Bubble Dynamics and Nucleate Pool Boiling Heat Transfer - Numerical Simulations and Experimental Validation.*

V.K. Dhir, H.S. Abarajith, G. Warrier

Mechanical and Aerospace Engg. Dept., University of California, Los Angeles, USA.

G. Son, Sogang University, Seoul, S. Korea.

ABSTRACT

Results from numerical simulation and experimental validation of the growth and departure of single and multiple merging bubbles and the associated heat transfer on a horizontal heated surface during pool nucleate boiling under low and earth normal gravity conditions have been reviewed here. A finite difference scheme was used to solve the equations governing mass, momentum and energy in the vapor and liquid phases. The vapor-liquid interface is captured by a level set function while including the influence of phase change at the liquid-vapor interface. Water and PF5060 were used as test liquids. The effects of gravity levels, contact angle, liquid subcooling and wall superheat on the bubble diameter, interfacial structure, bubble merger time, departure time and local heat fluxes are reported. A scheme to calculate the wall heat flux for a large surface is also introduced.

INTRODUCTION

Boiling, being the most efficient mode of heat transfer is employed in various energy conversion systems and component cooling devices. In the past, boiling has been studied extensively. The process allows accommodation of high heat fluxes at low wall

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superheats and at the same time the process is very complex and its understanding imposes severe challenges. Prior to incipient boiling, at low heat fluxes, the heat transfer is controlled by natural convection. At higher heat fluxes, the heat transfer is controlled by bubble dynamics. Initially during partial nucleate boiling, discrete bubbles form on the heater surface. At moderately higher heat fluxes, bubbles start merging laterally as well as vertically. The transition from partial nucleate boiling to fully developed nucleate boiling is characterized by the lateral and vertical merger of bubbles.

Owing to the effectiveness of boiling as a heat transfer process in space, boiling heat transfer based systems can offer significant advantages. Applications of boiling heat transfer in space applications include thermal management, fluid handling, and control and power systems. The key factors that are to be addressed for the space systems based on Rankine cycle are the boiling heat transfer coefficients and the critical heat flux under reduced gravity conditions.

Siegal and Keshock (1964) observed that the bubbles grow larger and show longer growth periods before detaching from the heater surface under reduced gravity conditions. Merte (1994) and Lee and Merte (1997) have reported the results of pool boiling experiments conducted in the space shuttle for a surface similar to that used in drop tower tests. The subcooled boiling was found to be unstable during long periods of micro gravity conditions. It was concluded that the subcooling has negligible influence on the steady state heat transfer coefficient.

Straub et al. (1992) and Straub (1994) conducted a series of nucleate boiling experiments using thin platinum wires and gold coated flat plate as heaters at low gravity conditions in the flights of ballistic rockets and in KC-135 aircraft. For the flat plate
heater with R12 as the test liquid, boiling curves similar to those at normal gravity were obtained. Using R113 as the test liquid, rapid bubble growth and large bubbles were observed. However, neither the bubble growth rate nor the bubble diameter at the departure was given. For subcooled boiling under microgravity conditions, they observed a reduction of up to 50% in heat transfer coefficient in comparison with the normal gravity cases.

Qiu et al. (2000a, 2000b) conducted experimental studies on the growth and detachment mechanisms of a single bubble and multiple bubble merger during the parabolic flights of the KC-135 aircraft. Experiments were carried out under normal and reduced gravity conditions for various wall superheats and liquid subcoolings at system pressure varying from 1 bar to 1.13 bar. Artificial cavities were made on a polished silicon wafer. The wafer was heated on the backside in order to control the wall superheat. Bubbles were produced on artificially etched cavities at the middle of the wafer in degassed and distilled water. The gravity in the experiments was not always constant but varied with time.

The experimental investigations of boiling heat transfer for space applications impose serious restrictions in terms of the duration of the experiments, maintaining the gravity conditions, number of experiments that can be conducted, size of the experimental apparatus required etc.. Thus numerical simulations can play an important role in predicting the boiling heat fluxes in microgravity conditions.

Several attempts have been made in the past to model bubble growth and bubble departure processes on a heated wall. Lee and Nydahl (1989) calculated the bubble growth rate by solving the flow and temperature fields numerically. They used the
formulation of Cooper and Lloyd (1969) for micro layer thickness. However they assumed a hemispherical bubble and wedge shaped microlayer and thus they could not account for the shape change of the bubble during growth.

Zeng et al. (1993) used a force balance approach to predict the bubble diameter at departure. They included the surface tension, inertial force, buoyancy and the lift force created by the wake of the previously departed bubble. But there was empiricism involved in computing the inertial and drag forces. The study assumed a power law profile for growth rate with the proportionality constant exponent determined from the experiments.

