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FOOTPRINTS OF ISOLATED TAYLOR BUBBLE ON WALL TEMPERATURE OF A HORIZONTAL SQUARE MINI-CHANNEL

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ABSTRACT

It is generally known that when a gas Taylor bubble is allowed to pass through the flowing viscous liquid, it modifies the flow field inside the liquid slugs at the head and the tail of the bubble depending upon the respective bubble and mean liquid velocities. It is expected that because of the modified flow field, heat transfer pattern will also be affected. With this motivation, IR thermography of an isolated Taylor bubble, passing through a one-side heated square mini-channel of cross-section 3 mm × 3 mm, has been performed in this study. Degassed and distilled water and clean humidified air are used as liquid and gas phases, respectively. Temporal variation of heated wall has been observed during the passage of the isolated Taylor bubble under uniform heat flux condition. It has been observed that when a Taylor bubble is injected in a steady flow, it creates flow disturbances in its wake which enhances mixing in the fluid and reduces the wall temperature below its steady-state value. The reduction in wall temperature depends on the ratio of length of the Taylor bubble to the heated length (L_b/L_h).

NOMENCLATURE

A	Area of cross section (m^2)
Bi	Biot number ($h.L_c/k_w$)
D	Diameter (m)
Fo	Fourier number ($\alpha.t/L_c^2$)
h	Heat transfer coefficient (W/m^2-K)
J	Superficial velocity (m/s)
k	Thermal conductivity (W/m-K)
L	Length (m)
m	Relative bubble slip
Nu	Nusselt number ($h.D_h/k_l$)
T^*	Non-dimensional temperature (-)
U	Phase velocity (m/s)
α	Thermal diffusivity (m^2/s)

SUBSCRIPTS

b	bubble, bulk
c	characteristic
f	fluid
h	Hydraulic, heated
in	inlet
inst	instantaneous
l	liquid
w	wall

INTRODUCTION

It is a well-established fact that insertion of a gas bubble in a steady liquid flow modifies the flow field, especially in the wake region of the bubble [1]. Therefore, it is plausible that this flow field modification will also change the thermal pattern, and heat transfer may get altered either momentarily or for a long run. There may be two different viewpoints of thermal exchange during the bubble passage. One is change in the convection pattern of the bulk fluid during the bubble passage, and second, is the evaporation of the liquid at the interface [2-4]. There are many studies available which deals with hydrodynamics of non-boiling Taylor bubble flows in vertical/horizontal channels. Many experimental studies have been performed related to the velocity distribution of liquid slugs in the vicinity of nose and wake of Taylor bubble flows [5-8]. It has been clearly shown in these studies that recirculation patterns exist inside the liquid slugs for a particular range of the relative slip ($m < 0.5$) of the bubble.

$$m = \frac{U_b - U_l}{U_b} \tag{1}$$

It has been explained that, if bubble velocity is less than the maximum velocity of liquid element, circulation will be in the liquid slug otherwise, flow will completely bypass. After invoking the assumption used in [9], Eqn (1) can be written as:

$$m = \frac{U_b - J_{tot}}{U_b} = \frac{U_b - \left(Q_l + Q_g\right)/A}{U_b} \tag{2}$$

Hence, for an isolated Taylor bubble with very thin liquid film surrounding it

$$Q_g = U_b A_b \cong U_b A \therefore m \cong \frac{Q_l}{AU_b} \cong \frac{J_l}{U_b} \tag{3}$$

Quite a few studies of hydrodynamics and heat transfer have been performed on Taylor bubble train flow, which provide bulk average transport data. Very few studies are available for non-boiling isolated Taylor bubble flow. In a very recent study [10], wall and fluid temperature measurement at three different axial locations have been shown in the case of isolated Taylor bubble flow in horizontal square channel. In this study, it has been shown that, when a bubble moving in steady liquid flow, reaches to the thermocouple measurement location, fluid temperature increases momentarily but wall thermocouple does not show appreciable change. This was mainly because of higher thermal inertia of the heated wall. No explicit conclusion has been drawn from the heat transfer point of view. Hence, it will be interesting to study the heat transfer in an isolated Taylor bubble flow system.

In the present study, an attempt has been made to experimentally demonstrate the trace of an isolated Taylor bubble flow on the heated wall in a square mini-channel of size 3 mm × 3 mm using IR thermography. Bubbles of various lengths have been injected in steady liquid flow to observe the effect of bubble length on heat transfer. In some experiments, very long bubbles are generated by reducing the liquid flow rate and change in corresponding wall temperature has been reported in comparison to its steady-state value.

