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A new model for heat flow through macrolayer in pool boiling at high heat flux

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Abstract—An analytical heat transfer model for transient one-dimensional heat flow through a decreasing thickness macrolayer and with an invariant impressed heat flux at the heated solid–macrolayer interface has been numerically solved. The model predicts the average conduction heat flow rates at the liquid–vapour interface for water and methanol that are in excellent agreement with experimental values of input heat flux. It is shown that a non-dimensional parameter Z_0 may be used as an index of the combined effect of the physical parameters namely, initial macrolayer thickness, initial wall superheat and the impressed heat flux, on the conduction heat flux across the liquid layer.

INTRODUCTION

SEVERAL investigators [1–3] have demonstrated that in nucleate pool boiling at high heat flux, heat transfer is primarily by latent heat transport although the mechanism underlying this phenomenon is not yet fully known. Yu and Mesler [4], through measurement of the transient heated surface temperature during nucleate boiling of water, have shown the existence of a liquid film, called the macrolayer, between the growing vapour mass and the heating surface. It has been hypothesized that this macrolayer plays a predominant role in transferring thermal energy from the heated surface to the boiling fluid.

Analytical as well as experimental investigations into the mechanism of formation and heat flow through the macrolayer have been carried out by several researchers [5–10]. Bhat *et al.* [6] analysed the mechanism of formation of the macrolayer and derived an expression for the initial macrolayer thickness. Heat transfer through the macrolayer has been analysed [7] assuming constant heated wall temperature and linear temperature distribution in the macrolayer. They hypothesized that the heat transfer in the high heat flux region takes place mainly due to the heat conduction through the liquid macrolayer. In a subsequent paper [8] experimental values of the initial macrolayer thickness and frequency of vapour mass for water at atmospheric pressure on a horizontal copper surface have been reported. Prasad [9] also measured the initial macrolayer thickness in water and methanol at atmospheric pressure on a horizontal copper surface. He found that the values for water were in agreement with those reported by Bhat *et al.* Prasad [9] analysed heat transfer through the macrolayer considering constant heat input at the bottom of the heater and assuming a linear temperature distribution in the macrolayer. His model predicted better values of heat flux than those of Bhat *et al.* [7]. The basic differences in the two heat transfer models lie in the assumption of a constant wall superheat by Bhat *et al.* [7] whereas in the model of Prasad [9] an initial temperature gradient in the solid was assumed.

Chyu [10] solved the one-dimensional transient moving boundary conduction problem for heat transfer through a macrolayer that evaporates into the vapour mass assuming an invariant heated surface temperature. The location of the liquid–vapour interface, i.e. transient macrolayer thickness has been assumed to be of the form obtained from the exact solution of the semi-infinite solidification problem, namely, Neuman's solution. It was shown [10] that the rate of heat conduction through the macrolayer is essentially too small to justify the major contribution to the heat flow through the macrolayer as claimed by Bhat *et al.* [7] and Prasad [9]. He consequently suggested an alternative model that hypothesizes the major contribution to heat flow through the microlayers formed under the individual bubbles within the macrolayer. This appears unlikely because, in the case when the entire energy supplied to the heating surface is to be transferred through this small fraction (one-ninth) of the area, it would require an extremely high rate of heat transfer. The detailed reasoning for this statement is given in the Appendix.

Although there appear to be good reasons to believe that the liquid layer does play an important role in transporting heat from the heated surface to the boiling fluid, the deviations between experimental results and those predicted using macrolayer conduction models have been found to be high. Therefore, it is necessary to model the conduction heat flow phenomenon on more realistic lines. As the heat flux at the wall is usually invariant, any resistance to heat flow from the heated surface to the vapour is likely to

NOMENCLATURE

h_{fg}	latent heat of vaporization [J kg^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
q	heat flux [W m^{-2}]
$\bar{q}$	time averaged heat flux [W m^{-2}]
T	temperature [K]
ΔT	wall superheat, $(T_w - T_s)$ [K]
t	time [s]
t_c	cycle period [s]
y	distance from the heating surface [m]
Z	dimensionless parameter $k_\ell \Delta T/\delta q_i$.

Greek symbols

α	thermal diffusivity [m^2 s^{-1}]
δ	macrolayer thickness [m]
ρ	mass density [kg m^{-3}].

Subscripts

0	initial
i	input
ℓ	liquid
ℓv	liquid–vapour interface
s	saturation
w	wall.

raise the temperature of the solid surface; the fact experimentally verified by several investigators [4, 11]. It appears that the heat flow model with invariant heat flux at the heated solid surface is an essential boundary condition that allows wall temperature variations as hypothesized to represent a situation closer to the actual physical condition. In this paper an analytical heat transfer model, involving transient one-dimensional heat flow through a decreasing thickness macrolayer and with time invariant impressed heat flux at the superheated solid wall below the macrolayer, has been solved using the finite difference technique. The results of this numerical solution technique with respect to the temperature distribution in the macrolayer and the conduction heat flow at the liquid–vapour interface have been compared with heat transfer models proposed by previous investigators.

HEAT TRANSFER MODEL

Figure 1 depicts the heat transfer model with one-dimensional heat conduction in the region $(0 \leqslant y \leqslant \delta)$ bounded by the plane $y = \delta$ at a fixed temperature T_s and the plane $y = 0$ has prescribed heat flux. It is assumed that the macrolayer attains the uniform superheated temperature equal to the wall temperature at the end of the waiting period, the temperature of the evaporating face of the macrolayer drops to the saturation value at the instant of vapour-mass formation, and the effect of vapour stems in the macrolayer is negligible. The transient heat conduction equation along with initial and boundary conditions is

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial y^2}; \quad t > 0, 0 \leqslant y \leqslant \delta, \delta = \delta(t) \tag{1}$$

$$T(y, 0) = T_w \tag{1a}$$

$$T(\delta, t) = T_s; \quad t > 0 \tag{1b}$$

$$-k_\ell \left(\frac{\partial T}{\partial y}\right)_{y=0} = q_i \tag{1c}$$

$$\delta(0) = \delta_0 \tag{1d}$$

$$\delta = \delta_0 - \frac{1}{h_{fg}\rho_\ell} \int_0^t \left[-k_\ell \left(\frac{\partial T}{\partial y}\right)_{y=\delta}\right] dt. \tag{1e}$$

The closed form solution of equations (1) with prescribed conditions (1a)–(1e) is not possible mainly due to moving boundary $y = \delta(t)$. A finite difference method has therefore been used for the solution of the above boundary value problem with the moving boundary $y = \delta(t)$.