Mei et al. (1995) studied the bubble growth and departure time assuming a wedge shaped microlayer. They also assumed that the heat transfer to the bubble was only through the microlayer, which is not totally correct for both subcooled and the saturated boiling. The study did not consider the hydrodynamics of the liquid motion induced by the growing bubble and introduced empiricism through the shape of the growing bubble. Welch (1998) has studied bubble growth using a finite volume method and an interface tracking method. The conduction in the solid wall was also taken into account. However, the microlayer was not modeled explicitly.

Son et al. (1999) numerically simulated the single bubble growth during nucleate boiling using the Level Set method. This method had been previously applied to adiabatic incompressible two-phase flow by Sussman et al. (1994) and to film boiling near critical pressures by Son and Dhir (1998). Singh and Dhir (2000) have obtained numerical results for low gravity conditions by exercising the numerical simulation model of Son et al. (1999), when the liquid is subcooled. The computational domain was divided into two
regions viz. micro and macro regions. The interface shape and velocity and temperature field in the liquid in the macro region were obtained by solving the conservation equations. For the micro region, lubrication theory was used, which included the disjoining pressure in the thin liquid film. The interface shapes obtained from the micro region and macro region were matched at the outer edge of the micro layer.

Abarajith and Dhir (2002) studied the effects of contact angle on the growth and departure of a single bubble on a horizontal heated surface during pool boiling under normal gravity conditions. The contact angle was varied by changing the Hamaker constant that defines the long-range forces. They also studied the effect of contact angle on the microlayer and macrolayer heat transfer rates. It was shown that the predicted diameter at departure normalized with the corresponding values for a contact angle of 90° for water and PF5060 when plotted against contact angle, fell on the same curve.

Son et al. (2002) studied vertical merger of bubbles at a single nucleation site for various wall superheats and waiting times. Abarajith et al. (2003) studied effects of wall superheat and liquid subcooling on the bubble departure diameter and growth time during nucleate pool boiling on a horizontal surface for both earth normal and low gravity conditions. Mukherjee and Dhir (2003) studied the effects of orientation of cavities on the bubble departure diameter and growth time. Abarajith et al. (2003) studied the dynamics of multiple bubble mergers in horizontal plane during pool nucleate boiling. They quantified the “lift force” that causes the merged bubble to lift-off earlier.
NUMERICAL MODEL

Mathematical Development of the Model

The two-dimensional model of Son et al. (1999) is extended to three dimensions to study bubble merger under low gravity conditions. The computational domain is divided into two regions viz. micro region and macro region as shown in Fig.1. The micro region is a thin film that lies between the solid wall and bubble, whereas the macro region consists of the bubble and its surrounding. Both the regions are coupled through matching of the shape at the outer edge of the micro layer and are solved simultaneously. The assumptions made in the development of the model are:

1) The process is three-dimensional.
2) The flows are laminar.
3) The wall temperature remains constant throughout the process.
4) Pure PF5060 and water at atmospheric pressure are used as the test fluid.
5) The thermodynamic properties of the individual phases are assumed to be insensitive to the small changes in temperature and pressure. The assumption of constant property is reasonable as the computations are performed for low wall superheat range.
6) Static contact angle is assumed to be known. Variations of contact angle during advancing and receding phases of the interface are not included.

Micro Region

A two-dimensional quasi-static model is used for the micro region and no azimuthal variations are considered. As such, the solution for the microlayer thickness is obtained in the radial direction from the center of the bubble base. This solution is assumed to be
valid for all the azimuthal positions. This assumption is still valid during the merger process when the bubble shapes are not symmetrical along the azimuthal positions because the length of the microlayer is assumed constant (though varies with contact angle) throughout the bubble growth.

The equation of mass conservation in micro region is written as,

\[
\frac{q}{h_{fg}} = -\frac{1}{r} \frac{\partial}{\partial r} \int_0^\delta \rho_r u dy
\]

where \( q \) is the conductive heat flux from the interface, defined as \( \frac{k_f(T_{wall} - T_{int})}{\delta} \) with \( \delta \) as the thickness of the thin film.

Lubrication theory has been assumed in a manner similar to that in the earlier works by Stephan and Hammer (1994), Lay and Dhir (1995) and Wayner (1999). According to the lubrication theory, the momentum equation in the micro region is written as,

\[
\frac{\partial p_l}{\partial r} = \mu \frac{\partial^2 u}{\partial y^2}
\]

where \( p_l \) is the pressure in the liquid. Heat conducted through the thin film must match that due to evaporation from the vapor-liquid interface. By using modified Clausius-Clapeyron equation, the energy conservation equation for the micro region yields,

\[
\frac{\kappa_f(T_{wall} - T_{int})}{\delta} = h_{ev} \left[ T_{int} - T_v + \frac{(p_l - p_v)T_v}{\rho_l h_{fg}} \right]
\]

The evaporative heat transfer coefficient is obtained from kinetic theory as,
\[ h_{\text{v}} = 2 \left[ \frac{M}{2\pi R T_v} \right]^{1/2} \frac{\rho_{\text{v}} h_{\text{fg}}^2}{T_v} \quad \text{and} \quad T_v = T_v(p_v) \] (4)

The pressure in the vapor and liquid phases at the interface are related by,

\[ p_i = p_v - \sigma K - \frac{A_0}{\delta^3} + \frac{q^2}{2 \rho_{\text{v}} h_{\text{fg}}^2} \] (5)

where \( A_0 \) is the dispersion constant. The second term on the right-hand side of equation (5) accounts for the capillary pressure caused by the curvature of the interface, the third term is for the disjoining pressure, and the last term originates from the recoil pressure.