EXPERIMENTAL SET-UP AND DATA REDUCTION

The schematic and dimensional detail of experimental setup is shown in Figure 1. A square channel of cross-section size 3 mm × 3 mm is machined on 500 mm × 250 mm × 12 mm polycarbonate substrate. T-junction assembly is provided to facilitate the generation of two-phase Taylor slug flow. Lengths of the two inlets before reaching to the T-junction are kept as 165 mm and 200 mm. After T-junction, an unheated length of 110 mm is provided for the hydrodynamic development of the flow. This unheated length also serves the purpose to stabilize the additional interfacial disturbance which is generated due to shearing of the gas-liquid interface by the inertia of the flowing liquid at the T-junction. The heated length is kept as 125 mm and another unheated 65 mm length is provided after the heater to minimize the end effects in the heated test section. The strip heater is made of 70 µm thin SS strip. 150 mm long and 4 mm wide SS strip is pasted over the polycarbonate square channel such that heater itself will act as fourth wall of the channel, the other three remaining walls being cut in the polycarbonate material itself and therefore they remain insulated. The strip is heated by Joule heating using a high current DC power supply (V: 0 – 60 volts and I: 0 – 50 amps) which ensures a constant heat flux thermal boundary condition. As the thickness of the heated wall is only 70 microns, taking a large value of h = 1000 W/m^{2}K, the corresponding value of the Biot number Bi = $hL_c/k_w << 0.004$, clearly suggesting that the heater wall is thin enough so that its surface temperature can be effectively taken to be equal to the temperature of fluid-wall interface. The Fourier number Fo = $\alpha.t/L_c^2$, is very high (at least of the order of 10^6) because of minuscule thickness of the heater strip. Heat capacity of the heater is equal to 0.15 J/K. Therefore, it is safe to assume that there is negligible temperature gradient as well as no significant signal lag in the transverse direction of the thin heater strip. The wall temperature of the heater is imaged by IR camera and thermocouples (Make: Omega®) of bead diameter 0.3 mm has been inserted in fluid domain at the inlet and outlet cross-section, to measure the fluid temperature. Constant temperature de-ionized and de-gassed water from thermal bath (Make: Julabo® F34 ME, accuracy ± 0.1 K) is supplied in the square mini-channel. A flow meter with a time response of 20 ms (Make: Cole-Parmer®; L-series; laminar-orifice Poiseuille flow meter) is used to measure the liquid flow rate. Clean and humidified air is injected from the T-junction which will generate the Taylor bubble flow in the heated test section.

A pre-calibrated IR camera (Make: FLIR®, Model: SC4000; Indium Antimonide detector array) which has an operational spectral band of 3-5 µm, 14 bit signal digitization and a Noise Equivalent Temperature Difference of less than 0.02 K at 30°C, is used to measure the wall temperature. ThermaCAM™ Researcher V-2.9 is used to acquire and post process the images from IR camera. Acquisition rate has been kept at 20 Hz and IR thermograms are captured at 320 pixels × 256 pixels (105 mm × 96 mm) which give a spatial resolution of 330 microns in the axial z-direction and 375 microns in the transverse y-direction, on the heated thin wall. NI-cDAQ 9172 with NI 9205 module and NI USB 9162 (Make: National Instruments®) cards are used to acquire the voltage signal from flow meter and fluid thermocouple temperature located at the inlet and outlet, respectively. These acquisitions are also done at 20 Hz. In addition, the flow visualization has been done by using Photron-Fastcam®-SA3 high speed camera. Image acquisition has been done at 250 fps with 1024 × 512 pixel resolution for all the isolated Taylor bubble flow experiments.

