heat flux [W/m^2]
- q_{ex} sensible heat flux on external surface [W/m^2]
- q_{inf} external heat flux within the area of influence [W/m^2]
- q_{lat} latent heat flux [W/m^2]
- Q_{bub} time-averaged heat transfer rate per bubble site [W]
- r radial distance from the center of the nucleation site [m]
- t time [s]
- t_c time period of the bubble departure and formation cycle [s]
- T_{sat} temperature of saturated pool liquid [K]
- T_w wall temperature [K]
- u_∞ radial velocity at the outer edge of hydrodynamic boundary layer [m/s].

Greek symbols

- α thermal diffusivity of the liquid ($k/\rho c_p$) [m^2/s]
- δ_{th} thermal boundary-layer thickness [m]
- ν kinematic viscosity of the liquid [m^2/s]
- ρ density of saturated pool liquid [kg/m^3]
- ρ_v density of vapor under bubble departure conditions [kg/m^3]
- σ surface tension of the liquid [N/m].

latent heat transport is about 10% of the total heat flux.

In addition to these studies, Thome [10] concluded that the latent heat transfer is of minor importance in nucleate pool boiling of binary mixtures. Tong *et al.* [4] also concluded that sensible, rather than latent, heat transport plays the dominant role in nucleate boiling. The theories for bubble growth on a heated surface of Han and Griffith [1] and Mikic and Rohsenow [11] neglect microlayer evaporation.

This discussion shows that the latent part of the nucleate boiling heat transfer on plain surfaces is not significant compared to the sensible part for practical operating conditions. This suggests that the contribution of the microlayer evaporation to the total heat transfer rate is of second-order importance. Nevertheless, it gives rise to bubble growth that, in turn, causes transient heat transfer to the periodic thermal boundary layer. For these reasons, the transient heat transfer caused by the pumping action of the bubble growth and detachment is considered to be the dominant heat transfer mechanism in nucleate pool boiling.

Boiling models for plain surfaces

An early popular semi-empirical correlation was developed by Rohsenow [12]. Rohsenow assumed a form similar to correlations established in single-phase forced convection. The Rohsenow correlation requires knowledge of an empirical term C_{sf}, which depends upon the fluid–surface combination. The correlation needs determination of two more empirical constants that determine the slope of the boiling curve and Prandtl number dependency. The Rohsenow correlation has been successful in predicting the effect of pressure.

Tien [13] presented a pool-boiling model based on the hydrodynamic similarity between the flow field associated with a rising bubble column and an inverted stagnation flow, as illustrated by Fig. 1(a) and (b). Assuming laminar, axisymmetrical stagnation flow, the following analytical relations were used:

$$u_\infty = ar \tag{1}$$

$$\frac{hr}{k} = 1.32\left(\frac{u_\infty r}{\nu}\right)^{0.5} Pr^{0.33} \tag{2}$$

where a is a proportionality constant related to the velocity field outside the hydrodynamic boundary layer and r is the radial distance from the center of the axisymmetric flow. Tien visualized pool boiling as steady-state, single-phase convective heat transfer across the thermal boundary layer induced by the rising bubbles. He assumed that the area of influence of a nucleation site was a circle of diameter equal to

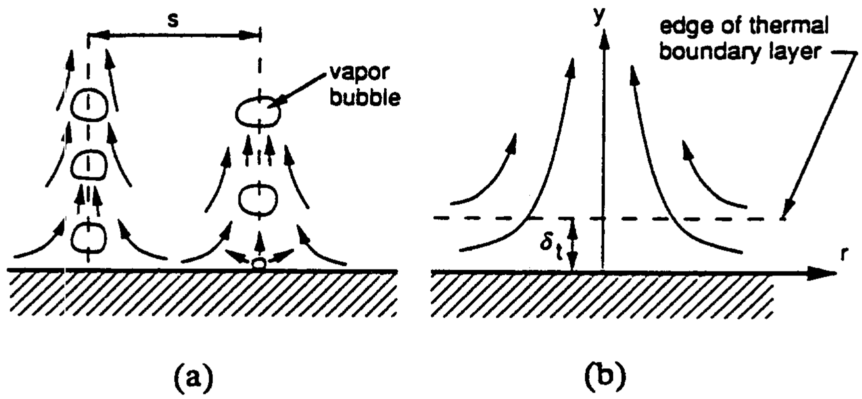

Fig. 1. (a) Actual liquid flow in nucleate boiling; (b) idealized inverted stagnation flow.

the bubble spacing. Based on the nucleation site density (n_s) data of Yamagata *et al.* [14] for boiling of water, Tien concluded that the quantity $a/n_s v = 2150$ and is independent of fluid properties. Combining this information with equations (1) and (2), he found the following expression for the pool-boiling heat flux.

$$q_{ex} = 61.3k(n_s)^{0.5} Pr^{0.33}(T_w - T_{sat}). \quad (3)$$

The model is in general agreement with data at moderate to high nucleation site densities for a wide variety of fluids. However, the assumption that $a/n_s v$ = constant has not been confirmed.

Zuber [15] analyzed nucleate pool boiling using an analogy with turbulent natural convection and expressed the nucleate boiling heat flux as a function of wall superheat and nucleation site density. He concluded that nucleate boiling consists of two regions: (a) the region of isolated bubbles; and (b) the region of interference. In the isolated bubble regime, he assumed that the heat transfer is caused by the "up-draught" induced circulation. Nishikawa and Fujita [16] also derived a similar correlating equation of heat transfer in nucleate boiling. They assumed that the main driving force for convection in nucleate boiling is the stirring action of the generated bubbles and used the analogy between nucleate boiling and free convection.

Mikic and Rohsenow [17] presented a mechanistic model for nucleate boiling. Following Han and Griffith's [1] pioneering work on the mechanism of heat transfer in nucleate pool boiling, they assumed that the periodic bubble formation and departure resulted in transient conduction heat transfer to the liquid within the area of influence of the nucleation site. They assumed that bubbles growing at nucleation sites influence heat transfer over an area of a circle of twice the bubble departure diameter. On the uninfluenced surface area, heat is transferred by natural convection. Individual areas of influence of the nucleation sites are assumed not to overlap even at high heat fluxes. Like Tien's stagnation flow model, this model is another demonstration that single-phase convection heat transfer concepts can be extended to nucleate boiling.

Dhir and Liaw [18] proposed a quantitative model for nucleate boiling on plain surfaces. Evaporation at stationary vapor stems at the wall is assumed to be the dominant mode of heat transfer. Heat is conducted from the wall into the liquid thermal layer surrounding the stems and is used in evaporation at the stationary liquid–vapor interface. The model requires the size distribution of active cavities and uses contact angle as a dominant parameter. It is applicable only to fully developed nucleate boiling, but not to the isolated bubble regime.