The curvature of the interface is defined as,

\[ K = \frac{1}{r} \frac{\partial}{\partial r} \left[ r \frac{\partial \delta_f}{\partial r} \sqrt{1 + \left( \frac{\partial \delta_f}{\partial r} \right)^2} \right] \] (6)

The combination of the mass conservation, equation (1), momentum conservation, equation (2), mass balance and energy conservation, equation (3) and pressure balance equation (5) along with equation (6) for the curvature for the micro-region yields a set of three nonlinear first order ordinary differential equations (7),(8) and (9),

\[ \frac{\partial \delta_f}{\partial r} = -\frac{\delta_f(1+\delta_f^2)}{\sigma} + \frac{(1+\delta_f^2)^{3/2}}{r} \]

\[ \times \left[ \frac{\rho_{\text{v}} h_{\text{fg}}}{T_v} \left( T_{\text{int}} - T_v - \frac{q}{h_{\text{v}}} \right) - \frac{A_0}{\delta^3} + \frac{q^2}{2 \rho_{\text{v}} h_{\text{fg}}^2} \right] \] (7)

\[ \frac{\partial T_{\text{int}}}{\partial r} = -\frac{q \delta_f}{\kappa_f + h_{\text{v}} \delta_f} + \frac{3T_v h_{\text{v}} \mu_f \Gamma}{(\kappa_f + h_{\text{v}} \delta_f) h_{\text{fg}}^2 \delta_f^2} \] (8)

\[ \frac{\partial \Gamma}{\partial r} = -\frac{r q}{h_{\text{fg}}} \] (9)

where \( \Gamma = -\frac{r \delta_f^3 \rho_f \partial \rho_f}{3 \mu_f \partial r} \), is the mass flow rate in the thin film.
The above three differential equations can be simultaneously integrated by using a Runge-Kutta method, when boundary conditions at \( r = R_0 \) are given. In present case, the interface shape obtained from micro and macro solutions is matched at radial location \( R_1 \). The radius of dry region beneath a bubble, \( R_0 \), is related to \( R_1 \) from the definition of the apparent contact angle, \( \tan \varphi = 0.5h/(R_1 - R_0) \).

The boundary conditions for film thickness at the end points are:

\[
\delta = \delta_0, \quad \delta_r = 0, \quad \Gamma = 0 \quad \text{at} \quad r = R_0 \\
\delta = h/2, \quad \delta_r = 0 \quad \text{at} \quad r = R_1
\]

(10)

where, \( \delta_0 \) is the interline film thickness at the inner edge of microlayer \( r = R_0 \), which is calculated by combining equations (3) and (4) and requiring that \( T_{\text{int}} = T_{\text{wall}} \) at \( r = R_0 \) and \( h \) is the spacing of the three dimensional grid for the macroregion. For a given \( T_{\text{int}} \) at \( r = R_0 \), a unique vapor-liquid interface is obtained. The static contact angle, \( \varphi \), for PF5060-silicon system based on measurements was taken to be \( 10^\circ \) and that for water-silicon system is taken to be \( 54^\circ \).

**Macro Region**

For numerically analyzing the macro region, the level set formulation developed by Son et al. (1999) for nucleate boiling of pure liquid is used. The interface separating the two phases is captured by distance function, \( \phi \), which is defined as a signed distance from the interface. The negative sign is chosen for the vapor phase and the positive sign for the liquid phase. The discontinuous pressure drop across vapor and liquid caused by surface tension force is smoothed into a numerically continuous function with a \( \delta \)-function formulation (refer to Sussman et al., 1994, for details). The continuity,
momentum and energy, conservation equations for the vapor and liquid in the macro region are written as,

\[ \rho_t + \nabla \cdot (\rho \vec{u}) = 0 \]  

(11)

\[ \rho (\vec{u}, + \vec{u} \cdot \nabla \vec{u}) = -\nabla p + \nabla \cdot \mu \nabla \vec{u} + \nabla \cdot \mu \nabla \vec{u}^2 \]

(12)

\[ + \rho \vec{g} - \rho \beta_T (T - T_s) \vec{g}(t) - \sigma K \nabla H \]

\[ \rho c_p (T, + \vec{u} \cdot \nabla T) = \nabla \cdot \kappa \nabla T \text{ for } H > 0 \]  

