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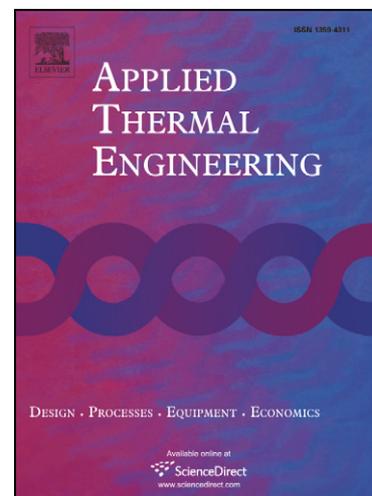
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## SURFACE HEAT TRANSFER DUE TO SLIDING BUBBLE MOTION

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### Abstract:

The presence of a rising bubble in a fluid can greatly enhance heat transfer from adjacent heated surfaces such as in shell and tube heat exchangers and chemical reactors. One specific case of this is when a bubble impacts and slides along the surface. The result is heat transfer enhancement by two main mechanisms: firstly the bubble itself acting as a bluff body, and secondly, the wake generated behind the bubble leads to increasing mixing. The current research is concerned with measuring the heat transfer from a submerged heated surface that is subject to a sliding bubble flow. An ohmically heated 25 micron thick stainless steel foil, submerged in a water tank, forms the test surface. An air bubble is injected onto the lower surface of the test plate, it slides along its length and the effects are monitored by two methods. Thermo-chromic liquid crystals (TLC's) are used in conjunction with a high speed camera to obtain a time varying 2-D temperature map of the test surface. A second synchronised camera mounted below the foil records the bubble motion. Tests are performed at angles of 10°, 20° and 30° to the horizontal. This paper reports on the enhancement of the heat transfer due to the bubble. It has been found that the angle made between the heated surface and the horizontal influences heat transfer by changing the bubble's motion. In general, a steeper angle leads to a higher bubble velocity, which results in greater heat transfer enhancement.

## NOMENCLATURE

$A_{aff}$	area affected by the bubble ( $m^2$ )
$A_{bub}$	cross sectional area of the bubble ( $mm^2$ )
$\alpha$	plate angle to horizontal ( $^\circ$ )
$\varepsilon$	enhancement factor (-)
$h$	heat transfer coefficient ( $W/m^2K$ )
$q''$	heat flux ( $W/m^2$ )
$T$	temperature ( $^\circ C$ )
$\Delta T$	temperature difference, foil to bulk water ( $^\circ C$ )
$v_{avg}$	average bubble velocity ( $mm/s$ )
$v_{inst}$	instantaneous bubble velocity ( $mm/s$ )

## INTRODUCTION

It is known that the presence of bubbles in a system can lead to increased heat transfer from adjacent heated surfaces. In applications such as shell and tube heat exchangers, bubbles are created at nucleation sites on the liquid-solid boundary during boiling. These bubbles grow and detach and can rise, impact and slide along downwards facing heated surfaces; they are known as sliding bubbles. Heat transfer enhancement depends on the interaction between the bubble (and its wake) and the thermal boundary layer.

In an investigation performed by Cornwell [1] it was found that vapour bubbles created on the upstream tubes in a shell and tube heat exchanger impacted and slid around the downstream tubes. This interaction was found to significantly increase the local heat transfer coefficient on the downstream tubes. The heat exchanger consisted of 34 horizontal tubes arranged in two columns; the flow direction was upwards. Heat transfer enhancement is achieved by several mechanisms such as bubble nucleation and detachment, the behaviour of the wake of the bubble, the fluid flow around the bubble and evaporation of the micro-layer between the bubble and the heated surface. There has been much debate in identifying the relative contribution of each mechanism to the overall heat transfer enhancement. Although the geometry in this heat exchanger is different to that under investigation in the current study, the basic mechanisms by which heat is transferred are similar.