THEORETICAL MODELING OF THE SENSIBLE HEAT FLUX (q_{ex})

The Mikic and Rohsenow model [17] for nucleate boiling was chosen as the starting point for modeling q_{ex}, because this is the only model that is based on the physically realistic assumption of transient heat transfer during the bubble cycle, takes into account the periodic bubble growth and departure, and recognizes the frequency (f) of the bubble cycle as an important parameter. According to the model, a bubble departing from the heated surface removes a part of the superheated thermal layer within its area of influence (a circle of twice the departure diameter). Following the departure of the bubble and the superheated layer, the liquid at T_{sat} from the main body of the pool rushes into the area of influence and contacts the heating surface at T_w. This causes a sudden increase at the liquid surface temperature from T_{sat} to T_w. The Rayleigh solution for transient wall conduction heat flux within the area of influence is

$$q_{\mathrm{inf,Ra}}(t) = \frac{k(T_w - T_{sat})}{\sqrt{\pi \alpha t}} \quad (4)$$

where the denominator of the above equation represents the thermal boundary layer thickness.

The time-averaged external heat flux within the area of influence is

$$q_{\text{inf}} = \frac{1}{t_c}\int_0^{t_c} q_{\text{inf,Ra}}(t)\,\mathrm{d}t = 2k\sqrt{\frac{f}{\pi\alpha}}(T_w - T_{\text{sat}}). \quad (5)$$

Multiplying q_{inf} by the circular area of influence, πd_b^2, gives the heat transfer per bubble site, which is corrected for the projected area by multiplying by the nucleation site density (n_s) to obtain the external heat flux, as follows.

$$q_{\text{ex,MR}} = 2\sqrt{\pi k\rho c_p f}\; d_b^2 n_s (T_w - T_{\text{sat}}). \quad (6)$$

The essence of the Mikic and Rohsenow model is that if wall superheat ($T_w - T_{\text{sat}}$), bubble frequency (f), bubble departure diameter (d_b) and nucleation site density (n_s) are known, one can predict the sensible part (q_{ex}) of the boiling heat transfer. To the best of our knowledge, the Mikic and Rohsenow model has never been tested against the required five quantities observed simultaneously. Mikic and Rohsenow introduced two empirical constants into the model, that accounted for nucleation site density (n_s). They backed out these constants from the wall heat flux (q_{ex}) vs wall superheat ($T_w - T_{\text{sat}}$) data, rather than from nucleation site density data. This method does not conclusively validate their model.

Nakayama et al. [19, 20] *boiling data*

Simultaneous measurements of the five quantities (q_{ex}, $T_w - T_{\text{sat}}$, f, d_b, and n_s) required to validate the Mikic and Rohsenow model have not been reported in the published literature for plain or enhanced surfaces, with one exception. The only such data found were reported by Nakayama *et al.* [19, 20]. The data were taken for R-11 boiling on the Thermoexcel-E surface with wall superheat ranging from 0.30 to 1.75 K (0.8–31 kW/m^2). The Thermoexcel-E surface geometry consists of a flat horizontal surface having subsurface tunnels covered with pores, through which vapor bubbles escape into the bulk liquid. The overall boiling heat transfer from the Thermoexcel-E surface is the sum of the latent heat transfer occurring inside the subsurface tunnels and the sensible heat transfer due to the external, single-phase convection occurring from the external surface. The subsurface tunnels of the Thermoexcel-E surface promote a high evaporation rate and nucleation site density; hence, a much higher heat flux is expected as compared to that on the plain surface for the same wall superheat.

For R-11 boiling on the Thermoexcel-E surface, Nakayama *et al.* [20] counted the active nucleation sites per unit projected area (n_s), the bubbles leaving each site per second (f) and measured the bubble departure diameter (d_b). Using these bubble dynamics data, they were able to compute the latent heat flux per unit projected area by the following equation

$$q_{\text{lat}} = \frac{\pi}{6}\rho_v i_{fg} d_b^3 f n_s \quad (7)$$

where ρ_v and i_{fg} are the vapor density and the latent heat of vaporization, respectively. Nakayama *et al.* [20] expressed their bubble departure diameter by the following semi-empirical equation

$$d_b = c_b\left(\frac{2\sigma}{(\rho-\rho_v)g}\right)^{1/2} \quad (8)$$

which is similar to the Fritz [21] equation. In this formulation, Fritz neglected inertial forces and assumed that the bubble would break off when the buoyancy force exceeds the surface tension force holding the bubble to the heating surface. Nakayama *et al.* backed out the empirical constant $c_b = 0.442$ from their R-11 boiling data.

Nakayama *et al.* calculated the external convection part (q_{ex}) by subtracting the latent part (q_{lat}) from the measured total heat flux (q). They correlated q_{ex} in the following steady-state form, originally proposed by Zuber [15].

$$q_{\text{ex}} = \left(\frac{T_w - T_{\text{sat}}}{c_q}\right)^{5/3} n_s^{1/3} \quad (9)$$

where c_q is an empirical constant that depends upon the fluid and pressure. Nakayama *et al.* backed out $c_q = 1.95$ from their experimental data of q_{ex} on the Thermoexcel-E surface. The absence of bubble frequency "f" and departure diameter "d_b" in equation (9) distinguishes it from equation (6), which is based on physically more realistic transient quenching model. Figure 2 from Nakayama *et al.* [20] shows their measured ratio of latent to total heat flux (q_{lat}/q) for: (1) a plain surface, and (2) two Thermoexcel-E boiling surfaces (#1 and #3). The figure shows that at $q = 31$ kW/m^2, 73% of the overall heat transfer on surface #1 is sensible. Their measurements of latent and sensible heat flux on the plain surface are consistent with other studies that have concluded negligible latent heat contribution of microlayer evaporation on plain surfaces. Because of the similarity in the external boiling mechanisms, the microlayer evaporation beneath the growing bubble should also be insignificant on the Thermoexcel-E surface.

Table 1 summarizes the detailed R-11 bubble dynamics data of Nakayama *et al.* [19, 20] on the 46 fins/inch Thermoexcel-E surface #1. The column labeled q_{lat} is calculated using equation (7). The tabled q_{ex} values are calculated as ($q - q_{\text{lat}}$). The R-11 thermophysical properties ($p_{\text{sat}} = 1.0$ atm) used in the calculations are: $\rho_v = 5.83$ kg/m^3, $i_{fg} = 181.91$ kJ/kg, $k = 0.089$ W/m-K, $\rho = 1479.4$ kg/m^3, $c_p = 884$ J/kg-K and $Pr = 4.21$. The d_b values listed in Table 1 are calculated using equation (8) with $c_b = 0.442$, as recommended by Nakayama *et al.* [20]. This is because Nakayama *et al.* did not report measured d_b data. Using the Table 1 data with $d_b = 0.70$ mm and equation (7) will not yield the q_{lat}/q results shown on Fig. 2. This is because d_b also depends on heat flux, which is not accounted for by equation (8). Nakayama *et al.* [20] state that they observed a +20 to −20% vari-