(13)

\[ T = T_s (p_v) \text{ for } H = 0 \]

(14)

The fluid density, viscosity and thermal conductivity are defined in terms of the step function, \( H \), as,

\[ \rho = \rho_v + (\rho_l - \rho_v) H \]  

(15)

\[ \mu^{-1} = \mu_v^{-1} + (\mu_l^{-1} - \mu_v^{-1}) H \]  

(16)

\[ \kappa^{-1} = \kappa_l^{-1} H \]  

(17)

where, \( H \), is the Heaviside function, which is smoothed over three grid spaces in the interface region as described below,

\[ H = \begin{cases} 
1 & \text{if } \phi \geq 1.5h \\
0 & \text{if } \phi \leq 1.5h \\
0.5 + \frac{\phi}{3h} + \sin\left(\frac{2\pi \phi}{3h}\right)/(2\pi) & \text{if } |\phi| \leq 1.5h 
\end{cases} \]  

(18)

The mass conservation equation (11) can be rewritten as,

\[ \nabla \cdot \vec{u} = -\left( \rho_t + \vec{u} \cdot \nabla \rho \right)/\rho \]  

(19)
The term on right hand side of equation (19) is the volume expansion due to liquid-vapor phase change. From the conditions of the mass continuity and energy balance at the vapor-liquid interface, the following equations are obtained,

\[ \tilde{m} = \rho (\tilde{u}_{\text{int}} - \tilde{u}) = \rho_i (\tilde{u}_{\text{int}} - \tilde{u}_i) = \rho_v (\tilde{u}_{\text{int}} - \tilde{u}_v) \] (20)

\[ \tilde{m} = \kappa \nabla T / h_{fg} \] (21)

where \( \tilde{m} \) is the evaporation rate, and \( \tilde{u}_{\text{int}} \) is interface velocity. If the interface is assumed to advect in the same way as the level set function, the advection equation for density at the interface can be written as,

\[ \rho_i + \tilde{u}_{\text{int}} \cdot \nabla \rho = 0 \] (22)

Using equations (18), (20) and (21), the continuity equation, (19) for macro region is rewritten as,

\[ \nabla \cdot \tilde{u} = \frac{\tilde{m}}{\rho^2} \cdot \nabla \rho \] (23)

The vapor produced as a result of evaporation from the micro region is added to the vapor space through the cells adjacent to the heated wall, and is expressed as,

\[ \left( \frac{1}{V_c} \frac{dV}{dt} \right)_{\text{micro}} = \frac{\dot{m}_{\text{micro}}}{V_c \rho_v} \delta_c(\phi) \] (24)

where, \( V_c \) is the volume of the control volume and \( \dot{m}_{\text{micro}} \) is the evaporation rate from the microlayer and is expressed as,

\[ \dot{m}_{\text{micro}} = \int_{r_0}^{r_i} \frac{\kappa_i (T_{\text{wall}} - T_{\text{amb}})}{h_{fg} \delta} r dr \] (25)

The bubble expansion due to the vapor addition from micro layer is smoothed at the vapor-liquid interface by the smoothed delta function as done by Sussman et al. (1994)
\[ \delta_z(\phi) = \frac{\partial H}{\partial \phi} \]  

(26)

In the level set formulation, the level set function, \( \phi \), is used to keep track of the vapor-liquid interface location as the set of points where \( \phi = 0 \), and it is advanced by the interfacial velocity while solving the following equation,

\[ \phi_t = -\vec{u}_m \cdot \nabla \phi \]  

(27)

To keep the values of \( \phi \) close to that of a signed distance function, \(|\nabla \phi| = 1\), \( \phi \) is reinitialized after every time step as,

\[ \frac{\partial \phi}{\partial t} = -\frac{u_t \phi_0}{\sqrt{\phi_0^2 + h^2}} (1 - |\nabla \phi|) \]  

(28)

where, \( \phi_0 \) is a solution of equation (27) and \( u_t \) is the characteristic interface velocity, which is set to unity.

The boundary conditions for velocity, temperature and level set function for the governing equations (11)-(14) are (symmetry on four sides of computational domain, free surface on the top of the domain and heater surface at the bottom):

\[
\begin{align*}
    u &= v_x = w_x = 0, & T_x &= 0, & \phi_x &= 0, & \text{at } x &= 0 \\
    u &= v_y = w_y = 0, & T_y &= 0, & \phi_y &= 0, & \text{at } x &= X \\
    u &= v = w = 0, & T &= T_{\text{wall}}, & \phi_y &= -\cos \varphi, & \text{at } y &= 0 \\
    u_y &= w_y = v_y = 0, & T_y &= 0, & \phi_y &= 0, & \text{at } y &= Y \\
    u_z &= v_z = w_z = 0, & T_z &= 0, & \phi_z &= 0, & \text{at } z &= 0 \\
    u_z &= v_z = w_z = 0, & T_z &= 0, & \phi_z &= 0, & \text{at } z &= Z
\end{align*}
\]  

(29)

For the numerical calculations, the governing equations for micro and macro regions are nondimensionalized by defining the characteristic length scale, \( l_0 \), the characteristic velocity, \( u_0 \), and the characteristic time, \( t_0 \) as,
The characteristic length scales for water and PF5060 at 1.0ge are 2.5 mm and 0.734 mm respectively.

