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Motion of Vapor Bubbles in Subcooled Heated Channel*

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The growth and collapse data as well as the slip data of vapor bubbles were obtained under surface boiling conditions with water flowing in an annular channel with the velocity ranging from 0.1 to 5 meters per second, and with the subcooling ranging from 20 to 80°C under the atmospheric pressure. With the aid of the existing correlations and related informations, several discussions were made on the temperature profile in the boundary layer and on the bubble characteristics such as the bubble lifetime, the maximum diameter, the time of the maximum diameter and the average velocity in the radial motion during the growth and the collapse stages.

1. Introduction

For the proper design and operation of an engineering equipment having a process of heat transfer with surface boiling, it is desirable to understand the individual characteristic of an independent vapor bubble as well as the statistical behavior of numerous bubbles in surface boiling. The present paper is concerned with the individual characteristic, in particular, with the growth and collapse of an isolated vapor bubble observed in the surface boiling in a forced flow channel with a stationary heat generation in a heater.

The purpose of this study is to add information on the characteristics of the growth and collapse of an isolated vapor bubble in surface boiling, to the past works of Gunther⁽¹⁾, Rohsenow-Clark⁽²⁾, Sato et al.⁽³⁾, by observing the motion of a bubble as well as measuring the temperature profile in the proximity of the heated surface. Data were obtained under higher heat flux conditions than in the previous works. Included are the temperature profile near the surface, the bubble growth and collapse curve, the maximum bubble diameter, the bubble lifetime, the average growth and collapse rate, the slip motion on the surface, and so on.

The experiments were made on the flowing water under the atmospheric pressure in an annulus having a fairly large cross-sectional area.

Nomenclature	
D	: Equivalent bubble diameter mm
D_m	: Maximum equivalent bubble diameter mm

H	: Thickness of thermal boundary layer mm
Re	: Reynolds number
T_b	: Bulk liquid temperature °C
ΔT_{SL}	: Liquid superheat °C
ΔT_{sat}	: Wall superheat °C
ΔT_{sub}	: Bulk liquid subcooling °C
d_1	: Outer diameter of the inner tube mm
d_2	: Inner diameter of the outer tube mm
f	: Friction factor
$\bar{h}$	: Roughness mm
q	: Heat flux kcal/m ² hr
t	: Time after the bubble started to grow msec
t_d	: Time of bubble departure msec
t_e	: Bubble lifetime msec
t_m	: Time of maximum diameter msec
U	: Velocity in the boundary layer m/sec
v	: Bulk mean velocity m/sec
y	: Normal distance from the surface mm
e	: Base of natural logarithm
$\ln$	: Natural logarithm
$\langle \rangle$	: Average of plural bubbles
$\bar{}$	: Time average
$ $	: Absolute value
$\dot{}$	: Time derivative

2. Experiment

2.1 Test apparatus

Figure 1 shows a diagram of the test loop and the test section. The test section is made of a concentric annulus, in which the atmospheric water flows upward over the inner heated tube. The outer tube is made of acryl resin having 30 mm inner diameter, while the inner tube is made of stainless steel having 10 mm outer diameter, about 0.2 mm thickness and about 100 mm length silver-soldered with copper tubes of the same diameter and large thickness, to extend to a total length of 250 mm. At about

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1 Circulation pump	6 Make-up pump
2 Orifice flow meter	7 Ion exchanger
3 Preheater	8 Head tank
4 Test section	9 Deaerator
5 Heat exchanger	10 Vacuum pump

Fig.1 Schematic diagram of the water circulation systems and the test section.

60 mm downstream from the heated edge, i.e., about 150 mm from the inlet, the motion of the bubble was photographed by a high speed camera operated at about 14000 frames per second. Flash photos were also taken to observe the bubbles in a wider view area.

In the same axial position, the temperature distribution in the liquid was measured by traversing a fine (about $50\mu\phi$) CA thermocouple, and, in order to know the surface temperature, the inner wall temperature was measured by a CA thermocouple pressed to the wall by a spring. Changes in the axial direction are not the interest of the present study.

The water, passed through an ion exchanger and then degassed, was circulated by a gear-pump. The test section was heated by the three-phase full-wave rectified electric current. The heat flux was calculated from the electrical input by ignoring the end loss.

2.2 Test conditions

The bulk liquid temperature, the bulk velocity and the heat flux were widely changed, whereas the pressure remained atmospheric. The bulk liquid temperature was set 20°C , 50°C and 80°C at the inlet, and the bulk velocity 0.1 m/sec, 1 m/sec and 5 m/sec. The heat flux ranged from incipient boiling level upto half the burnout level.