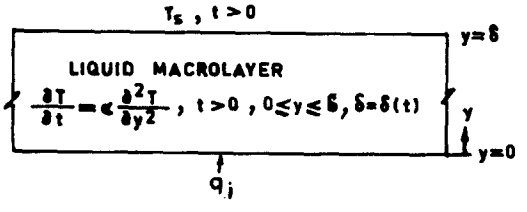

FIG. 1. Heat transfer model.

FINITE DIFFERENCE SOLUTION

The Crank–Nicolson implicit method of finite differences which is a combination of the explicit and the implicit method of finite differences, has been used. It gives the smallest truncation error and is unconditionally stable for all values of the time step Δt. The differential equation (1) can be written by this method as [12]

$$\frac{T_i^{n+1} - T_i^n}{\Delta t} = \tfrac{1}{2}\alpha \left[\frac{T_{i-1}^{n+1} + \bar{T}_{i+1}^{n+1} - 2T_i^{n+1}}{(\Delta y)^2} + \frac{T_{i-1}^n + T_{i+1}^n - 2T_i^n}{(\Delta y)^2}\right] \tag{2}$$

where the temperature at the nodal point $i \cdot \Delta y$ at time $n \cdot \Delta t$ is denoted by

$$T(i \cdot \Delta y, n \cdot \Delta t) = T_i^n$$

if $r = \alpha \cdot \Delta t/(\Delta y)^2$, then the above equation may be rearranged as

$$-rT_{i-1}^{n+1} + (2+2r)T_i^{n+1} - rT_{i+1}^{n+1} = rT_{i-1}^n + (2-2r)T_i^n + rT_{i+1}^n \quad (3)$$

where $i = 0, 1, 2, \ldots, N$.

Writing the difference equation for boundary conditions (1a)–(1c), we have

$$T_0^0 = T_1^0 = T_2^0 = \cdots = T_N^0 = T_w \quad (3a)$$

$$T_N^1 = T_N^2 = T_N^3 = \cdots = T_N^{n+1} = T_s \quad (3b)$$

and by central difference

$$\left(\frac{\partial T}{\partial y}\right)_{y=0} = \frac{T_1^n - T_{-1}^n}{2\Delta y}$$

or

$$q_i = \frac{k_\ell}{2\Delta y}(T_1^n - T_{-1}^n)$$

or

$$T_{-1}^n = T_1^n + \frac{2\Delta y q_i}{k_\ell}. \quad (3c)$$

Equation (3) gives $N+1$ simultaneous algebraic equations for the $N+1$ unknown temperatures at the time interval $(n+1)$ in terms of the known temperatures at the previous time interval n. In this case only N temperatures are needed which can be obtained by N equations.

The system of equations (3) when written in matrix form, gives a tridiagonal matrix as

$$\begin{bmatrix} (2+2r) & -2r & 0 & 0 \\ -r & (2+2r) & -r & \\ 0 & -r & (2+2r) & -r \\ \vdots & \vdots & \vdots & \vdots \\ & & -r & (2+2r) \end{bmatrix} \begin{bmatrix} T_0^{n+1} \\ T_1^{n+1} \\ \cdot \\ \vdots \\ T_{N-1}^{n+1} \end{bmatrix}$$

$$= \begin{bmatrix} (2-2r) & 2r & 0 & 0 \\ r & (2-2r) & r & \\ \cdot & \cdot & \cdot & \cdot \\ \vdots & \vdots & \vdots & \cdot \\ & & r & (2-2r) \end{bmatrix} \begin{bmatrix} T_0^n \\ T_1^n \\ T_2^n \\ \vdots \\ T_{N-1}^n \end{bmatrix} + \begin{bmatrix} \frac{4r\Delta y q_i}{k} \\ 0 \\ 0 \\ \vdots \\ 2rT_s\dagger \end{bmatrix} \quad (4)$$

The above tridiagonal matrix has been solved by the Thomas method.

In view of the sudden cooling ($T_w \to T_s$) of the liquid–vapour interface at $t > 0$, there is a singularity in the function. Numerical methods being somewhat unsatisfactory in the neighbourhood of the singularity, the best option would usually be a combination of analytical and numerical methods [13], an analytical solution for small time and a numerical method for larger time values. This switch over from analytical to numerical solution was at 1% of the total cycle period and it has been found that the sensitivity of the predicted results to the instant of switch over is very nominal, the maximum error being not more than 0.5%. Further the accuracy of the numerical method has been judged by comparing the results obtained with the exact and with the numerical solution for an assumed constant macrolayer thickness and an excellent agreement has been found both with respect to temperature distribution and heat flux.

To obtain the temperature profile near $t = 0$, equation (1) has been solved using the Laplace transform [14]; the solution thus obtained is

$$T(y,t) = T_w - (T_w - T_s)\left[\sum_{n=0}^{\infty}(-1)^n \operatorname{erfc}\frac{\delta(2n+1)-y}{2\sqrt{(\alpha t)}} + \sum_{n=0}^{\infty}(-1)^n \operatorname{erfc}\frac{\delta(2n+1)+y}{2\sqrt{(\alpha t)}}\right]$$

$$+ \frac{q_i}{k_\ell}\left[\sum_{n=0}^{\infty}(-1)^n \left\{2\left(\frac{\alpha t}{\pi}\right)^{1/2} e^{-(2n\delta+y)^2/4\alpha t} - (2n\delta+y)\operatorname{erfc}\frac{2n\delta+y}{2\sqrt{(\alpha t)}}\right\}\right.$$

$$\left. - \sum_{n=0}^{\infty}(-1)^n \left\{2\left(\frac{\alpha t}{\pi}\right)^{1/2} e^{-(2n\delta+2\delta-y)^2/4\alpha t} - (2n\delta+2\delta-y)\operatorname{erfc}\frac{(2n\delta+2\delta-y)}{2(\alpha t)}\right\}\right]. \quad (5)$$

With the above combination of an analytical and numerical method, a Fortran program was developed and was run on a DEC 2050 computer. With water and methanol at atmospheric pressure as macrolayer

Table 1. Boiling conditions for water and methanol

ΔT_0 (K)	δ_0 (μm)	q_i (MW m^{-2})
	Water	
20	30, 35, 50, 100	0.5, 0.75, 1.0, 1.35
20	75	1.0
30	50, 70, 100, 125	1.0
30	75	0.5, 0.75, 1.0, 1.25, 1.5
40	30	1.0, 1.35
40	35	0.75, 1.0, 1.35
40	50, 100	0.5, 0.75, 1.0, 1.35
40	75, 70	1.0
50	50	0.75, 1.0, 1.35
60	50, 100	0.75, 1.0, 1.35
	Methanol	
20	30, 50, 70	0.5, 1.0
30	30, 50, 70	0.5, 1.0
40	30, 50, 70	0.5, 1.0

† At the first step this term is $r(T_w - T_s)$.