Data Reduction

As noted above, the primary data collected is temporal IR thermograms and inlet/outlet fluid thermocouple data. From this primary data, instantaneous Nusselt number can be calculated by Eqn. (4). The fluid bulk mean temperature can be estimated by energy balance. Heat loss calculation is done by two different methods. In the first method, experiments are performed in 'no-flow' condition for different heat input. These experiments are performed for quite larger time than the actual experiments. At the end of the 'no flow' experiments thermograms are recorded and the average heater temperature is found for the corresponding heat input. With this data, we

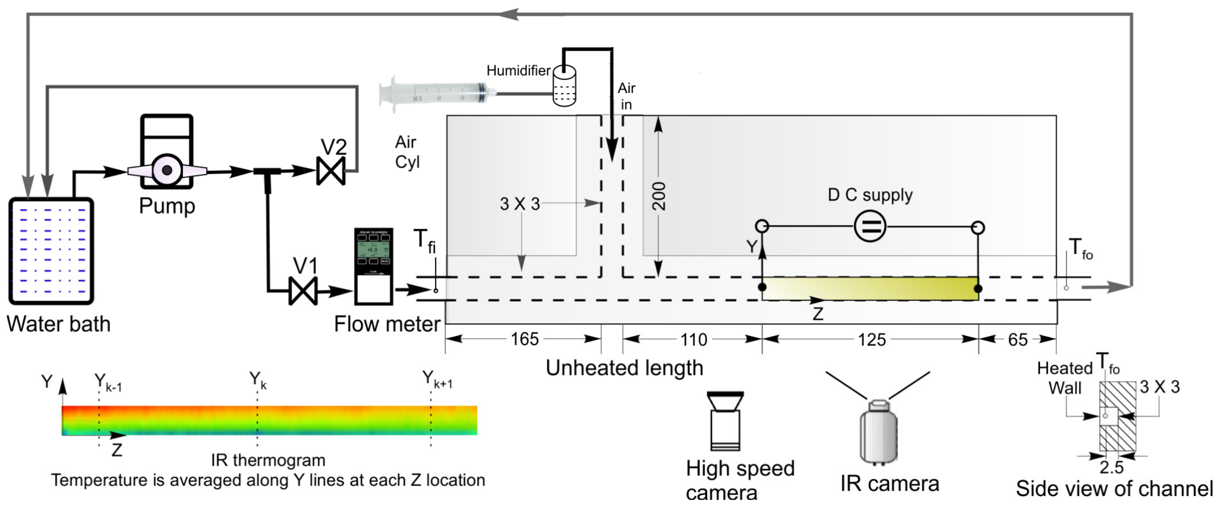

Figure 1. SCHEMATIC AND DIMENSIONAL DETAILS OF EXPERIMENTAL SET-UP.

obtain a calibration curve between average heater temperature and heat input. This calibration curve is used to quantify the heat loss during the main experiments. In the second method, heat balance is done by measuring inlet and outlet fluid temperature by the thermocouples. Heat loss calculated by both the methods are in good agreement and it was not more than 10% of the heat input in any of the experiments.

Time-averaged Nusselt number and non-dimensional temperature are defined as follows:

$$Nu_{inst} = \frac{q'' \cdot D_h}{\left(T_w - T_b\right)_{inst} \cdot k_f} \qquad (4)$$

T_w = local time averaged wall temperature at any given location along z-axis, averaged along the direction of y-axis.

T_b = local time averaged fluid bulk mean temperature estimated by energy balance.