In all the cases, the liquid temperature rise and also the net void increase between the inlet and the measuring section were small, and therefore the above values of the parameter were very close to those at the measuring section. The

pressure was essentially atmospheric on the liquid surface in the head tank, and it was experimentally ascertained that the static pressure remained approximately constant at the measuring section to correspond to the saturation temperature of $103 \pm 0.5^\circ\text{C}$ throughout this study.

About 33 combinations of the above parameters, including exceptional interpolations, say 65°C or 3 m/sec if necessary, were tested repeatedly 2 or 3 times to check the global reproducibility of the phenomena.

3. Results and discussion

3.1 Liquid temperature distribution

Figure 2 shows the temperature distribution normal to the surface in the boiling liquid. The temperature is practically that of liquid itself, since, in the range of the present experiment, the net void fraction is so small even in the extreme proximity of the heating surface. It is seen, from Fig.2, that the temperature gradient in a thermal boundary layer is controlled by the mass flow rate and is almost independent of the heat flux. The probable reasons are; first, the turbulent diffusion as well as the latent heat transport may be enhanced with an increase of the heat flux resulting in the almost unchanged temperature gradient, and second, the thermal bound-

Fig.2 Temperature distribution normal to the heating surface.

ary layer in surface boiling could be treated approximately as if it were for a single-phase so far as the void fraction is very small. Because the Prandtl number of water is close to unity, the non-dimensionalized temperature distribution may be well discussed by making reference to the velocity boundary layer.

From the correlation equation⁽⁴⁾ for the friction factor

$$f = 0.055 Re^{-0.2} \left(\frac{d_2/d_1 - 1}{d_2/d_1} \right)^{0.1} \quad (1)$$

which is valid for the smooth annuli, and from the generalized distribution function

$$u^+ = 4.4 \log y^+ + 6.2 \quad (2)$$

$$u^+ = \frac{u/v}{\sqrt{f/2}}, \quad y^+ = \frac{y Re}{d_2 - d_1} \sqrt{\frac{f}{2}}$$

the thickness $y_s (u/v = 1 - 1/e)$ where the velocity is $(1 - 1/e)$ times the bulk velocity could be calculated as seen in Fig.3.

Fig.3 Thickness of velocity boundary layer (calculated) and thermal boundary layer (measured) vs. bulk velocity.

Meanwhile, the thickness of the thermal boundary layer H following Forster's treatment⁽⁶⁾

$$\Delta T_{SL} = (\Delta T_{sat} + \Delta T_{sub}) e^{-y/H} - \Delta T_{sub} \quad (3)$$

i.e., the distance where the non-dimensionalized temperature difference drops to $1/e$, is obtained experimentally as also seen in Fig.3. These two curves show good agreement. In Fig.3, the curves of $y_s (y^+=5)$ and $y_s (y^+=30)$ show the thickness corresponding to $y^+=5$ and $y^+=30$ respectively, and H_∞ shows the thickness where the non-dimensionalized temperature difference is as low as 0.01.

From the standpoint that regards the bubble as a kind of surface roughness, it may be more appropriate to use the correlation for rough surfaces. Schlichting's evaluation equation⁽⁷⁾ teaches that the present situation is considered to be rough, if the bubble diameter is taken as a characteristic length. A correlation of the velocity distribution in a rough tube is given by Nikuradse^{(8),(9)} and Prandtl⁽¹⁰⁾

$$u^+ = 2.5 \ln(y/h) + 8.5 \quad (4)$$

where the friction factor in u^+ is given by

$$\frac{1}{\sqrt{f}} = 4 \log(d_2 - d_1)/2h + 3.48 \quad (5)$$

Representing h by $\langle D_m \rangle$, we obtain the velocity distribution and the boundary layer thickness. In Fig.3 are plotted the calculated results, where the subscripts R is attached to the rough surface. Of course it is of little importance to compare the curves for smooth annuli and for rough tubes in quantitative manner, but Fig.3 suggests that the thickness of the thermal boundary layer in surface boiling may be predicted, so far as the void fraction is very small, from the correlation for smooth surfaces better than rough ones.

3.2 Shapes of the bubble

Typically, two types of the bubble were observed. The one, generating on the surface, grows hemispherically in the early and soon approaches near-spherical shape (Fig.4(a)). The other, probably

Fig.4 Typical types of the bubble.