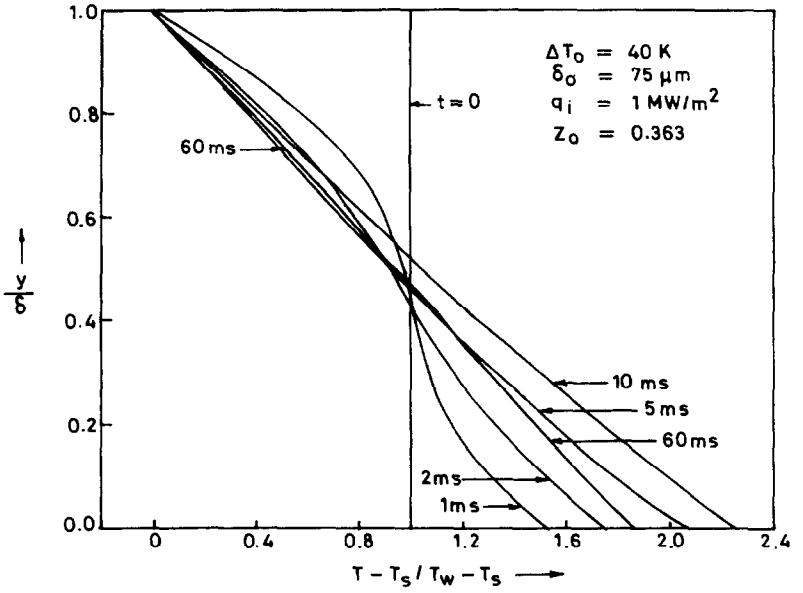

FIG. 2. Temperature profile in the macrolayer.

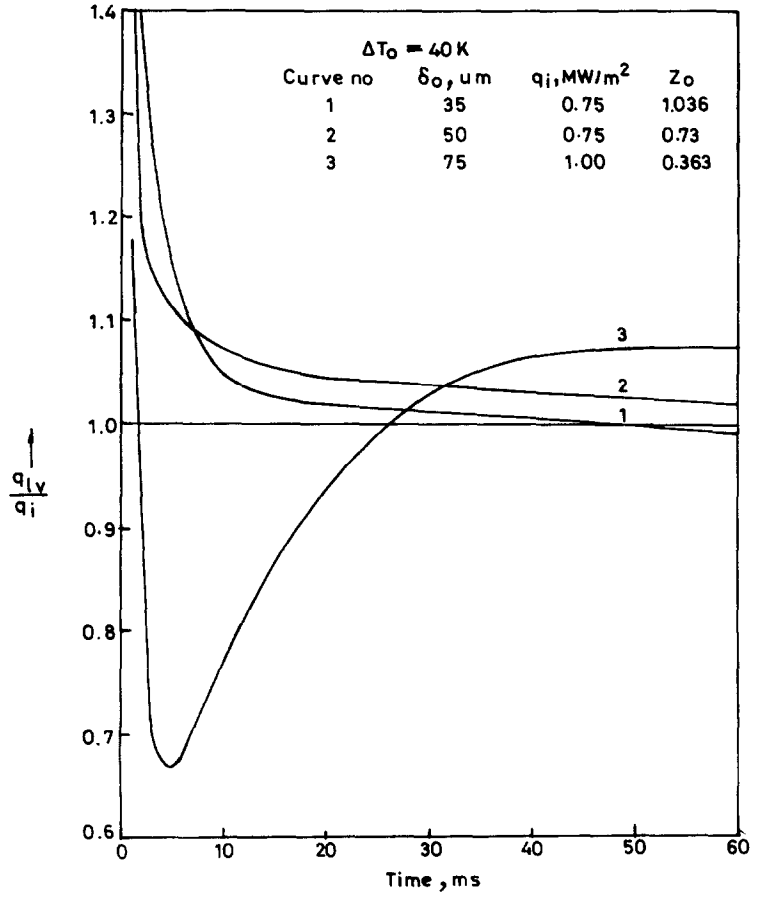

FIG. 3. Liquid–vapour interface heat transfer transients.

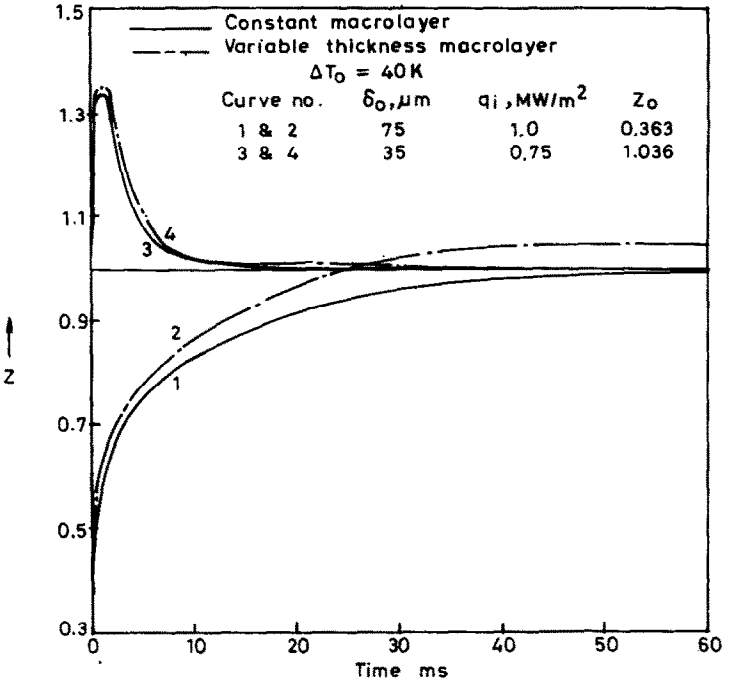

FIG. 4. Variation of Z with time.