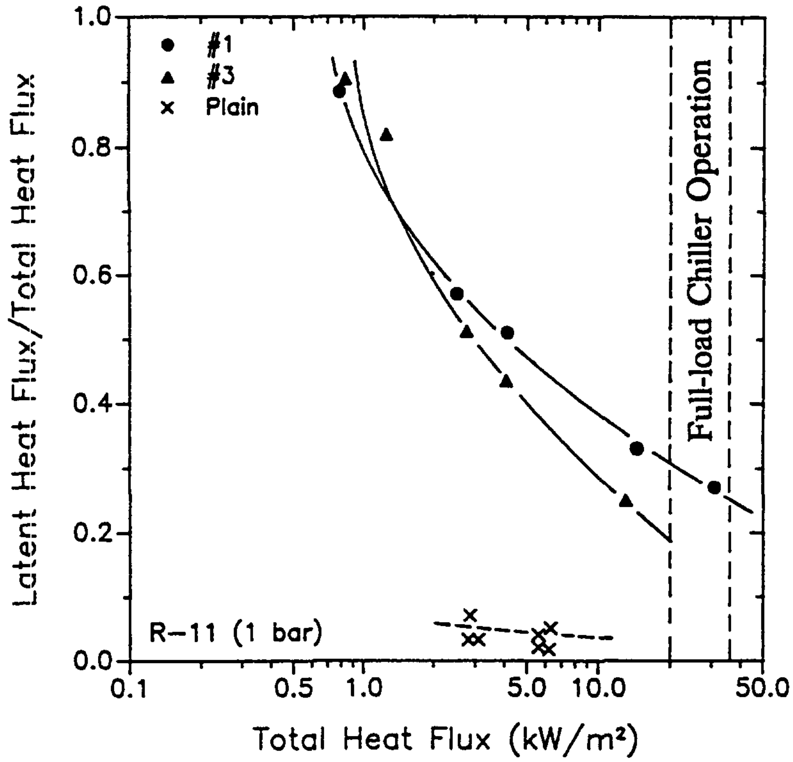

Fig. 2. Ratio of q_{lat}/q vs q for the two Thermoexcel-E surfaces and a plain surface; Nakayama *et al.* [19].

Table 1. Analysis of Nakayama *et al.*'s [19, 20] heat transfer and bubble dynamics data

	Measured heat transfer and bubble dynamics data					Analysis		
Sr. #	q (kW/m^2)	$T_w - T_{sat}$ (K)	f (1/s)	d_b (mm)	n_s (1/m^2)	q_{lat} (kW/m^2)	q_{ex} (kW/m^2)	q_{lat}/q
1	0.77	0.26	34	0.7	35 000	0.23	0.54	0.294
2	2.40	0.45	60	0.7	57 000	0.65	1.75	0.271
3	4.00	0.60	80	0.7	75 000	1.14	2.86	0.286
4	13.80	1.15	150	0.7	151 000	4.31	9.49	0.313
5	31.00	1.75	288	0.7	185 000	10.15	20.85	0.327

ation in d_b as the heat flux was increased from the lowest to the highest values on Fig. 2. The present authors calculated the values of d_b required for q_{lat}/q to agree with the Fig. 2 data. These varied from $d_b = 1.01$ mm at $q = 0.77$ kW/m^2 to $d_b = 0.66$ mm at $q = 31.0$ kW/m^2. These d_b values are 44% higher to 9% lower than the reference $d_b = 0.70$ mm in Table 1.

It was also found that the q_{ex} predicted by the Nakayama *et al.* correlation (equation (9)) agreed within -11 to $+8$% with the q_{ex} values in Table 1. However, the q_{ex} value obtained from Fig. 2 as $(q - q_{lat})$ at the lowest heat flux ($q = 0.77$ kW/m^2) was only 17% as large as the value predicted by equation (9). Other q_{ex} values obtained this way were up to 50% different from the values given by equation (9). Because of the uncertainty in the Fig. 2 data, no attempt was made to back out d_b and a fixed $d_b = 0.70$ mm is used in Table 1.

The significant latent heat transport on the Thermoexcel-E surface is because of the thin film evaporation inside the tunnels. However, the sensible heat transport mechanism from the external surface is still analogous to that on the plain surface. Thus, the Mikic and Rohsenow model (equation (6)) and Tien model (equation (3)) may be tested against the external sensible flux data on the Thermoexcel-E surface. The applicability of these models was assessed for the Nakayama *et al.* [19] R-11 sensible heat flux (q_{ex}) data on the Thermoexcel-E surface #1. Figure 3 compares

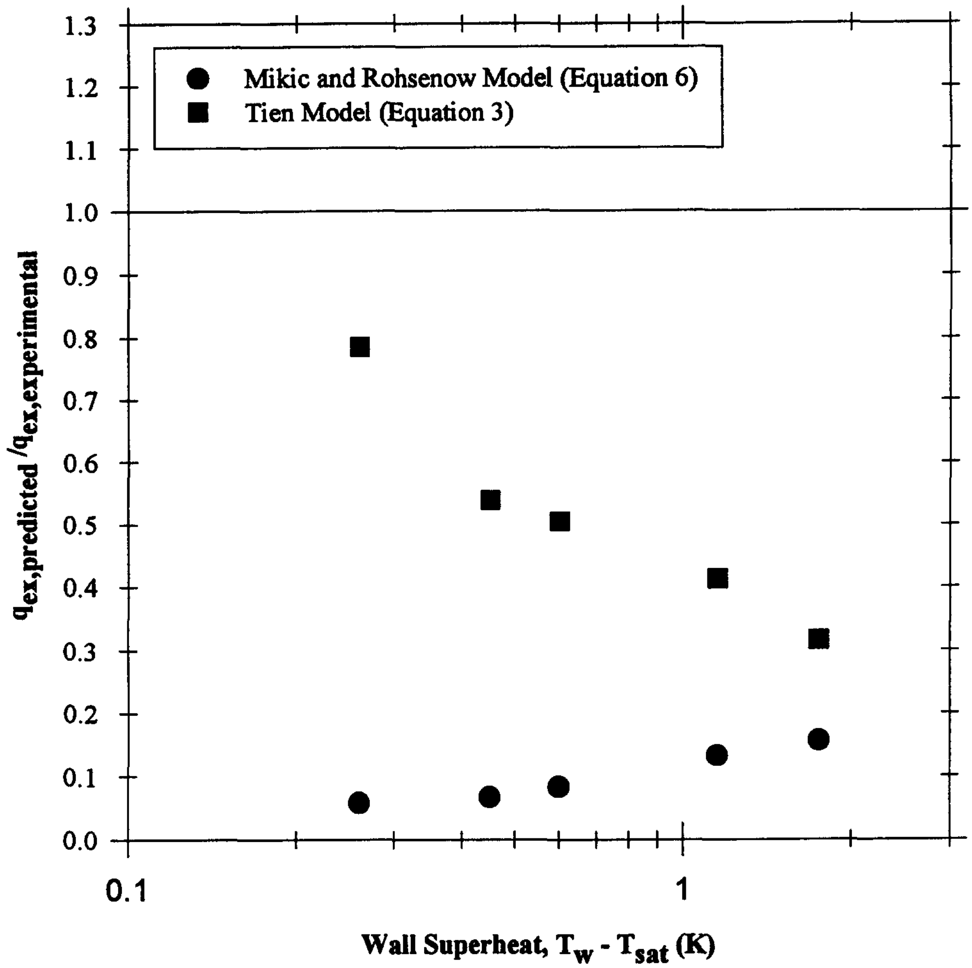

Fig. 3. Predictability of Nakayama *et al.*'s [19, 20] R-11 external boiling heat flux data by the Mikic and Rohsenow and Tien models.

the experimental q_{ex} data with predictions made by the Mikic and Rohsenow and Tien models. The figure shows that the Mikic and Rohsenow model (equation (6)) greatly underpredicts the q_{ex} data. The Tien model (equation (3)) also underpredicts q_{ex} by 24–67%.