**Solution**

The governing equations are numerically integrated by following the procedure of Son et al. (1999).

1) The value of $A_0$, the Hamaker constant is guessed for a given contact angle.

2) The macro layer equations are then solved to determine the value of $R_1$ (radial location of the vapor-liquid interface at $\delta = h/2$.)

3) The micro layer equations are then solved with the guessed value of $A_0$ to determine the value of $R_0$ (radial location of the vapor-liquid interface at $\delta = \delta_0$).

4) The apparent contact angle is calculated using equation $\tan \varphi = 0.5h/(R_1 - R_0)$ and steps 1-4 are repeated for a different value of $A_0$, if the values of the given and the calculated apparent contact angles do not match.

The initial velocity is assumed to be zero everywhere in the domain. The initial fluid temperature profile is taken to be linear in the natural convection thermal boundary layer and the thermal boundary layer thickness, $\delta_T$, is evaluated using the correlation for the turbulent natural convection on a horizontal plate as,

$$\delta_T = 7.14(\nu \alpha / g \beta \Delta T)^{1/3}$$

The mesh size for all calculations is chosen as 96 grids per 0.5$l_0$. It represents the best trade-off in calculation accuracy and computing time as done in the earlier works by
Son et al. (1999) and Singh and Dhir (2000). The difference in the bubble departure diameters computed with 96 grids per 0.5\(l_0\) and 192 grids per 0.5\(l_0\) of characteristic length is about 3 \%. The calculations are carried out either for quarter of the domain or for half of the domain depending on the symmetric nature of the problem in the X and Z axes.

**EXPERIMENTS**

The experimental apparatus for pool boiling experiments is schematically shown in Fig. 2. The system configuration is the same as that of Qiu et al. (2000a, 2000b). It consists of a test chamber, a bellow and a nitrogen (N\(_2\)) chamber. Three glass windows are installed on the walls of the test chamber for visual observation. For the control of the system pressure, transducers are installed in the test chamber and N\(_2\) chamber, respectively. The test surface for studying nucleate boiling is installed at the bottom of the test chamber. In the vicinity of the heater surface a rake of six thermocouples are placed in the liquid pool to measure the temperature in the thermal boundary layer, while another rake is placed in the upper portion of the test chamber to measure the bulk liquid temperature. Distilled, filtered and degassed water and PF5060 are used as the test liquid. Two video cameras operating at 250 frames/second were installed at an angle of 90° to record the boiling processes. The liquid temperature and pressure in the chamber were controlled according to the set points established by the operator on board. A triaxial accelerometer is installed on the test rig.

A polished silicon wafer was used as the test surface for the nucleate boiling experiments. From the manufacturer’s specification the roughness of the bare polished wafer was less than 5 Å. The contact angle for the liquid-surface combination was
measured before the experiment by taking photographs of liquid droplets placed on the heater surface.

At the back of the silicon wafer strain gages were bonded as heating elements. In the central area, the miniature elements of size $2 \times 2 \text{ mm}^2$ are used and are grouped so as to cover small areas of the test surface. In each group a thermocouple is directly attached to the wafer. The heater surface temperatures in different regions are then separately controlled through a multi-channel feedback control system. As such, the wall superheat can be maintained constant during an experimental run and can be changed to any desired set point. The local heating rate in the individual areas is recorded during the experiments. The power lead wires and the thermocouple wires are led out from the hole in the base made from Phenolic Garolite Grade 10 (G-10). The wafer is cast with RTV on the G-10 base. The base in turn is mounted in the test chamber. Cylindrical cavities of predetermined sizes and at specified locations were etched in the wafer via the Deep Reactive Ion Etching technique (DRIE).

**RESULTS AND DISCUSSION**

**Single Bubble**

Figure 3(a) shows the variation of bubble departure diameter and bubble growth period with wall superheat during boiling of saturated water at one atmosphere pressure. Both the bubble diameter and bubble growth period increases with wall superheat. Also shown in Fig. 3 (a) is the experimental data of Qiu et al. (2000a, 2000b). Good agreement between the experimental and numerically predicted bubble departure diameters is observed, though the bubble growth time is slightly over predicted. Figure 3(b) shows the
variation of bubble departure diameter and bubble growth period with liquid subcooling, for water ($\Delta T_w = 8 \, ^\circ \text{C}$). Both the bubble departure diameter and bubble growth period increase with increasing liquid subcooling. The contribution of the various heat transfer mechanisms (micro layer, evaporation around the bubble boundary, and condensation) as a function of time are shown in Fig. 4, for boiling of subcooled water (Singh and Dhir (2000)). The condensation around the bubble is zero in the initial stages of bubble growth (up to 32 ms), when the bubble is still surrounded by superheated liquid. Once the bubble diameter becomes large enough to be surrounded by subcooled liquid, the condensation rate increases (shown in Fig. 4 as a negative value).