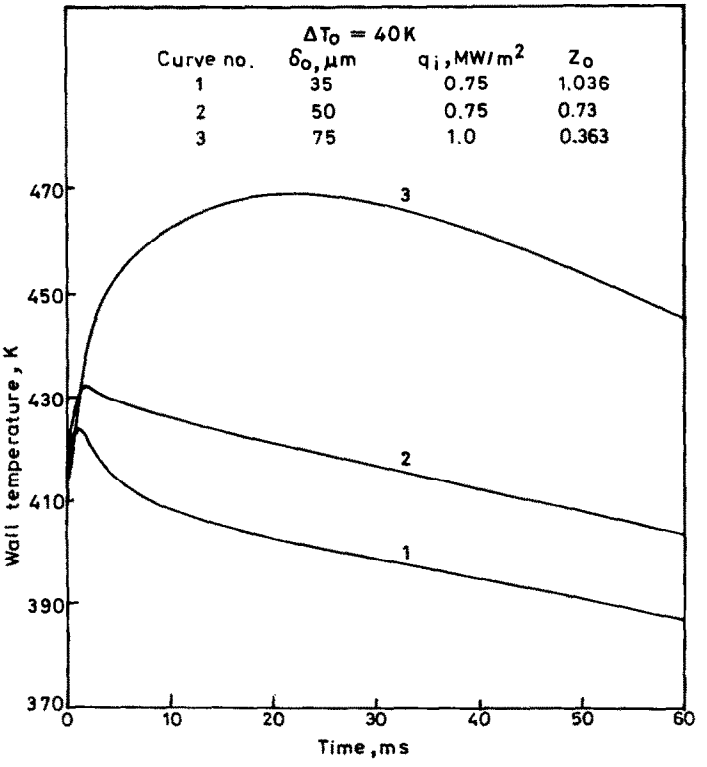

FIG. 5. Variation of wall temperature.

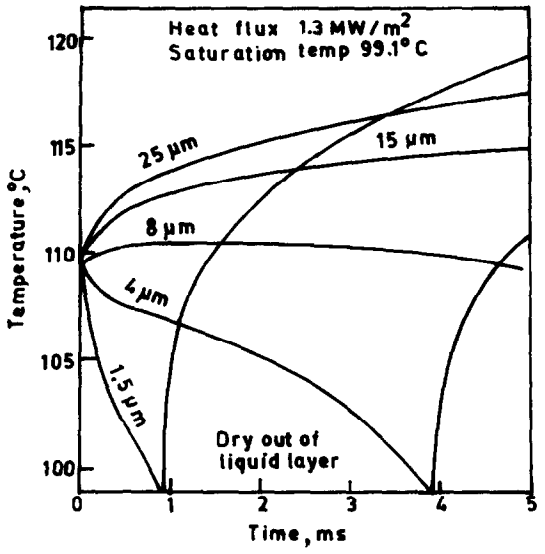

FIG. 6. Calculated surface temperatures beneath an evaporating film [4].

liquids and for a given value of the wall heat flux (q_i), initial macrolayer thickness (δ_0) and initial wall superheat (ΔT_0), the temperature gradient at the liquid–vapour interface has been obtained by a four-point formula [12] and then the instantaneous heat flux at the liquid–vapour interface is obtained. This value of instantaneous heat flux evaporates a small portion of the liquid macrolayer during the time step Δt. The rate of change of macrolayer thickness with time is given by the relation

$$\frac{\partial \delta}{\partial t} = -\frac{q_{lv}}{\rho_l h_{fg}} \tag{6}$$

where

$$q_{lv} = -k_l \left(\frac{\partial T}{\partial y}\right)_{y=\delta}.$$

The algorithm

$$\delta_{i+1} = \delta_i - \frac{(q_{lv})_i \Delta t}{\rho_l h_{fg}}$$

can be used to calculate the new value of δ, where δ_i and δ_{i+1} are the macrolayer thickness at the ith and $(i+1)$th time step, respectively.

The process is then repeated with new values of macrolayer thickness and surface temperature for subsequent time steps. The average conductive heat flux may be obtained by numerical integration—Simpson's rule based on the equation

$$\bar{q}_{lv} = \frac{1}{t}\int_0^t q_{lv}\, dt. \tag{7}$$

The numerical method described above was applied to obtain the temperature profile and the average conduction heat flow rate through the macrolayer for the wide range of boiling conditions for water and methanol. Table 1 gives the range of boiling conditions used.

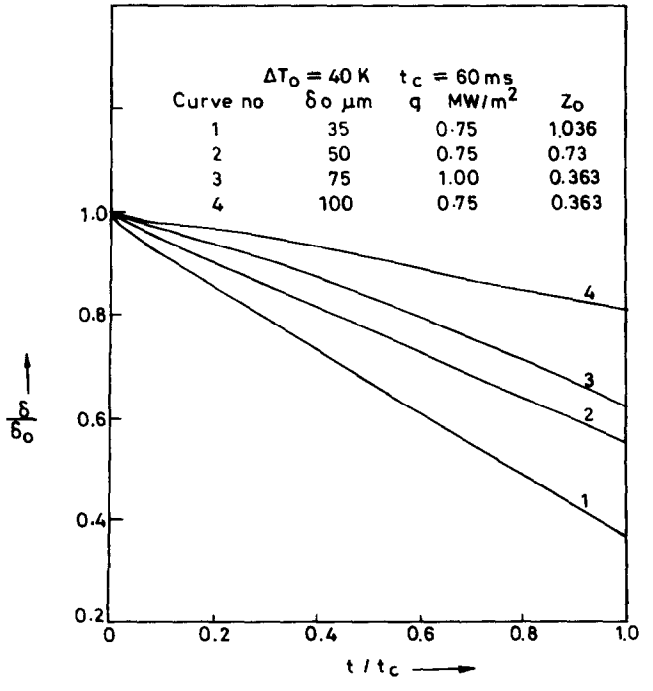

FIG. 7. Evaporation rate of macrolayer.

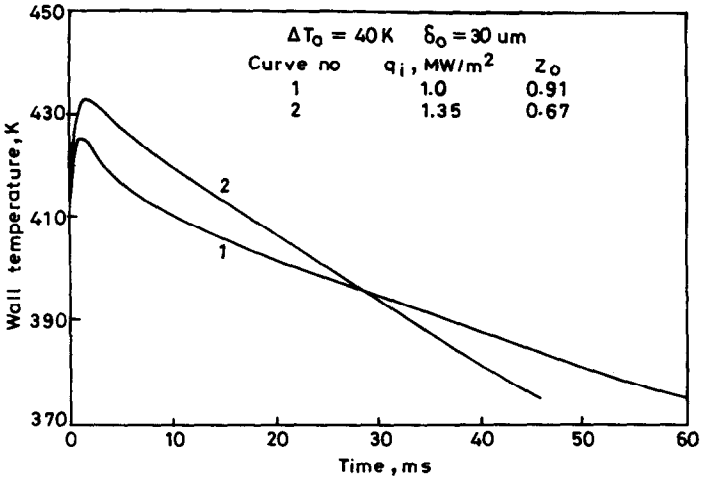

FIG. 8. Wall temperature behaviour for thin macrolayer.