Chien [22] *boiling data*

Recently, Chien [22] measured bubble dynamics and heat transfer data using a high speed photography system. The system included a video camera that can take photographs up to 6000 frames/s and a sophisticated motion analyzer for data analysis. A 60 mm micro lens with extension tubes was used on the video camera to take close-up photographs of the boiling surface. The high speed pictures were recorded on a special tape, which facilitated play back on a CRT at 20–30 power of magnification, and at slow motion (1–30 frames/s) by using the motion analyzer. This setup allowed detailed observation of the bubble growth and departure cycle and estimation of the nucleation site density on the boiling surface.

Chien used a single-tube pool boiling test cell made of glass to take the bubble dynamics and heat transfer data using R-123 at 1 atmospheric pressure. Further details of the test facility and the procedure are given by Chien and Webb [23]. The data were taken on a 19.1 mm diameter, horizontal copper tube having a pored-type structured surface similar to the Thermoexcel-E surface used by Nakayama *et al.* [19, 20], as discussed above. Chien's subsurface tunnels were made by using a circular tube with 40 fins/in of 0.6 mm fin height. The tube was covered with a thin sheet with pores of 0.23 mm diameter. Chien's bubble dynamics data were measured at three different circumferential locations for each heat flux. His measurements of the overall heat flux (q), wall superheat ($T_w - T_{sat}$), bubble frequency (f), bubble departure diameter (d_b) and nucleate site density (n_s) are summarized in Table 2. Chien observed that while decreasing heat from the highest to lowest value, the bubble departure diameter increased by 15–26% for different test runs. For comparable heat fluxes, this is about half the variation reported by Nakayama *et al.* [20] for R-11. Table 2 also tabulates the resulting values of latent heat flux (q_{lat} as calculated by equation (7)), external sensible heat flux ($q_{ex} = q - q_{lat}$) and the latent

Table 2. Analysis of Chien's [22] heat transfer and bubble dynamics data

	Measured heat transfer and bubble dynamics data					Analysis		
Sr. #	q (kW/m^2)	$T_w - T_{sat}$ (K)	f (1/s)	d_b (mm)	n_s (1/m^2)	q_{lat} (kW/m^2)	q_{ex} (kW/m^2)	q_{lat}/q
1	30.30	2.14	126	0.68	243 650	5.57	24.73	0.184
2	23.00	1.61	107	0.72	212 351	4.90	18.10	0.213
3	15.70	1.22	95	0.75	160 000	3.70	12.00	0.236
4	9.96	1.00	87	0.76	120 000	2.65	7.31	0.266
5	5.81	0.92	71	0.78	82 000	1.59	4.22	0.274
6	2.63	0.76	57	0.78	51 000	0.80	1.83	0.303
7	30.30	2.59	117	0.69	174 087	3.86	26.44	0.127
8	23.00	1.98	104	0.72	149 448	3.35	19.65	0.146
9	15.70	1.69	93	0.76	98 555	2.32	13.38	0.148
10	9.96	1.54	85	0.79	68 907	1.67	8.29	0.167
11	5.81	1.43	75	0.82	45 279	1.08	4.73	0.186
12	2.63	1.26	65	0.83	29 536	0.63	2.00	0.241
13	23.00	2.51	141	0.65	152 272	3.40	19.60	0.148
14	15.70	1.94	132	0.67	110 229	2.54	13.16	0.162
15	9.96	1.21	111	0.70	84 743	1.86	8.10	0.187
16	5.81	0.90	101	0.75	61 971	1.52	4.29	0.262
17	2.63	0.55	80	0.78	42 394	0.93	1.70	0.353
18	30.30	2.19	162	0.62	230 874	5.15	25.15	0.170
19	23.00	1.65	155	0.64	210 292	4.93	18.07	0.214
20	15.70	1.25	147	0.66	150 397	3.67	12.03	0.234
21	9.96	1.04	122	0.69	97 210	2.25	7.71	0.226
22	5.81	0.83	107	0.75	55 865	1.46	4.35	0.251
23	2.63	0.44	85	0.78	39 171	0.91	1.72	0.347

heat transfer contribution (q_{lat}/q). The R-123 thermophysical properties used at one atmosphere were: $\rho_v = 6.468$ kg/m^3, $i_{fg} = 170.44$ kJ/kg, $k = 0.0758$ W/m-K, $\rho = 1456.0$ kg/m^3, $c_p = 1001$ J/kg-K and $Pr = 5.39$.

The applicability of the Mikic and Rohsenow and Tien models was also assessed for Chien's data. Figure 4 compares the predictability of Chien's R-123 sensible heat flux (q_{ex}) data as made by both models. The figure shows that the Mikic and Rohsenow model (equation (6)) results in approximately 90% underprediction. The Tien model (equation (3)) underpredicts q_{ex} by 50–70% for most of the data.

PROPOSED ALTERNATE MODEL FOR q_{ex}

The above analysis of the Nakayama *et al.* and Chien boiling data suggests that neither the Mikic and Rohsenow nor the Tien models accurately model the actual physical process. As discussed before, microlayer evaporation beneath the growing bubble is expected to be too small to explain this discrepancy. It is proposed that the underprediction of Nakayama *et al.* q_{ex} data by the Mikic and Rohsenow model results from assuming pure conduction heat transfer through a quiescent liquid and neglecting the local convective effects of the liquid rushing towards the surface to fill the space of the departing bubble. It is also suggested that Tien's model underpredicts the data because it does not account for the cyclic nature of the phenomenon. Thus, Tien's model does not include the coupling between the micro-convection effects and the bubble growth and departure cycle. As a result, the bubble departure diameter (d_b) and frequency (f) are not present in the formulation of the model (equation (3)). Tien's assumption of the quantity "$a/n_s v$" being a constant (2150) makes "a" independent of frequency. In the isolated bubble regime, the velocity of liquid in the bubble wake (as dictated by "a" in equation (1)) is a local phenomenon and should be related to "f" rather than to "n_s". Once the n_s increases to the extent that the influence areas of individual sites start to overlap, the n_s is expected to have a retarding effect on the liquid velocity.

The liquid rushing to the nucleation site in the wake of the departing bubble induces eddies. These bubble-driven eddies impose a combination of front and inverted stagnation flows of liquid on the surface, which gives rise to an unsteady laminar forced-convection heat transfer from the nucleation site. A conceptual sketch of this flow pattern is shown in Fig. 5. The superheated liquid in the eddies would follow the bubble wake and would evaporate the liquid into the rising bubble. The nucleation site would act as an axisymmetrical unsteady stagnation point and the stagnation flow in the wake of the departing bubble would slow down the growth of the thermal boundary layer. Depending upon the length of the bubble cycle, the thermal boundary-layer thickness may attain an asymptotic steady-state value. Figure 6 shows the envisioned variation of the thermal boundary-layer thickness during one bubble cycle. Several analytical studies on impulsive stagnation flow heat transfer, e.g., Chao and Jeng [24], Chen and Chao [25], Watkins [26], Sano [27] show this trend. Experimental data on "jet impingement" have shown the heat trans-