Velocity	0.1 m/sec
Water temperature	24.6°C
Heat flux	8.6×10^5 kcal/m ² hr
Surface temperature	133°C

generating a little apart from the surface, has a comparatively flat shape (Fig.4(b)). The former was more usually observed than the latter. The hypothetical cause of the latter may be attributed to an eddy in a transition layer or a residual vapor in the liquid⁽¹¹⁾.

The flatness of a bubble, defined by $F = D_1/D_2$ where D_1 is the diameter and D_2 is the height measured from the side, changes with time on N-shaped curve in usual case. In the range of the present

experiment, the maximum of F ranges 1.3~1.6 at $t/t_d = 0.2 \sim 0.3$, and the minimum 0.7~0.8 at $t/t_d = 1.0 \sim 1.1$. After the departure, all bubbles approach spherical shape oscillatingly and then collapse completely in subcooled liquid.

It was revealed by the shadow photographs that the interaction of a bubble with the thermal boundary layer was locally far more restricted than imagined. This will substantiate the conclusion in the previous section.

3.3 Bubble growth and collapse curve

One or two of the biggest and the smallest bubbles, and several of the middle bubbles were selected in each case to serve for the analysis. Examples are plotted in Fig.5, where about a half has been omitted to avoid complication. It is clearly observed that the scale of the coordinates becomes smaller as the flow rate or the subcooling is increased.

Figures 6(a) and 6(b) are the plots of D/D_m versus t/t_e for two extreme cases in the present study. The notations of D_m and t_e are the maximum equivalent diameter and the lifetime of a bubble, respectively. For other cases in this study, $0.2 < (t/t_e)_m < 0.5$ was found, where $(t/t_e)_m$ denotes (t/t_e) at $D/D_m = 1$. Gunther's data⁽¹⁾ fall in $0.3 \leq (t/t_e)_m \leq 0.55$. It is anticipated that, under high pressure conditions where the degree of superheat may be far smaller than the degree of subcooling, $(t/t_e)_m$ may be larger than the above values.

The curves of D/D_m versus t/t_e are correlated approximately by

$$\frac{D}{D_m} = 1 - 2 \left| \frac{1}{2} - \left(\frac{t}{t_e} \right)^N \right|^\kappa \quad (6)$$

where N is given by $(t/t_e)_m^N = 1/2$, and κ is determined empirically. Bankoff⁽¹²⁾ assumed $\kappa = 2$ from Gunther's data. For the atmospheric water, $\kappa = 3$ satisfies the data.

With respect to the ratio of the lifetime to the time of maximum diameter, Bankoff⁽¹²⁾ used $t_e/t_m = 2$ in his analysis of subcooled-boiling heat transfer, and Zuber⁽¹³⁾ assumed $t_e/t_m = 4$ to derive an equation

Fig.5 Examples of bubble growth and collapse curves.

Fig.6 Two extreme cases of D/D_m vs. t/t_e .

$$\frac{D}{D_m} = \sqrt{\frac{t}{t_m}} \left(2 - \sqrt{\frac{t}{t_m}} \right) \quad (7)$$

The experimental data are; $2 \lesssim t_e/t_m \lesssim 4$ for the present study, and $1.75 \lesssim t_e/t_m \lesssim 3.1$ for Gunther's study. Consequently, the above assumptions seem to be valid on the atmospheric water when the subcooling is not so small as to prevent the collapse of a bubble by the thermal as well as the mechanical nonequilibrium force.⁽¹⁴⁾

3.4 Characteristic variables with respect to bubble growth and collapse

In this section, many characteristic values are discussed such as the lifetime, the maximum equivalent diameter, the averaged growth-collapse rate and so on.

Figure 7 shows the averaged lifetime --which may have the meaning of expected lifetime if the number of sampled data is sufficiently increased--versus the flow velocity. The reasons for the shortening of the lifetime with an increase of velocity are two-fold: due to the thinning of a thermal boundary layer, hence the reduction of the driving force behind the bubble growth, and due to the enhanced heat transfer in the collapse stage. In the authors' opinion, the former may play the dominant role, since the bubble collapse in highly subcooled water is believed not

Fig.7 Bubble lifetime vs. bulk velocity.

Fig.8 Time of the maximum diameter vs. bulk velocity.

to be controlled by heat transfer but mainly controlled by the radial inertia force of a liquid. The lifetime changes also by the subcooling, the trend of which is in good accord with Gunther's.