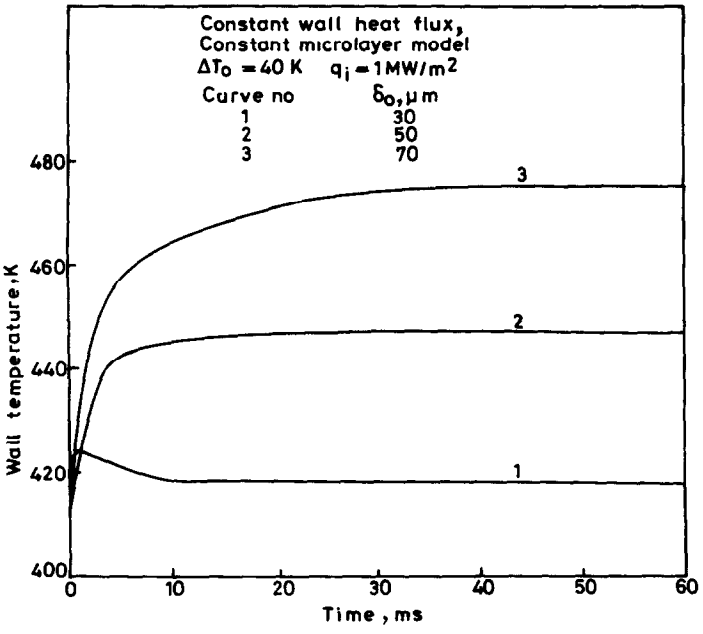

FIG. 9. Effect of initial macrolayer thickness on wall temperature.

Table 2. Average liquid–vapour interface conduction heat flux ($MW\,m^{-2}$): $T_0 = 40$ K, $q_i = 0.75\ MW\,m^{-2}$

Initial macrolayer thickness (μm)	Constant wall heat flux model		Constant wall temperature model	
	Constant macrolayer	Variable thickness macrolayer	Constant macrolayer	Variable thickness macrolayer
100	0.644	0.698	0.345	0.359
35	0.783	0.794	0.791	1.24

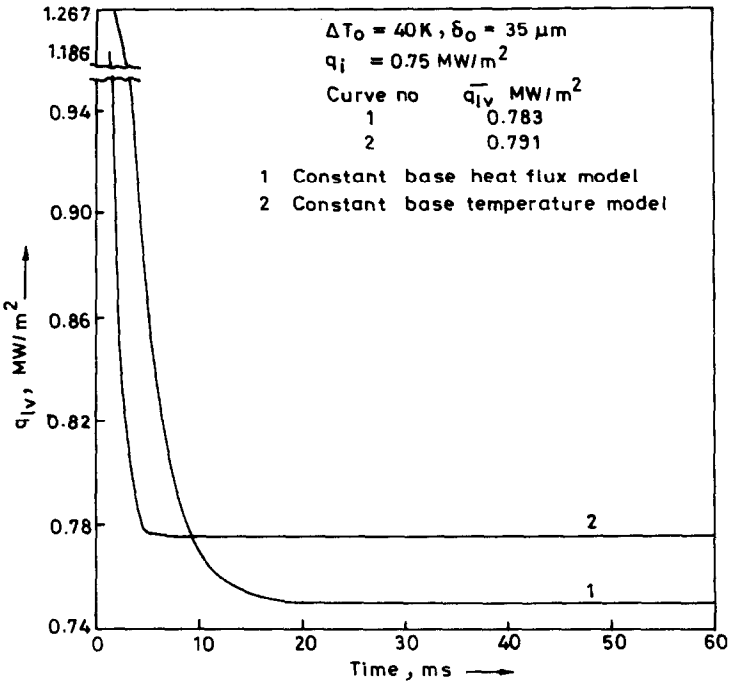

FIG. 10. Comparison of heat flux models.

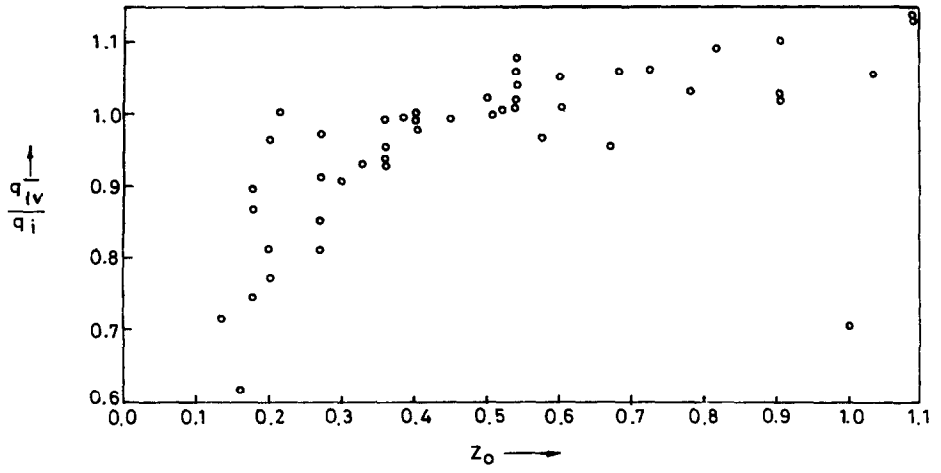

FIG. 11. Dependence of $\bar{q}_{lv}/q_i$ on Z_0.