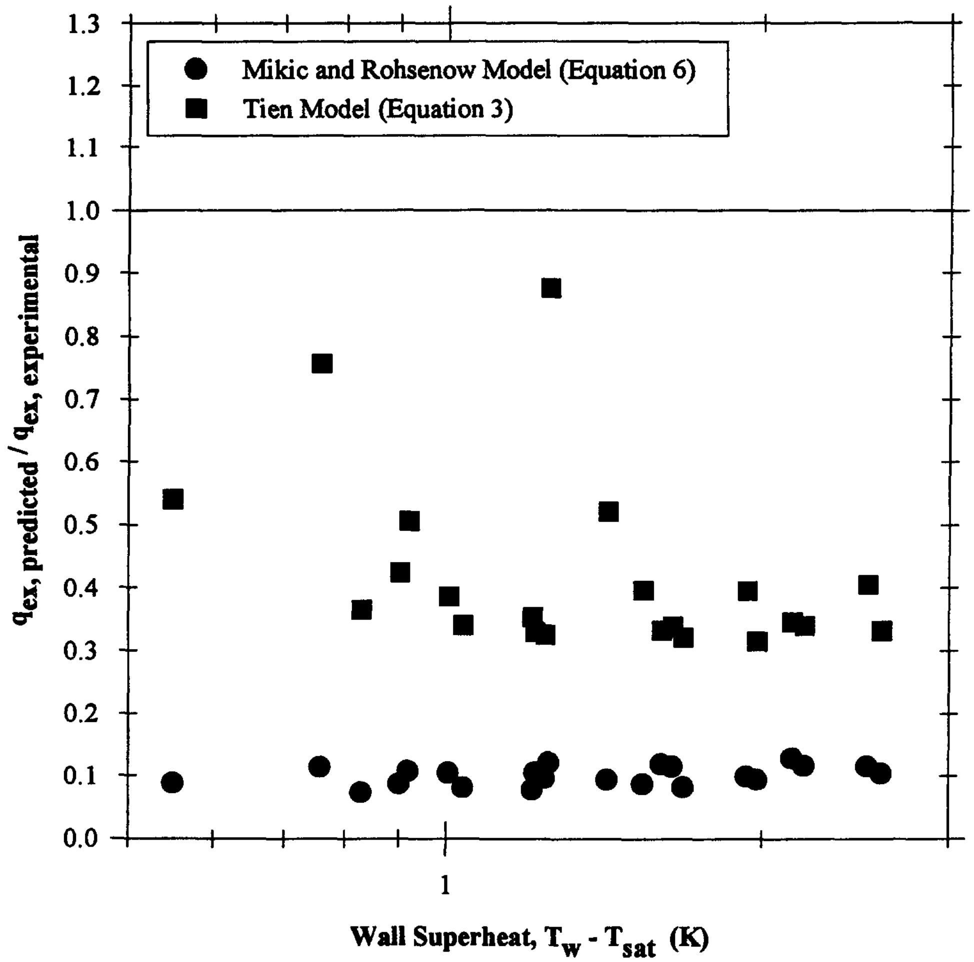

Fig. 4. Predictability of Chien's [22] R-123 external boiling heat flux data by the Mikic and Rohsenow and Tien models.

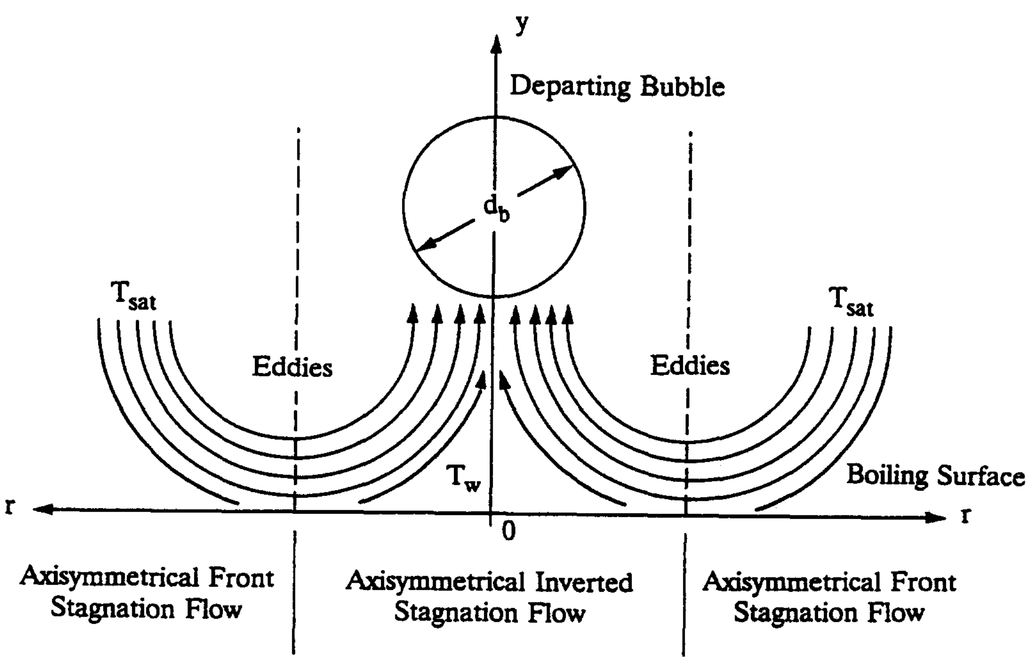

Fig. 5. Flow pattern in the wake of a departing bubble.

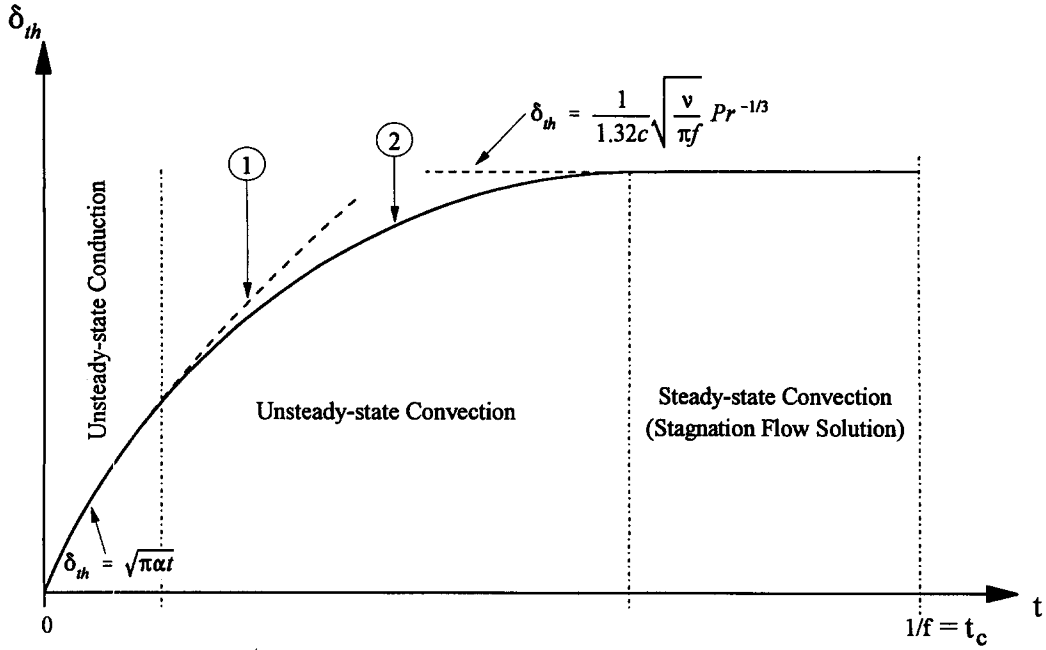

Fig. 6. Envisioned growth of the thermal boundary layer during one bubble cycle.

fer coefficients are several times higher than observed in other single-phase flow patterns. It is postulated that the thinner thermal boundary layer resulting from periodic impingement will explain the underprediction of the Nakayama *et al.* data by the Mikic and Rohsenow model.