**Effect of Contact Angle**

Abarajith et al. (2002) studied the effects of contact angle on bubble departure diameter and growth time. The bubble growth history for two fluids with different contact angles (water and PF5060) are shown in Fig. 5. The results of numerical calculations are in good agreement with data. In general, the lower the contact angle, the smaller is the bubble departure diameter and the bubble growth time. The corresponding evaporative heat transfer rates from the micro layer and macro region are shown in Fig. 6. The micro layer evaporation rate increases with increasing contact angle because the bubble base area and interfacial area increases with increasing contact angle. A corresponding increase in the evaporation rate from the macro layer is also observed.

The contact angle was varied by changing the Hamaker constant that defines the long-range forces. From numerical simulations, it has been shown in the Fig. 7 that the departure diameter and time period of growth for water and PF5060 when plotted versus
the contact angle, fall on the same curve with the proper choice of non-dimensional parameter, in this case the values corresponding to a contact angle of 90°. The data obtained with water and PF5060 bear out the predictions. For water, the contact angle is varied by oxidizing the silicon wafer surface to different degrees.

**Effects of Gravity**

In Fig. 8(a), the ratio of the bubble diameter at departure normalized with that at $g_z/g_e = 1$ for water and PF5060 is plotted as a function of gravity level. It is found that bubble diameter at departure, $D_{ds}$, for water with a contact angle of 54° scales with gravity as

$$\frac{D_{d,s}}{D_{d,s(g_z/g_e=1)}} = \left(\frac{g_z}{g_e}\right)^{-0.5}$$

(32)

The bubble diameter at departure for PF5060 with a contact angle of 10°, $D_{d}$ scales as

$$\frac{D_{d,s}}{D_{d,s(g_z/g_e=1)}} = \left(\frac{g_z}{g_e}\right)^{-0.42}$$

(33)

In Fig. 8 (b), the single bubble growth periods normalized with those at earth normal gravity for both water and PF5060 are plotted, as a function of normalized gravity level. The time period of growth for water with a contact angle of 54° scales as,

$$\frac{t_{g,s}}{t_{g,s(g_z/g_e=1)}} = \left(\frac{g_z}{g_e}\right)^{-0.93}$$

(34)

whereas that for PF5060 with a contact angle of 10° scales as

$$\frac{t_{g,s}}{t_{g,s(g_z/g_e=1)}} = \left(\frac{g_z}{g_e}\right)^{-0.82}$$

(35)
It is concluded that there is a weak non-linear effect of contact angle on both the bubble diameter at departure and growth period. A weaker dependence of gravity is seen as the contact angle decreases. Experimental validation of results has been provided by Qiu et al. (2000).

**Effect of Cycles**

The area-averaged Nusselt number, \( \text{Nu}_w \), for repeated bubble growth and departure cycles, as published in Son et al. (1999) are plotted in Fig. 9. It is seen that the microlayer contributes about 20% of the total heat transfer rate. Also, it takes about 10 to 12 cycles before quasi-static conditions are achieved.

**Multiple Bubble Merger**

During nucleate boiling, increasing the wall superheat results in the increase in the bubble release frequency and in the number of nucleation sites that become active. As a result, merger of bubbles both normal and along the heater surface can occur which results in the formation of vapor columns and mushroom type bubbles. Qualitative comparison of the numerical and experimental bubble shapes during the merger of two bubbles in the vertical direction is shown in Fig. 10, as published in Son et al. (2002). Fig. 11(a) shows photographs and numerically predicted shapes when two bubbles merge in the lateral direction by Mukherjee and Dhir (2003). From these figures it can be seen that there is very good agreement between the observed and predicted bubble shapes. A comparison of the bubble growth rate during lateral merger of two bubbles is shown in Fig. 11(b). The predicted growth rate, time of merger, time of departure, and departure
diameter are in good agreement with the experimentally obtained values. Fig. 12 shows comparison of experimentally observed and numerically predicted bubble shapes during lateral bubble merger under low-gravity conditions.