It is important to visualise how the bubble interacts with the fluid it moves in to understand how this influences heat transfer. In a study by Qui and Dhir [2], holographic interferometry was used to visualize the wake of the bubble for different angles of plate inclination. At  $15^\circ$  vortices were observed to form downstream of the bubble, detach, and move into the bulk fluid where they dissipated. These vortices can impact on a heated surface, transporting heated fluid away and allowing cooler fluid from the bulk to replenish it, thus increasing the heat transfer coefficient. In a study by Brucker [3], PIV (Particle Image Velocimetry) was used to obtain the temporal evolution of the flow field in the near wake of single rising bubbles of 5-7 mm diameter in water. The existence of a pair of counter-rotating vortices close to the bubble base was confirmed. Qiu and Dhir [2] also used PIV to observe the flow field around a bubble sliding under a heated inclined plate at  $30^\circ$  to the horizontal. Results showed liquid at the front of the bubble being pushed outwards, away from the heated surface. Towards the rear of the bubble, liquid is pulled inwards creating a vortical structure in the near wake. The effect of plate inclination angle on rise velocity and the volumetric growth of bubbles has been investigated by Chen et al. [4] and Maxworthy [5]. For vapour bubbles moving under a submerged surface in water they concluded that bubble velocity increases with bubble volume and plate angle, reaching a maximum at an angle of  $50^\circ$  to the horizontal. In a numerical study by Yoon et al. [6] of boiling heat transfer from a flat surface it was concluded that fluid agitation caused by bubble development and detachment contributes to 80% of the overall predicted enhancement of heat flux. In another experimental study, Thorncroft & Klausner [7] conclude that sliding bubbles can account for as much as 52% of the total energy transfer, outweighing the contribution of bubble nucleation.

Manickam and Dhir [8] used holographic interferometry to visualize the variation in fluid temperature surrounding a sliding vapour bubble. The heat transfer to and from the bubble was quantified. It is known that a liquid layer exists between the bubble and the heated surface but its thickness and contribution to heat transfer is much debated. In the study performed by Qui and Dhir [2], the existence of a wedge-like liquid gap in front of the bubble (determined by the angle made with the surface) is confirmed. The apparent wedge angle is seen to increase as the heater inclination angle increases. Cornwell & Grant [9] also report the existence of a thin evaporating liquid layer beneath a bubble sliding under a horizontal tube. Both water and Flutec (a commercial refrigerant) were used in the study. High speed photography and thermo chromic liquid crystal paints were used to evaluate the contribution of the evaporation of this layer to the overall heat transfer. Results indicated that the liquid layer evaporation can cause 'dry out' (a dry spot between the bubble and surface) which is quenched by the surrounding fluid after the bubble moves away. The existence of 'dry out' was also confirmed by Yan et al.

[10]. Sliding bubbles were observed under inclined plane and curved surfaces with heat transfer enhancement factors of 3-5 reported close to the trailing edge of the bubble compared to the undisturbed state. For large, slow moving bubbles, liquid layer evaporation was found to be the dominant heat transfer mechanism, whereas for smaller bubbles the reduction in evaporation was compensated by the higher velocity and therefore higher wake turbulence. In a study carried out by Kenning et al. [11] where heat transfer to a sliding bubble moving through saturated water was analyzed it was concluded that, for a micro layer thickness of approximately 60  $\mu\text{m}$ , micro layer evaporation could account for only a small fraction of the heat energy transferred from the hot surface to the bubble. In a similar investigation by Qiu and Dhir [2], it was concluded that the heat transferred to the bubble via micro layer evaporation was small (17% of total) in comparison to the heat transfer resulting from induced liquid agitation caused by bubble motion.

The primary objective of this research is to contribute to the current understanding of heat transfer enhancement from a heated inclined surface subject to a bubble flow. Work presented here is for surface inclination angles of 30°, 20° and 10° degrees to the horizontal with a bubble of approximately 6 mm diameter. Whole field temperature measurement of the test surface is achieved using high speed liquid crystal thermography. From this, heat transfer enhancement is calculated. Bubble dynamics are analysed using a second synchronised camera.

Future work is planned to observe the bubble and heated plate using a high speed thermal imaging camera synchronised with two high speed digital still cameras. This will give 3D position and velocity information on the bubble coupled with accurate temperatures of the surface. In addition, stereo PIV will be used to obtain the three component velocity of fluid surrounding the sliding bubble and in its wake in order to accurately understand how the bubble induced mixing enhances heat transfer.