Like the Mikic and Rohsenow model, the present model assumes that each bubble influences a boiling surface area of twice the bubble departure diameter. It also assumes that the steady-state boundary-layer thickness predicted by front stagnation flow theory is also applicable to the inverted stagnation flow in the wake of the bubble. The present model neglects free convection occurring on the uninfluenced areas. As supported by the experimental works on plain surfaces, the latent part of the heat transfer is assumed to be negligible. The following section deals with the formulation of the present model for q_{ex}.

FORMULATION OF THE PRESENT MODEL FOR q_{ex}

In the initial phase of a bubble cycle, Fig. 6 shows that unsteady heat transfer prevails with the following time-dependent thermal boundary-layer thickness proposed by Lord Rayleigh [28] (see equation (4)).

$$\delta_{th,Ra} = \sqrt{\pi\alpha t}. \qquad (10)$$

Then, convective effects become important, and departure from the unsteady-state conduction solution takes place. In Fig. 6, the dashed curve 1 represents the extrapolation of the thermal boundary layer thickness, as predicted by equation (10) beyond the unsteady-state conduction phase. The curve 2 in Fig. 6 shows the departure from the unsteady-state conduction thickness due to convective effects. With increasing time, the heat transfer approaches the steady-state stagnation solution as described in the Tien [13] model. Combining equations (1) and (2), one can deduce the following stagnation-flow thermal boundary-layer thickness applicable to the steady-state phase of the bubble cycle

$$\delta_{th,ss} = \frac{1}{1.32}\sqrt{\frac{\nu}{a}}\,Pr^{-1/3} \qquad (11)$$

where "ss" stands for steady-state stagnation and "a" is a proportionality constant related to the stagnation velocity field outside the hydrodynamic boundary layer, as defined by equation (1). The "a" (s^{-1}) reflects the strength of the stagnation flow field and is conveniently normalized as

$$a = c^2\left(\frac{\text{Velocity scale}}{\text{Length scale}}\right) \qquad (12)$$

where "c" is the non-dimensional form of "a" and will be called the "flow strength parameter." A power of 2 was chosen over "c" so that the convective correction to the Mikic and Rohsenow formulation of q_{ex} would vary linearly with c, thus making "c" physically more meaningful. The bubble departure diameter (d_b) and the bubble rise velocity were chosen as the appropriate length and velocity scales, respectively. Malenkov [29] shows by experimental data that the bubble rise velocity can be approximated as $\pi f d_b$. With the length and velocity scales established, equation (12) gives

$$a = \pi f c^2. \qquad (13)$$

Recall that Tien assumed that "a" depends only on

"n_s." The above equation relates "a" to "f." Substituting the above formulation of "a" in equation (11) gives:

$$\delta_{\text{th,ss}} = \frac{1}{1.32c}\sqrt{\frac{\nu}{\pi f}}Pr^{-1/3}. \tag{14}$$

The constant boundary layer thickness predicted by the above equation is shown as the horizontal asymptote of the curve 2 in the steady-state convection phase.

Asymptotic matching of the transient and steady-state heat transfer

The present analysis assumes that the beginning of the bubble cycle is dominated by transient conduction, and the final part of the cycle is dominated by steady-state micro-convection. These two asymptotic heat transfer modes are dictated by the respective thermal boundary layers described by equations (10) and (14). To include the transient and steady-state heat transfer rate during the bubble cycle, within the area of influence of each nucleation site, we propose the following expression for the overall heat flux within the area of influence, q_{inf}.

$$\left(\int_0^{t_c} q_{\text{inf}}(t)\,\mathrm{d}t\right)^n = \left(\int_0^{t_c} q_{\text{inf,Ra}}\,\mathrm{d}t\right)^n + \left(\int_0^{t_c} q_{\text{inf,ss}}\,\mathrm{d}t\right)^n \tag{15}$$

where n is an empirical constant. The above form of asymptotic matching has been frequently used in the literature where two asymptotic conditions are known. For example, this asymptotic, power-type addition was used by Churchill [30] to correlate the transition between natural convection and forced convection heat transfer. Along with several others, Kutateladze [31] has also used the same form to superimpose the nucleate and convective boiling components of flow boiling heat transfer. Most researchers have found "n" ranging from 2 to 3.

The power-type asymptotic approach assumes that the two effects are always present. Thus, in the present analysis, a "purely" transient conduction phase or a "purely" steady-state convection phase cannot be defined. The transient conduction heat flux, $q_{\text{inf,Ra}}(t)$, is given by equation (4). The steady-state heat flux, $q_{\text{inf,ss}}$, is found by using the steady-state boundary layer thickness given by equation (14), as follows

$$q_{\text{inf,ss}} = \frac{k(T_w - T_{\text{sat}})}{\delta_{\text{th,ss}}} = 1.32ck\sqrt{\frac{\pi f}{\nu}}Pr^{1/3}(T_w - T_{\text{sat}}) \tag{16}$$

where "c" is the empirical constant that quantifies the strength of the stagnation flow. As the steady-state heat flux, though frequency dependent, is constant, equation (15) is simplified to give the following time-averaged external heat flux, q_{inf} (W/m^2), within the area of influence of each nucleation site.

$$q_{\text{inf}} = \frac{1}{t_c}\left(\left(\int_0^{t_c} q_{\text{inf,Ra}}\,\mathrm{d}t\right)^n + (q_{\text{inf,ss}}\cdot t_c)^n\right)^{1/n}. \tag{17}$$

The above equation can also be considered as the asymptotic solution which connects the $(\pi\alpha t)^{1/2}$ boundary layer thickness in the transient conduction phase with that in the steady-state convection phase, as depicted on Fig. 6. Equation (17) can also be considered to be matching the inverse of respective boundary layer thicknesses in the two phases.