Merger of three bubbles located at the corners of an equilateral triangle for microgravity conditions (fluid: water, $\Delta T_w = 7 \, ^\circ C$, $\Delta T_{\text{sub}} = 0.0 \, ^\circ C$, $g = 0.01 g_e$, $\varphi = 54^\circ$, spacing = 6 mm) is shown in Fig. 13. It can be seen that the bubbles begin to merge at $t = 0.5$ sec. Thereafter, the merged bubble grows as a single bubble and finally lifts off at $t = 4.2$ sec. Fig. 14 (a) shows the bubble growth rate comparison of the three bubble merger process with that for a single bubble. It can be seen that the merged bubble lifts off at a much smaller diameter compared to the single bubble. The growth period for the merged bubble is also smaller than that for a single bubble. Fig. 14(b) shows the net force acting on the vapor mass for the three bubble merger case and the single bubble case. The force acting downward is negative while the force acting upward is positive. It is found that during bubble merger an additional vertical force (which we call the “lift force”) is induced by the fluid motion. At about 2.5 seconds when the force changes sign and the merged bubble starts to detach, the single bubble is still experiencing a negative force and continues to grow. The difference between the two at 2.5 seconds is designated as the “lift force” and this additional “lift force” causes the merged bubble to lift off earlier.

**Extension to Large Designed Surfaces**

To validate the numerical results for heat transfer over a large surface, artificial cavities were microfabricated on 40 mm x 40 mm silicon wafer. Because the computations are slow even for a small number of bubbles in a domain as small as 40
mm x 40 mm x 80 mm, a scheme was developed to quicken the computational process. In this scheme, the surface was subdivided into several reasonable subdomains so that computations could be carried out for each domain separately and the heat transfer contribution from each domain were added to obtain the result for the large domain. Fig. 15(a) shows a large designed heater surface with cavities of known size (4, 7, 10 µm) and locations. Using the locations of the cavities, the large domain is sub divided as shown in Fig. 15(b) so that the interacting bubbles are clustered together. The different subdomains are assumed not to interact with each other and turbulent natural convection is assumed to exist in the regions not covered by the subdomains. Fig. 16 shows the variation of wall heat flux with wall superheat for water at earth normal gravity calculated using the procedure shown above when the location and number of cavities that were found to be active in the experiments were given as input. The change in heat transfer with cycles was accounted for in the calculations. In Fig. 16, the solid symbols are the data obtained from experiments. Reasonable agreement is observed between the numerically predicted and experimentally measured wall heat fluxes. Further work will also involve the heat transfer calculations over several cycles of bubble growth.
CONCLUSIONS

1) Numerical simulations of the bubble dynamics during pool nucleate boiling were carried out without any approximation of the bubble shapes. The effect of microlayer evaporation is included.

2) The scaling with respect to gravity for the bubble departure diameter and time period of growth for both water and PF5060 has been established.

3) Effects of wall superheat, liquid subcooling, contact angle, and level of gravity on the bubble growth process, bubble diameter at departure and growth period have been quantified.

4) Bubble mergers normal to the heater and along the heater leading to the formation of vapor columns and mushroom type bubbles have been studied.

5) The merger process is highly nonlinear. A “lift force” leading to premature departure of bubbles from the heating surface after merger has been identified.

6) A scheme is developed to calculate the wall heat flux for a large surface by subdividing the region of interest into smaller domains.
NOMENCLATURE

\( A_0 \) = dispersion constant, J

\( c_p \) = specific heat at constant pressure, kJ/(kg K)

\( D_d \) = Lift-off diameter of the bubble, m

\( D_{d,s} \) = Lift-off diameter of the single bubble, m

\( g_e \) = gravitational acceleration at earth level, m/s\(^2\)

\( g \) = gravitational acceleration at any level, m/s\(^2\)

\( h \) = grid spacing for the macro region

\( H \) = step function

\( h_{ev} \) = evaporative heat transfer coefficient, W/(m\(^2\) K)

\( h_{fg} \) = latent heat of evaporation, J/Kg

\( k \) = thermal conductivity

\( K \) = interfacial curvature, 1/m

\( l_0 \) = characteristic length, m

\( M \) = molecular weight

\( \dot{m} \) = evaporative mass rate vector at interface, kg/(m\(^2\) s)

\( \dot{m}_{micro} \) = evaporative mass rate from micro layer, kg/s

\( Nu_w \) = Time averaged wall Nusselt number, \( \frac{\overline{q_w l}}{k \Delta T_w} \)

\( p \) = pressure, Pa

\( q \) = heat flux, W/m\(^2\)

\( r \) = radial coordinate, m
\( R \) = radius of computational domain, m

\( \bar{R} \) = universal gas constant, J/kg. K

\( R_0 \) = radius of dry region beneath a bubble, m

\( R_1 \) = radial location of the interface at \( y=h/2 \), m

\( t \) = time, s

\( t_{gs} \) = time period of growth of a single bubble, s

\( t_0 \) = characteristic time, \( l_0 / u_0 \), s

\( T \) = temperature, K

\( u \) = velocity in r direction, m/s

\( \tilde{u}_{int} \) = interfacial velocity vector, m/s

\( u_0 \) = characteristic velocity, m/s

\( u_1 \) = characteristic interface velocity, m/s

\( V_c \) = volume of a control volume in the micro region, m\(^3\)