The time of the maximum diameter $\langle t_m \rangle$ is plotted in Fig.8, which shows a monotonous decrease with an increase of the velocity.

Figure 9 is a plot of $\langle D_m \rangle$ versus velocity of a liquid. The features inherent to the forced flow appear when $v \gtrsim 0.3$ m/sec is satisfied. Figure 10 is a plot of $\langle D_m \rangle$ versus H , which indicates $\langle D_m \rangle$ increases proportionally to H when H is small. It is suggested from Fig.10 that a predominant condensation should occur on a large bubble to prevent a proportional increase of $\langle D_m \rangle$ to H .

As a kind of the average velocity of the growth stage, $\langle D_m \rangle / \langle t_m \rangle$ is plotted versus velocity in Fig.11. Complicated behaviors are observed in the figure.

Bankoff⁽¹²⁾ has given the equation

$$U_b = \frac{|\dot{D}|}{2} \left(\frac{2B}{1-B} \right)^{1/2} \quad (8)$$

to evaluate the averaged turbulent local velocity U_b in a boiling boundary layer, where B is the portion of the surface

Fig.9 Maximum equivalent diameter vs. bulk velocity.

Fig.10 Maximum equivalent diameter vs. thickness of thermal boundary layer.

Fig. 11 Average growth rate vs. bulk velocity.

area occupied by the bubbles. In the present experiment, B was found to be usually smaller than $1/3$. However, if B is assumed to be in the order of $1/3$ for convenience, then U_b is in the order of $|\dot{D}|/2$.

As observed in Fig. 12, the average of $|\dot{D}|$ for several bubbles, denoted by $\langle |\dot{D}| \rangle$ lay between about 0.2 and 1 mm/msec. So, considerable amount of local turbulence could be expected.

The present data on the maximum diameter D_m were compared with the theoretical prediction of Forster⁽⁶⁾ who had presented the equation

$$\frac{R_m/H}{1 - e^{-R_m/H}} = \frac{\Delta T_{sat} + \Delta T_{sub}}{\Delta T_{sub}} \quad (9)$$

by considering a thermal balance to predict the maximum radius R_m of a hemispherical bubble in surface boiling. For simplicity, the predicted equivalent diameter $\langle D_m \rangle_F$ is provisionally calculated from $\langle D_m \rangle_F = 2R_m$ and compared with the experimental data of $\langle D_m \rangle$. Figure 13 is a plot of $\langle D_m \rangle / \langle D_m \rangle_F$ versus velocity. For the atmospheric water, the experimental values are slightly greater than the predicted ones in wide range of parameters.

Fig. 12 Average growth-collapse rate vs. bulk velocity.

Fig. 13 Comparison between experimental data vs. theoretical value of the maximum equivalent diameter.

3.5 Slip ratio of the bubble

The slip ratio of a bubble on that surface, namely, the ratio of the flowing velocity of a bubble v_b to the bulk velocity v , has been shown by Gunther⁽¹⁾ to be about 0.8. In this study it is found that the slip ratio is $0.3 \sim 0.8$ for v larger than 0.3 m/sec, while v_b remains approximately constant throughout the whole process of growth and collapse. No salient dependence of the slip ratio on the bubble size was observed.

4. Conclusions

For the surface boiling of the atmospheric water in forced convection in an annulus with such slight net void production as in the present study, the following results are obtained:

- (1) The thickness of the thermal boundary layer is not much influenced by the bubble, and well predicted by neglecting the effect of the bubble.
- (2) The equivalent bubble diameter is correlated with time by Eq. (6).
- (3) The lifetime, the maximum equivalent diameter and its time of a bubble, change monotonously with the velocity and subcooling. They decrease linearly with $\log v$ (for $0.3 \lesssim v \lesssim 0.5$ m/sec) and ΔT_{sub} .
- (4) The maximum equivalent diameter is proportional to the thermal boundary layer thickness for velocities of $1 \sim 5$ m/sec. The prediction from the thermal balance gives about 2~5 times the present experimental values.
- (5) The radial velocity of a bubble is in the order of $0.1 \sim 0.5$ mm/msec when averaged throughout the whole process of growth and collapse, and therefore a considerable amount of turbulent heat transfer due to bubble motion can be expected.
- (6) The slip velocity of a bubble on the surface is approximately kept unchanged during growth and collapse.
- (7) The slip ratio is found to be $0.3 \sim 0.8$, not depending on the size of a bubble.

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