## EXPERIMENTAL SETUP

The experimental apparatus (figure 1) consists of a tilting test tank which can be set to any angle between 0 and 45 degrees by rotating a winding jack. The tank is constructed from 6 mm thick plate glass of dimensions 420 x 420 x 420 mm and is supported by aluminium structural members. An inclinometer mounted on the tank provides angle of inclination. Additional structural elements connected to the tank allow cameras to be mounted above and below the test surface.

The test surface for this experiment measures 300 x 100 mm and consists of a liquid crystal layer backed by black paint applied to a thin electrically heated foil mounted on a 10 mm thick Perspex sheet (figure 2). Liquid crystals are paints which change molecular shape with temperature; thus, it is possible to measure the surface temperature by observing the colour of the paint. The foil is 25 micron thick AISI 321 stainless steel supplied by Goodfellow Ltd. Both the black paint and the liquid crystal (Hallcrest: BM/R28C12W/S40) layers are applied using an Aztek A4702 artists airbrush in conjunction with a compressed air supply at 1.5 bar. The foil is bonded to the surface and electrical contact is made by two machined copper bars.

The test surface requires high intensity lighting to enhance the visibility of the liquid crystals and the bubble. This is provided by 4 high intensity light emitting diode (LED) strips (15 bulbs per strip) mounted on the tank which illuminate the test surface from above and below (see figure 1). Mounting both the cameras and the lighting to the tilting tank ensures consistency in results obtained for all angles of the tank.

Bubble generation is achieved by use of a surgical syringe machined to remove the tip. It is mounted as shown in figure 1 directly onto the test plate surface. The bubble is released by pressing a plunger connected to the syringe via rubber tubing.

Two NAC Hi-Dcam II digital high-speed colour cameras were used in these experiments (20000 fps maximum frame rate with maximum resolution of 1280 x 1024 pixels, dependant on frame rate). One camera observes the liquid crystal layer, the other the bubble motion. Both cameras are PC controlled via the manufacturer's PCI card which allows synchronisation of recording. For these experiments frame rates of 125 fps (0.004 s exposure time) were deemed suitable for both the liquid crystals and the bubble motion due to the dynamic response of the system.

## ANALYSIS

The foil is heated to the upper temperature limit of the liquid crystals,  $\sim 41^\circ\text{C}$ , through resistive heating. The power required to keep the foil at this temperature is dependent on the natural convection flow conditions and therefore on the angle of the surface. The bulk water is maintained at approximately  $25^\circ\text{C}$  throughout the tests. A bubble is introduced to the flow at the plate surface and slides along the plate through the test area (see figure 1) causing local regions on the plate surface to cool. As a result, the liquid crystals change colour passing through the full colour range down towards the lower temperature limit ( $\sim 28^\circ\text{C}$ ). Temperature measurement

outside these limits is not possible for the paints used in these tests. High speed images of the coloured surface (camera mounted above the test plate) are obtained and converted to temperature and enhancement maps through a hue based calibration. This is illustrated in figure 3. The heat transfer coefficient is defined as;

$$h = \frac{q''}{\Delta T} \quad (1)$$

where  $q''$  is the heat dissipated from the foil and  $\Delta T$  is surface to bulk water temperature difference.  $q''$  is the power dissipated through the foil divided by the area of the foil. Lateral conduction within the foil and resistivity changes with temperature is assumed to be negligible in this work. The heat transfer enhancement factor,  $\varepsilon$  is defined as the ratio of the forced convective heat transfer coefficient measured during bubble passage to that measured under natural convective conditions for each angle of inclination of the plate

$$\varepsilon = \frac{h_{bubble}}{h_{NatConv}} \quad (2)$$

which reduces to

$$\varepsilon = \frac{\Delta T_{NatConv}}{\Delta T_{bubble}} \quad (3)$$

Therefore, the heat transfer coefficient enhancement factor can be obtained from two temperature maps of the surface, one before and one after bubble injection.