Substituting $q_{\text{inf,Ra}}$ and $q_{\text{inf,ss}}$ from equations (4) and (16) into the above equation and simplifying, we obtain.

$$q_{\text{inf}} = 2\sqrt{k\rho c_p}\sqrt{\frac{f}{\pi}}(T_w - T_{\text{sat}})\left(1 + \left(\frac{0.66\pi c}{Pr^{1/6}}\right)^n\right)^{1/n}. \tag{18}$$

As initially postulated by Han and Griffith [1] and later used by Mikic and Rohsenow [17], the present model also assumes that the bubble growing at a nucleation site influences heat transfer over an area of a circle of twice the bubble departure diameter (d_b). Thus, multiplying equation (18) by the area of influence, $\pi d_b{}^2$, gives the following expression for the time-averaged heat transfer rate (Watts) per bubble site.

$$Q_{\text{bub}} = 2\sqrt{\pi k\rho c_p f}\,d_b^2(T_w - T_{\text{sat}})\left(1 + \left(\frac{0.66\pi c}{Pr^{1/6}}\right)^n\right)^{1/n}. \tag{19}$$

Finally, multiplying the above equation by the projected area nucleation site density, n_s (sites/m^2), we obtain the following formulation of the overall external sensible heat flux (W/m^2).

$$q_{\text{ex}} = 2\sqrt{\pi k\rho c_p f}\,d_b^2 n_s(T_w - T_{\text{sat}})\left(1 + \left(\frac{0.66\pi c}{Pr^{1/6}}\right)^n\right)^{1/n}. \tag{20}$$

The initial terms outside the parentheses are the external heat flux predicted by the Mikic and Rohsenow model (equation (6)). Thus, the terms inside the parentheses constitute the convective correction to the Mikic and Rohsenow model. Using equation (6), equation (20) can be rewritten as

$$\frac{q_{\text{ex}}}{q_{\text{ex,MR}}} = \left(1 + \left(\frac{0.66\pi c}{Pr^{1/6}}\right)^n\right)^{1/n} \tag{21}$$

where $q_{\text{ex,MR}}$ is the external heat flux based on the Mikic and Rohsenow's transient conduction model given by equation (6). As expected, the ratio in equation (21) reduces to unity for $c = 0$, i.e. no convective effects.

Present model vs experimental data

Nakayama *et al.*'s data [19, 20] and Chien's data [22] were predicted by equation (20) and the results are shown on Figs 7 and 8. To best fit Nakayama *et*

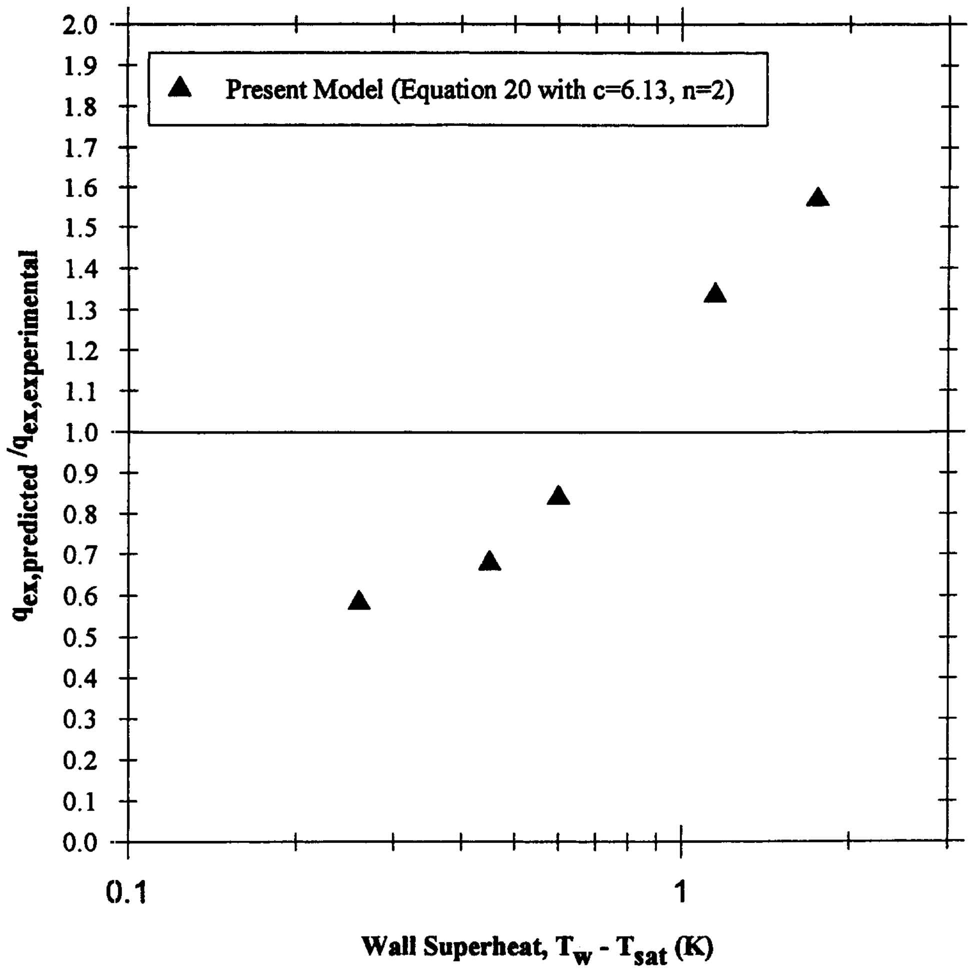

Fig. 7. Predictability of Nakayama *et al.*'s [19, 20] R-11 external boiling heat flux data by the present model.

al.'s R-11 data, the present model required $c = 6.13$ for $n = 2$ and $c = 6.15$ for $n = 3$. This suggests an insensitivity of the flow strength parameter "c" to the asymptotic matching power "n" within the commonly observed range of $2 \leqslant n \leqslant 3$. Figure 7 shows that the present model predicts the Nakayama *et al.* data within $\pm 40\%$. We have previously noted that equation (8) from Nakayama *et al.* (20) does not account for the effect of heat flux on bubble departure diameter. If the actual bubble diameters were available, the predictions will be improved.

Figure 8 shows the prediction of Chien's R-123 data made by the present model. To best fit Chien's data, the present model required $c = 6.42$ for $n = 2$ and $c = 6.45$ for $n = 3$. This again suggests the independence of "c" from "n". Figure 8 shows that the present model predicts Chien's data within $\pm 30\%$.

For both R-11 and R-123 data presented in this paper, the convective term in equation (20), $(0.66\pi c/Pr^{1/6})^n$, is much larger than the unity, which represents the transient conduction effects. This weak dependency of q_{ex} on the transient conduction contribution translates into its independence from "n." Based on the above analyses, $n = 2$ is recommended.

From the present model formulation, it appears that "c" is solely related to the geometry of the flow in the wake of rising bubbles, and is not expected to depend upon the fluid properties and heat flux. Note that the $c = 6.42$ for Chien's R-123 data is only 1.047 times higher than $c = 6.13$ for the Nakayama *et al.* R-11 data. This seems to suggest the independence of "c" from fluid properties and the possibility that a single value of "c" can reasonably predict both data sets. However, more boiling data are needed to improve the understanding of the flow strength parameter "c" in nucleate boiling and establish its independence from fluid properties.