RESULTS AND DISCUSSION

Non-dimensional parameter Z

Three interdependent boiling parameters namely, impressed heat flux q_i, macrolayer thickness δ, and wall superheat ΔT, involved in the analysis were proposed to be suitably combined into a non-dimensional parameter Z [15], which could be interpreted as the ratio of steady-state conductive heat flux ($k_l \Delta T/\delta$) to impressed heat flux (q_i), a higher value of Z representing a higher conductive contribution. In transient conditions, if initially $q_i > k_l \Delta T_0/\delta_0$ (here ΔT_0 represents initial wall superheat), i.e. the initial value of Z (call it Z_0) < 1, there is a tendency of increase of wall temperature T_w, thereby increasing ΔT accompanied by all the time a decrease in the value of the macrolayer thickness δ resulting in an increase of the term $k_l \Delta T/\delta$, which in turn increases Z and so the conditions are conducive for Z to approach a value of unity. On the other hand, if $q_i < k_l \Delta T_0/\delta_0$, i.e. the initial value $Z_0 > 1$, ΔT will tend to decrease accompanied by a decrease in δ, but the rate of decrease of δ being relatively less resulting in the reduction of the value of Z. In both these cases the

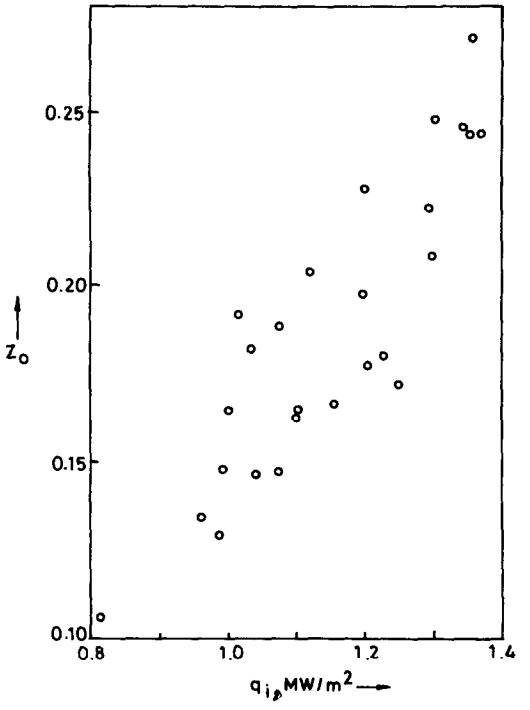

FIG. 12. Plot of q_i vs Z_0 for experimental values.

parameter Z tends to approach a value of unity as time elapses. In boiling, as will be discussed later in detail, Z_0 increases with increasing input heat flux and so does the contribution by heat conduction through the macrolayer.

Liquid–vapour interface heat flow and macrolayer

Figures 2 and 3 show the typical temperature profiles in the macrolayer and the resultant liquid–vapour heat flow rate (q_{lv}), respectively, for constant wall heat flux at the solid wall, moving macrolayer model. It can be observed that:

(1) There are sharp changes in the temperature gradient in the vicinity of both the liquid–vapour and the solid–liquid interfaces; at the liquid–vapour interface because of spontaneous temperature change from T_w to T_s at $t = 0$ and at the solid–liquid interface because of continuous energy input (impressed heat flux). In the mid region a slow change of temperature gradient can be clearly observed.

(2) The temperature profile tends to linearize as the time elapses with fixed temperature (T_s) at the top and increasing wall temperature at the bottom. The solid–liquid interface temperature might even decrease for initially thinner macrolayers. A study of temperature profiles showed that the trend depends upon the initial value of Z_0 and its rate of change during the heat flow process. The linear profile that is obtained near the termination of a cycle period is representative of the magnitude of the heat flux, δ and ΔT values that ultimately adjust to make parameter Z approach a value of unity which thereby indicates that q_{lv}/q_i approaches unity as can be observed from Figs. 3 and 4.

(3) A strong influence of the initial thickness (δ_0) of the macrolayer on the solid–liquid interfacial temperature for a given value of the impressed heat flux and initial wall superheat is seen in Fig. 5; wall temperature increasing with an increase in initial macrolayer thickness (or decreasing Z_0). After attaining a maximum value, the surface is seen to cool down and the minimum surface temperature attained is found to be a function of initial macrolayer thickness; decreasing when its value decreases. The effect of impressed heat flux and initial wall superheat on surface temperature was also investigated, the effect of initial wall superheat was found to be less pronounced than those of the initial macrolayer and input heat flux.

Similar solid–liquid interface temperature (T_w) fluctuations were predicted by Yu and Mesler [4] using a one-dimensional heat flow model that assumed a linear temperature distribution. Figure 6 shows the results obtained from the above model. In the present discussion, post dryout wall temperatures have not been discussed in view of the very little contribution to heat flow during this period.

In view of the large increase in heated wall temperature in certain cases, one is tempted to anticipate a corresponding increase in active site density thereby disturbing the quiescence of the macrolayer. However, this is unlikely in view of the findings of Gaertner [5] who showed that the maximum area covered by vapour stems is about 11% of the total heated surface area.

(4) In the region close to the critical heat flux when the initial macrolayer thickness is small, the ratio q_{lv}/q_i approaches a value of unity rather faster because of an apparently higher contribution to the conductive heat flow through a thinner layer (Fig. 3). Consequently a higher rate of thinning down of an initially thin macrolayer is seen in Fig. 7 where the slope of δ/δ_0 vs t/t_c curves is higher for such macrolayers for a given initial wall superheat and input heat flux. In such cases, the rise in heated surface temperature is found to be negligibly small (Fig. 8), indicating cooling of the surface most of the time. If the macrolayer is assumed to have invariable thickness, a steady state reached with wall temperatures (T_w) remaining nearly constant (Fig. 9) indicating that this case is analogous to the constant wall temperature heat flow model, i.e. for very thin macrolayers a constant base temperature model and a constant base heat flux model should produce the same results; this in fact is so, as seen in Fig. 10 which depicts typical plots of the liquid–vapour interface conduction heat flux with time. The time averaged values of heat flux over a cycle period are respectively 0.791 and 0.783 MW m^{-2}.

(5) Table 2 compares the predicted values of time averaged conduction heat flux for various conduction