Successive images of the bubble (obtained from the camera mounted below the test plate) are analysed to obtain the bubble boundary and centroid position. This is done by computer code written in Matlab<sup>®</sup> which compares an image containing the bubble to one with no bubble. This is illustrated in figure 4. Firstly the bubble is isolated in the frame by dividing an image containing the bubble with an image with no bubble. This gives the enhancement plot in figure 4, where any difference between the two images shows up as value greater than 1. The image is then converted to black and white (binary) using a threshold value slightly above 1. The binary image is then analysed to obtain the boundary and centroid. Successive images are analysed this way to obtain the instantaneous velocity,  $v_{inst}$ , of the bubble by tracking the centroid. The instantaneous cross sectional area of the bubble,  $A_{inst}$ , is calculated by summing the pixels within the boundary and converting to the appropriate units. The time separation between frames is obtained from the frame rate.

## RESULTS AND DISCUSSION

Results are presented of bubble position synchronised with heat transfer coefficient enhancement maps of the same surface area for a plate angle,  $\alpha$ , of 30°. Comparisons are then made between this and lower angles (20° and 10°). Bubble velocity and cross sectional area plots are presented for each angle. The bubble size in each test is approximately 6mm. The reference time,  $t$ , is calculated from the bubble first appears in the test section.

### Plate inclination angle: 30°

The test plate angle is set at 30° to the horizontal. The foil is dissipating 244 W of power which results in a nominal foil surface temperature of ~41°C, a slight variation of temperature exists over the length of the plate due to natural convection conditions. A single air bubble is released onto the surface of the inclined plate and travels approximately 85mm before entering the view of the camera. Figure 5 (a)-(d) present from left to right the bubble position on the foil, the corresponding heat transfer coefficient enhancement map and cross sections of this map parallel and perpendicular to the bubble flow. The results are presented for different reference times,  $t$ .

Before the bubble enters the frame the foil transfers heat by natural convection alone, therefore an enhancement factor of 1 is observed for the entire test area. Figure 5 (a) shows the bubble 0.04 s after entering the frame. The corresponding heat transfer plots show a relatively small amount of enhancement immediately behind the bubble with a value of approximately 1.2. There does not appear to be significant enhancement effects directly below the bubble although this may be a limitation of the measurement technique. Figure 5 (b) & (c) both show an increase in the enhancement factor with maxima of approximately 1.4 and 1.5 respectively. The cooling is confined to a region immediately behind the bubble and continues along the path from where the

bubble travelled. The enhancement to heat transfer observed can be attributed to the wake of the bubble, the motion of which is described by Manickam and Dhir [8]. Heated water on the surface is moved away by the bluff body motion of the bubble and replenished by cooler water drawn in from the surroundings. This cools the surface of the foil and increases the heat transfer coefficient, thus increasing heat transfer to the bulk water. After the bubble has passed through the test area a substantial increase in heat transfer can be seen. Figure 5 (d) is at a time of 0.52 s from when the bubble first enters the frame; thus it is approximately 0.3 s since the bubble was at the top of the test area where now a local maximum of 1.8 can be seen. This delay in heat transfer enhancement could possibly be attributed to the time it takes for the cooler liquid to reach the heated plate. The area over which enhanced heat transfer can be seen is almost one third of the observed area in Figure 5 (d) with the effects lasting up to and beyond 6.4 s (enhancement greater than 5%).

As the angle of the plate is decreased to 20° and 10°, similar trends were found to that of 30°. Figure 6 (a) & (b) show the heat transfer plots corresponding to the maximum heat transfer recorded in each case i.e. after the bubble has passed by the test area.

Two main observations can be made from these results. Firstly, the plots of heat transfer enhancement tend to deviate from a straight line as the angle is lowered. This is related to the bubble path and is discussed in the next section. Secondly, the maximum value of enhancement reduces from 1.8 to 1.4 and 1.25 respectively and the maxima occur at a later point in time as the angle is decreased. This is related to the bubble velocity and is also discussed in the next section.

#### ***Bubble Dynamics, and the effect on heat transfer:***

Results of the bubble boundary and centroid together with the bubble path over the test area are presented in figure 7 (a)-(c). The velocity and area as a fluctuation of time for each angle accompany them. These images are obtained as outlined in the analysis above.