$$T^* = \frac{\left(T - T_{f,in}\right)}{q'' \cdot D_h / k_f} \qquad (5)$$

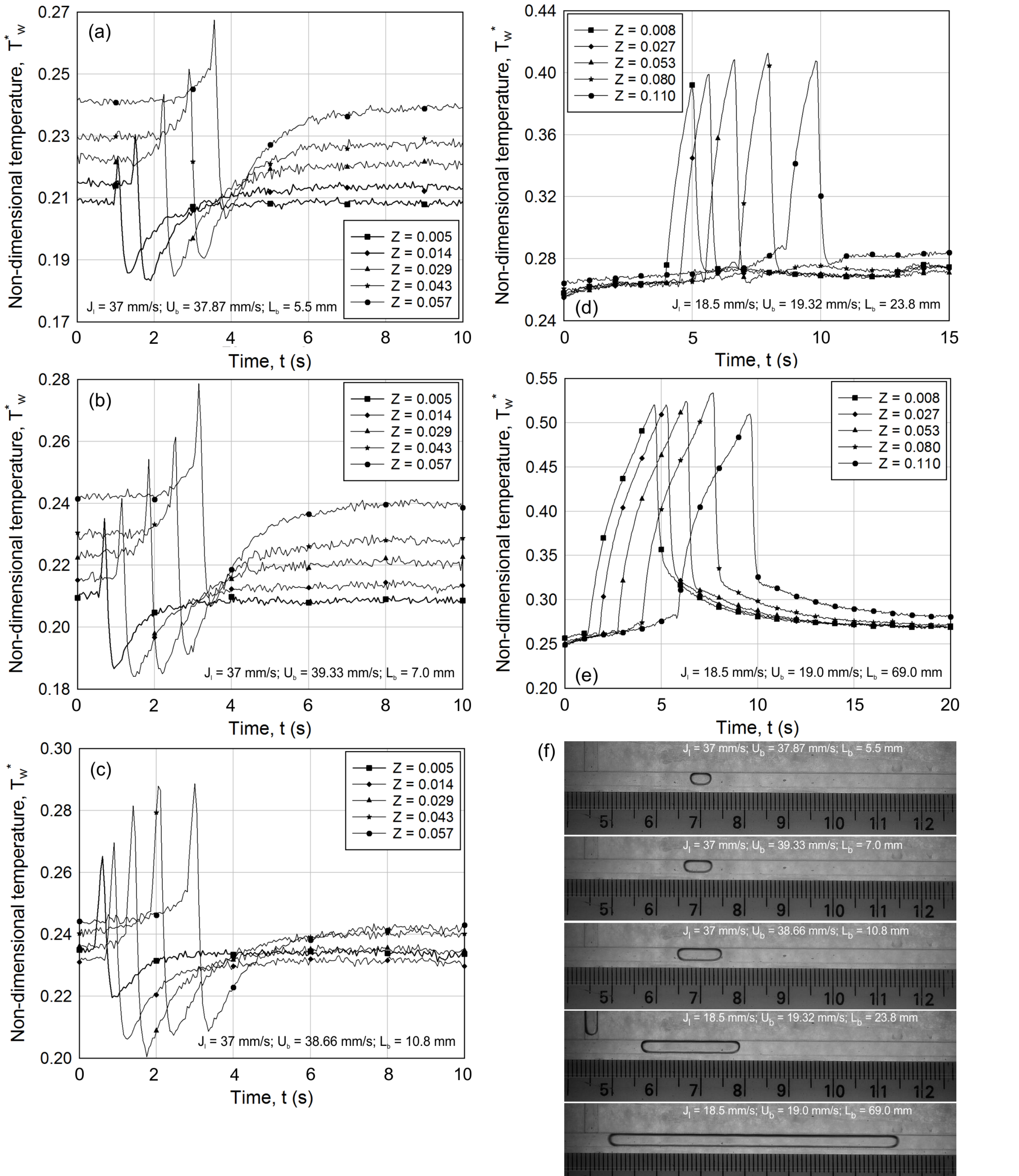

Figure 2. TEMPORAL TEMPERATURE RESPONSE OF HEATED WALL AT DIFFERENT AXIAL LOCATIONS. (a), (b) AND (c) ARE FOR LIQUID Re = 110; (d) AND (e) ARE FOR LIQUID Re = 55; (f) SHOWS IMAGES OF DIFFERENT BUBBLE LENGTH INJECTED IN THE CHANNEL CORRESPONDING TO Figure 2 (a) - (e) RESPECTIVELY.

RESULTS AND DISCUSSION

In Figure 2 (a-e), temporal temperature response of heated wall at different axial locations has been shown, for two different liquid flow rates, and with different bubble lengths. The first three figures, i.e., Figure 2 (a-c) correspond to liquid Re ~110, while the latter two, i.e. Figure 2 (d-e) correspond to liquid Re ~55. Figure 2 (f), shows the corresponding images of bubbles, respectively, with the relevant flow data.

Thermal footprint of the passing bubbles, for both the flow velocities, show some common features. It can be clearly observed that as the air bubble head reaches different axial locations in the heated zone, the local wall temperature of the heater wall first increases as it comes in contact of a low thermal capacity bubble. Subsequently, just after the point of time when bubble tail departs from that location and the liquid phase comes in contact there, the local temperature sharply drops below the corresponding steady-state value of the single-phase liquid flow. The degree of temperature drop is such that it takes some time to regain its steady-state value; this depends on the liquid flow rate. This clearly suggests that an isolated bubble flowing through the liquid creates significant disturbances in its own wake region. This enhances the local fluid mixing in the wake resulting in the thinning of the thermal boundary layer. It can also be observed that at any given location, for a given liquid flow rate, the length of the bubble does not significantly affect the drop in the local wall temperature. This essentially indicates that the disturbances caused in the wake region of the passing bubble are somewhat independent of the bubble lengths, as has also been indicated by several hydrodynamic studies on isolated Taylor bubble flows [for example, see [1, 11].