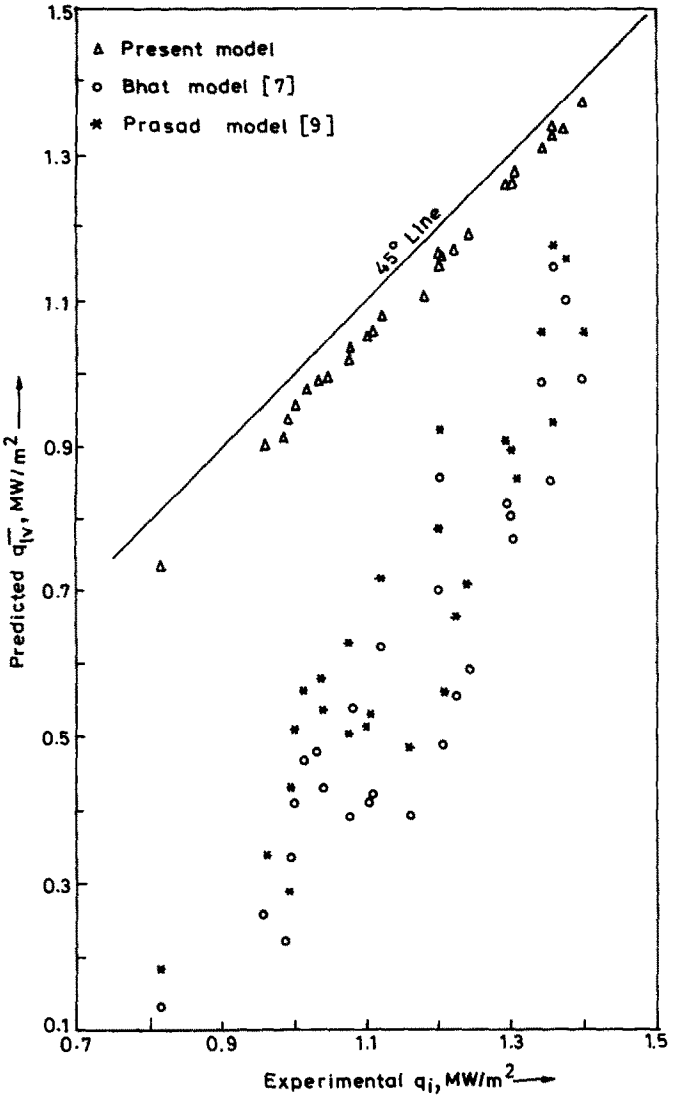

FIG. 13. Comparison of analytical and experimental results for water.

heat flow models. For initially thick macrolayers, the rate of consumption of the macrolayer is very low (Fig. 7) indicating that an assumption of constant macrolayer thickness will be reasonable under these circumstances. This observation is substantiated by the negligible difference between the heat flux values for the models with constant and variable thickness macrolayers both with constant wall temperature and with constant wall heat flux, although the constant base temperature models predict a value of average heat flux very much on the lower side, whereas the prediction with the constant heat flux model is reasonably good. On the other hand, for initially very thin macrolayers the rate of consumption of the macrolayer being high, the system cannot be approximated by constant macrolayer models though the predicted conduction heat flux values with all the models are comparable with impressed heat flux except constant base temperature (variable thickness macrolayer) model which predicts a very high value. The latter may be due to an unrealistic boundary condition of constant wall temperature because in reality with such thinner macrolayers, the wall temperature experiences a considerable dip on the lower side. It can therefore be concluded that the assumption of a constant wall heat flux for the conduction heat flow model, in general, predicts time averaged conduction heat flux values that are in very good agreement with impressed heat flux.

Combined effect of physical parameters (δ_0, ΔT_0, q_i)

It has been demonstrated with reasonable success that the effect of important boiling parameters namely, impressed heat flux, initial macrolayer thickness and initial wall superheat on the vapour–liquid interface heat flux (q_{lv}) can be discussed with reference

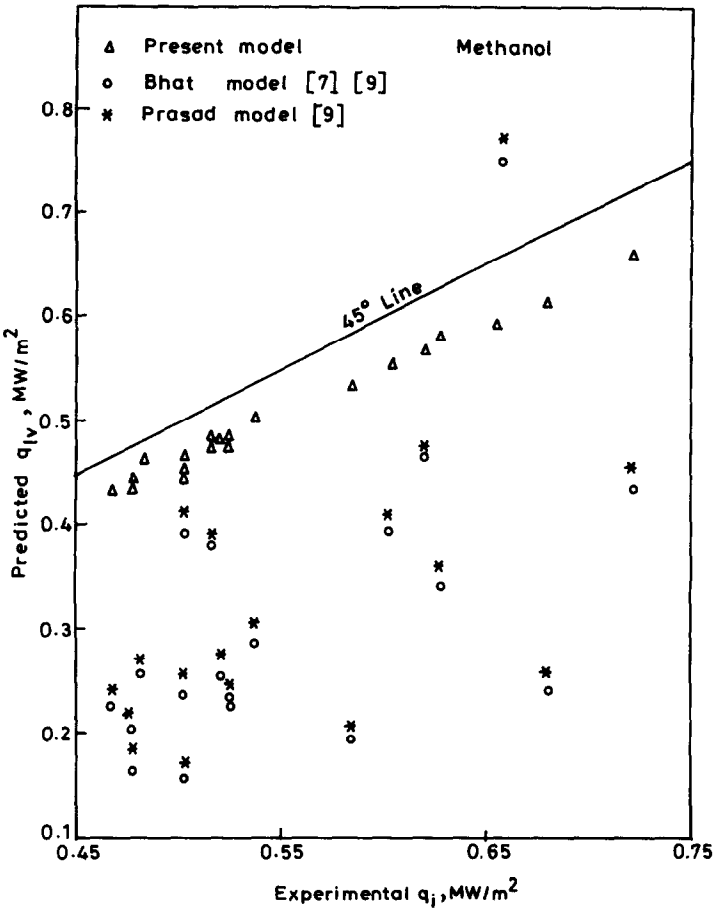

FIG. 14. Comparison of analytical and experimental results for methanol.

to the non-dimensional parameter Z_0. Figure 11 shows a plot of predicted values of $\bar{q}_{\ell v}/q_i$ vs Z_0 where the values of Z_0 have been obtained from the various combinations of δ_0, ΔT_0, and q_i given in Table 1. Corresponding average values of $\bar{q}_{\ell v}$ have been calculated for a 60 ms cycle period [7]. The values of the physical parameters chosen are arbitrary and have been selected so as to cover a wide range of boiling conditions.

It is seen from Fig. 11 that as the value of Z_0 increases, the ratio of average vapour–liquid interface heat flux $\bar{q}_{\ell v}$ to impressed heat flux q_i also increases. Figure 12 depicts the plot of experimental values of q_i vs Z_0 [16]. A monotonous increase in the value of Z_0 with increasing value of input heat flux q_i implies that for a given increase in q_i, the rate of increase of ΔT_0 is less compared to the rate of decrease in δ_0, so that ultimately the value of Z_0 increases. Such an increase in Z_0 also indicates that as input heat flux increases, the conditions are more conducive for a higher conductive contribution to heat flow as has been observed in Fig. 11 where in the regions of high Z_0 values, the $\bar{q}_{\ell v}/q_i$ ratio tends to attain a value of unity. Similar observations have been reported by previous investigators [1–3].