As the angle is decreased, two main observations can be made from figure 7; the bubble begins to follow an oscillating path and the bubble velocity reduces. This has an effect on how the bubble enhances heat transfer. As was previously noted, the heat transfer enhancement plots become less straight at lower angles (Figure 6). The cause of this is immediately obvious when the bubble path is plotted (figure 7). It has been observed that the majority of heat transfer enhancement occurs in the immediate wake of the bubble, so if the bubble follows a sinusoidal path then the enhancement is also observed over the same path.

Figure 6 shows the effect of inclination angle on heat transfer enhancement. Both the extent of enhancement and the time where maxima are observed change with angle. Figure 7 gives us insight into why this occurs; it shows the average velocity,  $v_{avg}$ , of the 30° bubble is 218 mm/s compared with 182 mm/s and 123 mm/s respectively for the 20° and 10° tests. It is the velocity of the bubble which gives it its mixing ability; the higher the velocity, the greater the mixing. Water from the cooler bulk gets drawn in to the surface where it replaces heated water displaced by the bubble. As the angle is lowered, the bubble velocity decreases and therefore the mixing reduces, thus reducing the extent of heat transfer enhancement. In addition, the slower bubble takes longer to traverse the test area which leads to the maxima occurring at a later reference time. Table 1 shows the general trend of the results.

As seen in table 1, increasing the plate angle leads to an increase in the duration of enhancement effects. Although duration and extent of enhancement are important parameters, the area over which the enhancement effects are observed is also important. The combination of these three factors determines the increase in thermal energy transferred to the fluid due to the bubble. For each angle, an image was chosen which contained the highest affected area for that test; the extent of the enhancement zone was then calculated. A threshold value of a minimum of 5% increase over natural convection levels was maintained. It is clear that increasing the plate angle leads to an increase in the area over which enhancement can be seen. Thus, for the same energy input (bubble generation and injection in this case), the interaction with a slightly different geometry can lead to very different heat transfer enhancement. This finding may help in exploiting bubble induced heat transfer enhancement in the future.

The next stage of this research will use a high speed, high resolution infra red camera to measure the temperature on the foil surface. This camera will be synchronised with two high speed imaging cameras (mounted parallel and perpendicular to the plate) in order to obtain the three dimensional bubble motion and bubble shape in two planes. Stereoscopic PIV will be used to visualise the three component fluid motion around the bubble and in its wake. This will further contribute to understanding this heat transfer mechanism.

## **CONCLUSIONS**

An experimental study has been conducted of flow dynamics and heat transfer for a bubble sliding along a heated inclined surface. The main conclusions of this research are detailed below:

- Increasing plate angle leads to higher sliding bubble velocities.

- The oscillating motion of the bubble path reduces with larger angles of inclination
- The bubble wake is responsible for most of the heat transfer enhancement
- The area showing elevated heat transfer increases with higher inclination angle as does the duration and extent of the enhancement

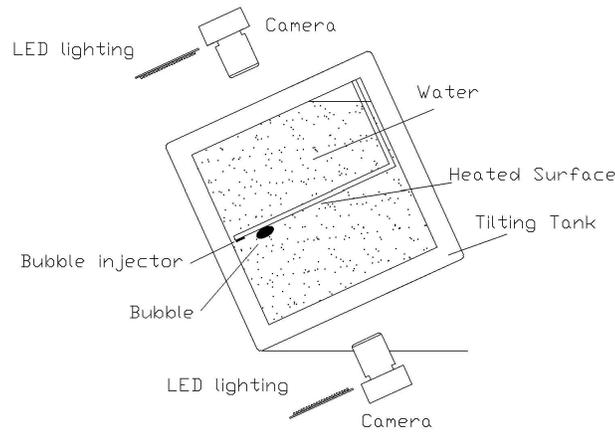
Vapour bubbles occur naturally in many engineering applications where boiling takes place. These bubbles have been shown to increase heat transfer from adjacent heated surfaces. Further investigation into this phenomenon could give rise to improved design that would maximise heat transfer through better interaction with the sliding bubble. This could be achieved by a change in geometry. Alternatively, numerous other applications which currently rely on natural convection alone could benefit from the introduction of gas bubbles to the liquid.