The present transient model is proposed to improve the theoretical understanding of nucleate pool boiling mechanism on plain surfaces. Comparison of the predictive ability of the present model shown on Figs. 7 and 8 with that of the Mikic and Rohsenow and Tien models shown on Figs. 3 and 4 demonstrates that the proposed model gives much better predictions. Its further validation requires accurate prediction or measurement of active nucleation site density (n_s), bubble frequency (f) and bubble departure diameter (d_b). Correlations to predict f and d_b do exist, although

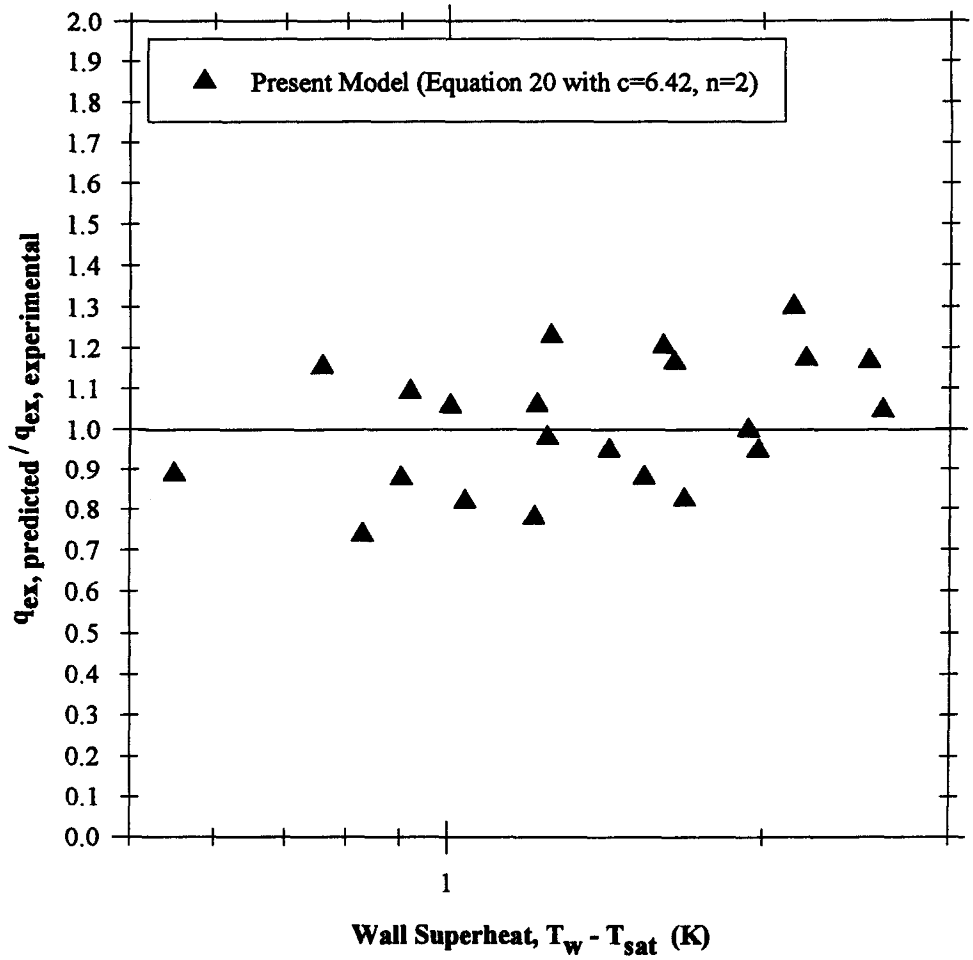

Fig. 8. Predictability of Chien's [22] R-123 external boiling heat flux data by the present model.

these are not regarded as "highly accurate." Comparable correlations for n_s do not exist. This is partly because n_s is affected by surface "roughness," whose effect has not been quantified. The authors suggest that the significance of this work is to establish the mechanism of nucleate pool boiling on plain surfaces for given n_s, f, d_b. It is recognized that more boiling data with different fluids of diverse thermophysical properties are needed to further validate the present q_{ex} model.

It is generally agreed that the latent term is small for boiling on plain surfaces. However, this is not the case for enhanced surfaces. The present model can be applied to enhanced surfaces by adding the latent and sensible terms as given by

$$q = \frac{\pi}{6}\rho_v i_{fg} d_b^3 f n_s + 2\sqrt{\pi k \rho c_p f}\, d_b^2 n_s (T_w - T_{sat}) \times \left(1 + \left(\frac{0.66\pi c}{Pr^{1/6}}\right)^n\right)^{1/n}. \quad (22)$$

Practical application of equation (22) to enhanced surfaces would require complimentary models, or correlations, to predict the site density (n_s), bubble frequency (f) and bubble departure diameter (d_b). Further details on the model can be obtained from Haider [32].

LOCAL CONVECTIVE EFFECTS IN NUCLEATE POOL BOILING

The present model for q_{ex} assumes that the local convective effects in pool boiling (as opposed to bulk convective effects in flow boiling) caused by the bubble-induced flow are important. Some other researchers have also recognized, though often implicitly, the importance of the local convective effects in nucleate pool boiling.

Tien's hydrodynamic model [13] for nucleate boiling is based on inverted stagnation flow. Although the model recognizes the importance of the convective effects, it does not account for the initial transience and is based on steady-state heat transfer. Later, in a discussion on Hsu's [33] transient-conduction based nucleation theory, Tien argued that transient convection, rather than transient conduction is the prevailing heat transfer mechanism in nucleate boiling. Based on the Yamagata *et al.* [14] optical measurements of the thermal boundary-layer thickness in nucleate pool boiling, Tien argued that the boundary-layer thickness in the vicinity of a nucleation site is less than that predicted by transient conduction and, thus, would be better predicted by transient convection. However, a nucleate boiling model based on transient convection has not been previously formulated.

The Zuber model [15] for nucleate boiling as previously discussed, is based on an analogy between nucleate boiling and turbulent natural convection. Nishikawa and Fujita [16] also used the same analogy for their model. However, these models are also based on single-phase, steady-state heat transfer.

Rohsenow's popular correlation [12] for nucleate boiling on plain surfaces is based on a forced convection model, in which the superficial velocity of the liquid rushing to the surface was chosen as the appropriate velocity scale. Based on experimental data, Rohsenow's correlation has a strong Prandtl number dependency, which is physically interpreted as the significance of the convective effects. Equation (6) shows that the Mikic and Rohsenow model has no Prandtl number dependency, a consequence of neglecting the convective effects. The Mikic and Rohsenow model is an exception, not the rule, in neglecting the convective effects.

In the above-mentioned studies, the significance of the local convective effects was recognized in nucleate pool boiling heat transfer; however, they are also recognized in other boiling-related phenomena, such as bubble nucleation. Ali and Judd [34] found that Hsu's [33] bubble nucleation theory based on transient conduction to the liquid, predicted bubble waiting time inconsistent with experimental data. They developed a bubble nucleation theory based on transient convection heat transfer to the liquid in the wake of the rising bubble. The bubble waiting time predicted by this theory agreed very well with experimental observations.

CONCLUSIONS

Based on the survey of existing models and understanding of nucleate pool boiling on plain surfaces, a rationally based analytical model is proposed to predict nucleate boiling. The model envisions that transient micro-convection, rather than transient conduction, from the external surface in the wake of departing bubbles is the dominant nucleate heat transport mechanism. A transient-convection formulation of the sensible heat transfer is presented, that accounts for the Prandtl number dependency missing in the transient conduction formulation. The present model provides an improved understanding of the mechanism of nucleate pool boiling on plain surfaces. Its application requires accurate prediction, or measurement, of active nucleation site density (n_s), bubble frequency (f) and bubble departure diameter (d_b). More boiling data with different fluids with diverse thermophysical properties are needed to establish the significance of micro-convection effects in nucleate pool boiling and test the "constancy" of the flow strength parameter "c." Although the R-11 and R-123 data analyzed in this paper show a relative insignificance of the transient conduction contribution, more data will be required to assess the transient conduction effects *vis-à-vis* steady-state convection. The overall heat flux (q) as the sum of latent heat flux (q_{lat}) as predicted by equation (7) and sensible heat flux (q_{ex}) as predicted by equation (20) is defined as the basis of a comprehensive pool boiling model.

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