\( v \) = velocity in y direction, m/s

\( x, y, z \) = coordinates of the computational domain, m

\( X, Y, Z \) = length, width and height of computational domain, m

Greek

\( \alpha \) = thermal diffusivity, m\(^2\)/s

\( \beta_T \) = coefficient of thermal expansion, 1/K

\( \delta \) = liquid thin film thickness, m

\( \delta_T \) = thermal layer thickness, m

\( \delta_\varepsilon(\phi) \) = smoothed delta function
\( \Gamma \) = mass flow rate in the micro layer, kg/s

\( \varphi \) = apparent contact angle, deg

\( \phi \) = level set function

\( \kappa \) = thermal conductivity, W/mK

\( \mu \) = viscosity, Pa s

\( \nu \) = kinematic viscosity, m\(^2\)/s

\( \rho \) = density, kg/m\(^3\)

\( \sigma \) = surface tension, N/m

\( \Delta T_w \) = heating wall superheat, K

Subscripts

\( \text{int} \) = interface

\( l,v \) = liquid and vapor phase

\( r,y,t \) = \( \partial / \partial r \), \( \partial / \partial y \), \( \partial / \partial t \)

\( s,\text{wall}, \text{sub} \) = saturation, wall, subcooling

\( T \) = temperature

\( \infty \) = infinite

REFERENCES


Fig. 1 Macro and micro regions of the mathematical model for numerical simulation.
Fig. 2 Schematic of the experimental set-up of Qiu et al. (2000).
Fig. 3 Comparison of numerical simulations with experimental data (a) Effect of wall superheat, (b) effect of liquid subcooling (fluid: water, $\phi = 54^o$, $g = 1.0g_e$) (Qiu et al. 2000)
Fig. 4 Contribution of the various heat transfer mechanisms during subcooled pool nucleate boiling (Singh and Dhir (2000)).

Fig. 5 Comparison of bubble departure diameter and bubble growth time for water and PF5060 (Abarajith et al. 2003)).
Fig 6. The variation of heat transfer rates with time for various contact angles (a) from micro layer and (b) from macro region (Fluid: water, p = 1.01 bar, $\Delta T_w = 8 ^\circ C$, $\Delta T_{sub} = 0 ^\circ C$)(Abarajith et al. (2002)).

Fig 7. Bubble growth time, departure diameter and Hamaker constant normalized with those at contact angle of 90° as a function of the contact angle (Abarajith et al. (2002)).
Fig 8 (a) Bubble departure diameter as a function of the gravity level, (b) bubble growth period as a function of gravity level.
Son et al. (1999) (fluid: water, $\Delta T_w = 6.2^\circ C$, $\Delta T_{sub} = 0.0^\circ C$, $g = 1.0 ge$, $\phi = 38^\circ$).

Merger from Son et al. (2002) (fluid: water, $\Delta T = 10^\circ C$, $\Delta T = 0.0^\circ C$, $g = 1.0 g$).

Fig. 9 Variation of Nusselt number with time for various bubble growth cycles from

Fig. 10 Comparison of numerical and experimental bubble shapes during vertical $= 38^\circ$).
Fig. 11 (a) Comparison of numerically predicted bubble shapes with experiments (b) Comparison of numerically predicted bubble growth with experimental data of Mukherjee and Dhir (2003) for saturated water at earth normal gravity (fluid: water, $\Delta T_w = 5.0 \, ^\circ C$, $\Delta T_{sub} = 0.0 \, ^\circ C$, $g = 1.0g_e$, spacing = 1.5 mm)
Fig. 12  Comparison of experimental and numerical bubble shapes during the merger of two bubbles at low gravity (fluid: water, $\Delta T_w = 5 \, ^\circ C$, $\Delta T_{sub} = 3 \, ^\circ C$, $g = 0.01g_e$, $\phi = 54^\circ$, spacing = 7 mm) (Abarajith et al. (2003)).

Fig. 13  Growth, merger and departure of three bubbles in a plane (fluid: saturated water, $g = 0.01g_e$, $\Delta T_w = 7 \, ^\circ C \, \phi = 54^\circ$) (Abarajith et al. (2003)).
Fig. 14 Comparison of (a) bubble growth history and (b) normalized three bubble merger cases (Abarajith et al. (2003)).
Fig. 15 (a) A representative section of a large commercially designed Surface

Fig. 15 (b) Division of the large domain into subdomains based on the nucleating cavities.
Fluid: Water  
Domain: 40 mm X 40 mm  
$\phi$: 54°  
$\Delta T_{sub} = 0^\circ$ C  

Wall Heat Flux (W/cm$^2$) vs Wall Superheat, $\Delta T_w$ (K)

Fig. 16 The variation of wall heat flux with wall superheat for Water at $g = 1.0g_e$. 