### Acknowledgments

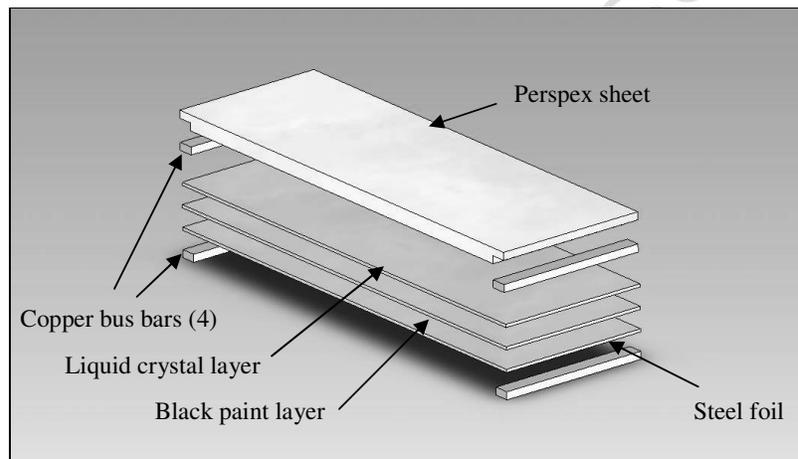
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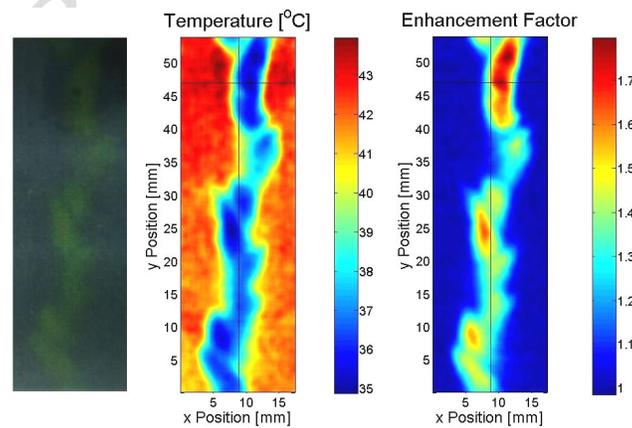
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**Figure 1: Schematic of tilting test tank**



**Figure 2: Exploded view of test surface**



**Figure 3: (left to right) Photo of liquid crystals, Corresponding temperature map and heat transfer coefficient enhancement map**