Comparison with experimental results

Using experimental data for q_i, δ_0, ΔT_0 and vapour–mass frequency from refs. [9, 16], $\bar{q}_{\ell v}$ has been calculated for a complete cycle period of vapour mass for methanol and water under atmospheric conditions. The predicted values of time averaged values of conduction heat flux have been compared with experimental values of input heat flux in Figs. 13 and 14 for water and methanol, respectively. The heat flux predicted by the heat flow models of Bhat *et al.* [7] and Prasad [9] are also shown in the figures. It can be seen that the present model predicts heat flux values which are in very good agreement (the average deviations being -4.2 and -8.3% for water and methanol, respectively) with the experimental values indicating the superiority of the present model. It also substantiates the postulate that the heat conduction across the macrolayer constitutes the major mode of heat flow from the heated wall to the boiling fluid in the macrolayer regime of pool boiling.

CONCLUSIONS

The following conclusions can be drawn from the above investigation:

(1) A heat transfer model assuming constant wall heat flux into the macrolayer predicts the average conduction heat flow rates at the liquid–vapour interface that are in excellent agreement with experimental values of input heat flux.

(2) Heat conduction across the liquid macrolayer constitutes the major portion of the heat transfer from the heated surface in the high heat flux region.

(3) The non-dimensional parameter Z_0 may be used as an index of the combined effect of the physical parameters (δ_0, ΔT_0 and q_i) on the conduction heat flux across the liquid layer.

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APPENDIX. REASON FOR DISAGREEMENT WITH THE ANALYSIS OF CHYU WITH REGARD TO THE ROLE OF HEAT CONDUCTION IN THE MACROLAYER

It has been reported [17] that for water near burnout under atmospheric conditions, the area occupied by the vapour stems is about one-ninth of the total surface area. As per the hypothesis of Chyu [10], the total amount of heat transfer, i.e. $q_i \times$ Area, must take place through the area occupied by the stems. This leads to the conclusion that the rate of heat transfer through the area occupied by stems must be roughly nine times the average wall heat flux. In order to support such a high value of heat flux under assumed steady-state conditions the microlayer thickness should be of the order of δ_0 given by

$$\frac{K_l \Delta T}{\delta_0} = 9q_i = q'. \tag{A1}$$

The resulting average rate of change of microlayer thickness $d\delta/dt$ should then be given by

$$\frac{d\delta}{dt} = \frac{q'}{\rho_l h_{lg}}.$$

This gives dryout time t_c as

$$t_c = \delta_0 \rho_l h_{lg}/q'. \tag{A2}$$

Using values of $\Delta T_0 = 40$ K and $q_i = 1.5$ MW m^{-2} for the conduction heat transfer model assuming a constant temperature heated surface, for water at atmospheric pressure. Equations (A1) and (A2) give a dryout time of 0.32 ms. This will require a bubble frequency, the minimum value of which should be $1/(0.32 \times 10^{-3})$, i.e. 3125 Hz.

In the high heat flux region, experimental values of frequency available in the literature are much lower than those anticipated above. For water at atmospheric pressure Bhat *et al.* [8] and Prasad [9] reported maximum values of 400 and 450 Hz, respectively.

A similar analysis for methanol at atmospheric pressure assuming an area ratio of 1/9 and critical heat flux of 0.72 MW m^{-2}, gives frequency of bubble generation as 6600 Hz.

Perkins and Westwater [18] have reported a maximum value of frequency of only 60 Hz whereas Prasad [9] reported frequency values which are lower than 252 Hz.

Haramura and Katto [19] obtained for water at atmospheric pressure, a vapour stem area ratio of about 0.013. This requires a still higher value of frequency, if Chyu's hypothesis is to hold good.

UN NOUVEAU MODELE DE TRANSFERT THERMIQUE A TRAVERS LA MACROCOUCHE DANS L'EBULLITION A TRES GRAND FLUX EN RESERVOIR

Résumé—On résout numériquement un modèle analytique de transfert thermique pour un transfert thermique variable monodimensionnel à travers une macrocouche d'épaisseur décroissante et avec un flux thermique imposé constant à l'interface entre le solide chauffé et la macrocouche. Le modèle prédit la densité de flux thermique de conduction à l'interface liquide–vapeur pour l'eau et le méthanol de façon très conforme aux données expérimentales. On montre qu'un paramètre Z_0 adimensionnel peut être utilisé comme un indice de l'effet combiné sur le flux thermique de conduction à travers la couche liquide, des paramètres physiques tels que l'épaisseur initiale de la macrocouche, la surchauffe initiale de la paroi et le flux thermique imposé.

EIN NEUES MODELL FÜR DEN WÄRMETRANSPORT DURCH DIE MAKROSCHICHT BEIM BEHÄLTERSIEDEN UND BEI GROSSER WÄRMESTROMDICHTE

Zusammenfassung—Ein analytisches Modell für den transienten eindimensionalen Wärmetransport durch eine Makroschicht abnehmender Dicke bei konstanter aufgeprägter Wärmestromdichte an der festen Begrenzungsfläche wird formuliert und numerisch gelöst. Mit Hilfe dieses Modells wird der mittlere Wärmestrom durch Wärmeleitung an der Grenzfläche zwischen Flüssigkeit und Dampf berechnet. Die Ergebnisse für Wasser und Methanol stimmen hervorragend mit Versuchsergebnissen überein. Es zeigt sich, daß ein dimensionsloser Parameter Z_0 zur Beschreibung der kombinierten Einflüsse der physikalischen Parameter verwendet werden kann. Diese sind die anfängliche Dicke der Makroschicht, die anfängliche Wandüberhitzung, die aufgeprägte Wärmestromdichte und die Wärmestromdichte bei der Wärmeleitung durch die Flüssigkeitsschicht.

НОВАЯ МОДЕЛЬ ПЕРЕНОСА ТЕПЛА ЧЕРЕЗ МАКРОСЛОЙ В УСЛОВИЯХ КИПЕНИЯ В БОЛЬШОМ ОБЪЕМЕ ПРИ БОЛЬШИХ ТЕПЛОВЫХ НАГРУЗКАХ

Аннотация—Численно решается аналитическое модельное уравнение теплопереноса при нестационарном одномерном тепловом потоке через сужающийся макрослой и постоянной подводимой тепловой нагрузке у нагреваемой границы раздела твердое тело–макрослой. Модель позволяет рассчитывать средний расход тепла теплопроводностью у границы раздела жидкость–пар для воды и метанола, и полученные результаты очень хорошо согласуются с экспериментальными значениями подводимого теплового потока. Показано, что безразмерный параметр Z_0 может использоваться в качестве критерия подобия совместного влияния таких физических параметров, как начальная толщина макрослоя, начальный перегрев стенки и подводимая тепловая нагрузка, на кондуктивный тепловой поток через слой жидкости.