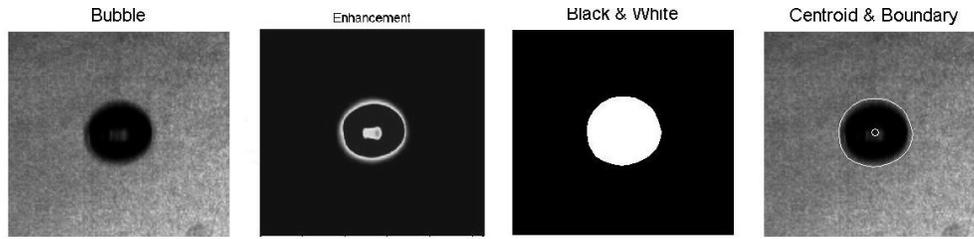
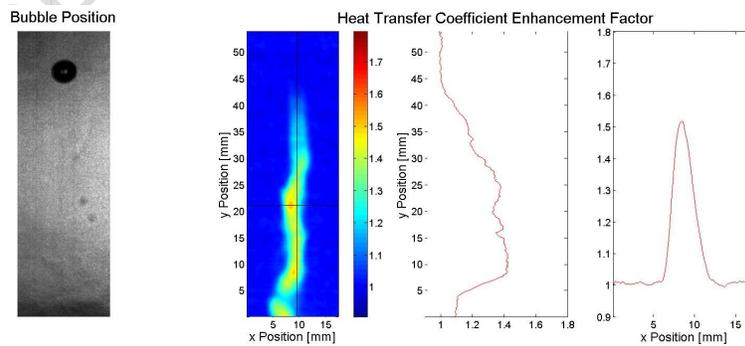
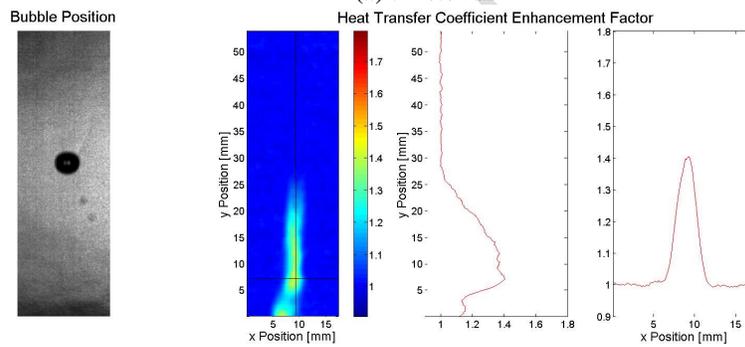
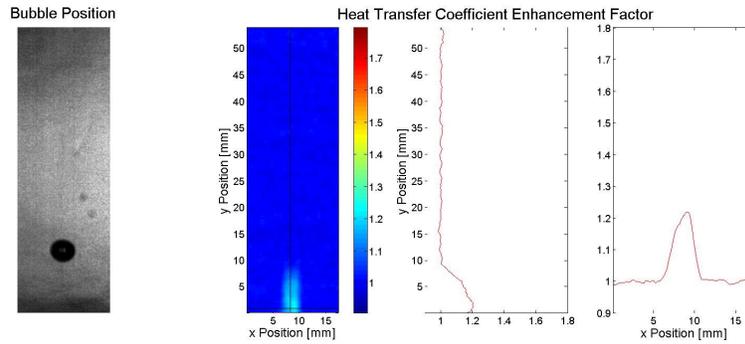
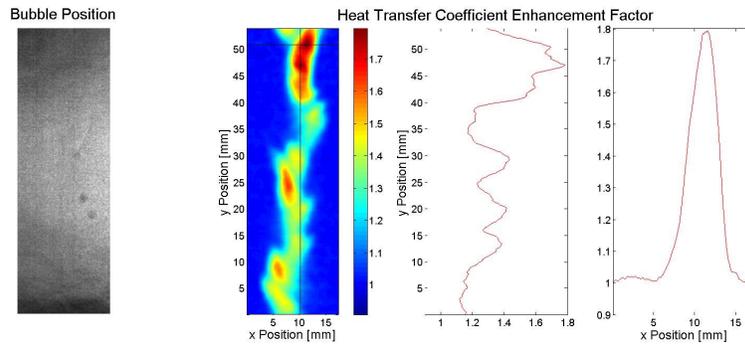
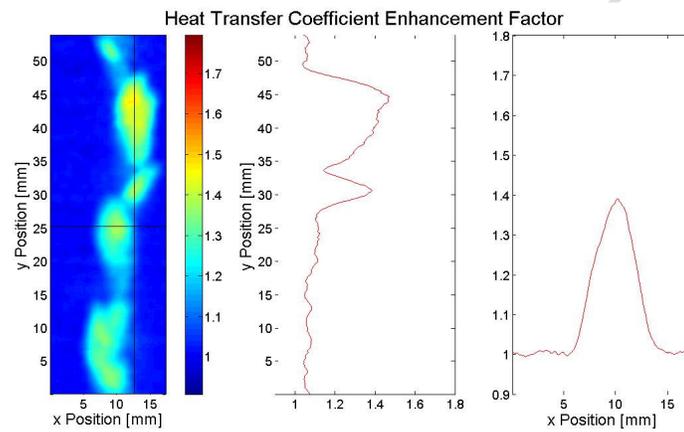
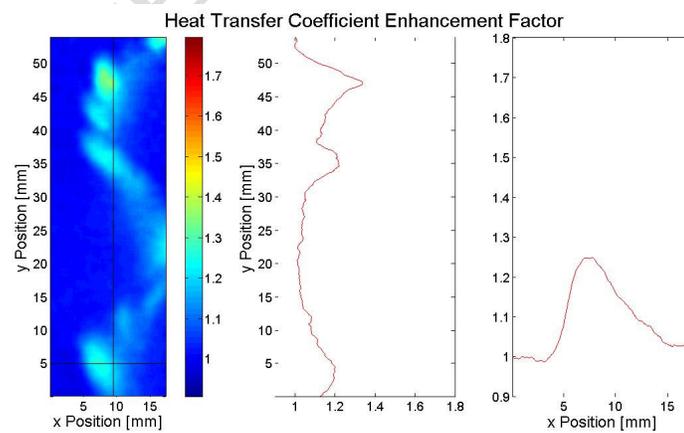


Figure 4 (left to right): Image with bubble, enhanced image, conversion to black and white and bubble boundary and centroid



(d)  $t = 0.52$  sFigure 5 (a)-(d): Bubble motion and heat transfer plots for plate inclination angle of  $30^\circ$  at time,  $t$ .(a)  $t = 0.88$  s,  $20^\circ$ (b)  $t = 1.28$  s,  $10^\circ$ Figure 6 (a) & (b): Heat transfer coefficient enhancement plots for  $20^\circ$  and  $10^\circ$  respectively

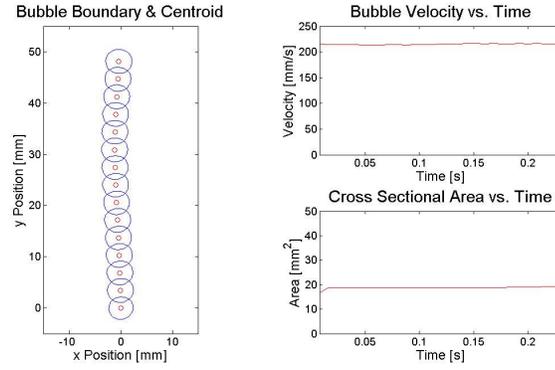
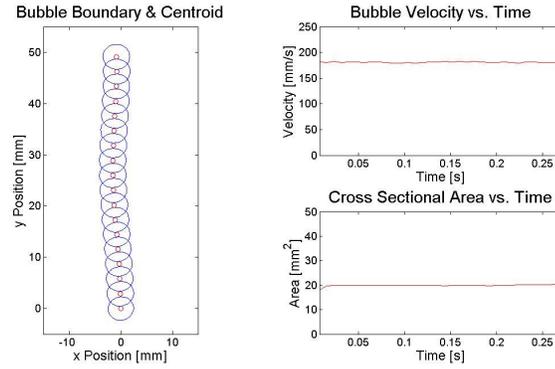
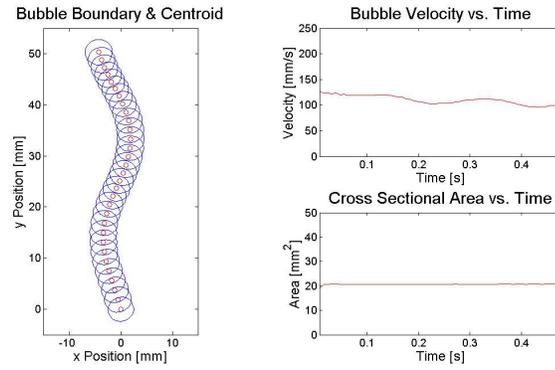
(a)  $30^\circ$ (b)  $20^\circ$ (c)  $10^\circ$ 

Figure 7 (a)-(c): Bubble boundary and centroid, instantaneous velocity ( $V_{inst}$ ) and instantaneous area ( $A_{inst}$ ) fluctuations with time for  $30^\circ$ ,  $20^\circ$  and  $10^\circ$

Plate Angle, $\alpha$ ( $^\circ$ )	10	20	30
Duration of bubble enhancement, (s)	4.72	6.16	6.47
Bubble 'in frame' time, (s)	0.496	0.24	0.216
Bubble Avg. Velocity, ( $mm/s$ )	123	182	218
Enhancement Zone, ( $m^2$ )	$1.94 \times 10^{-4}$	$2.89 \times 10^{-4}$	$5.07 \times 10^{-4}$

Table 1: Plate angle vs. enhancement duration, in frame time, average velocity and enhancement zone