Low liquid flow rates result in longer bubbles, as is shown in Figure 2 (d,e) and corresponding images in Figure 2(f)-(iv) and (v). The length of the bubble as compared to the length of

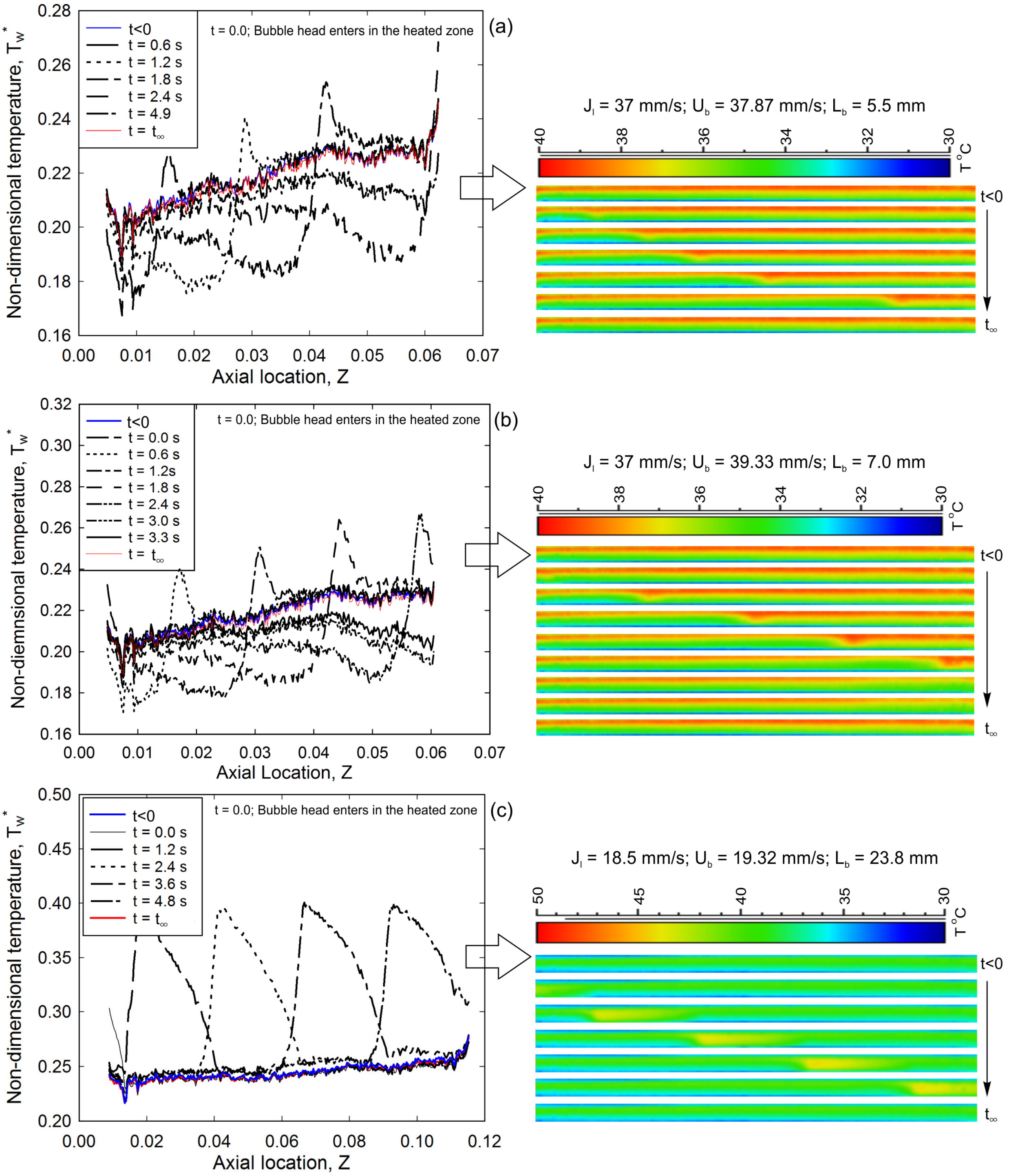

Figure 3. AXIAL VARIATION OF HEATED WALL TEMPERATURE AND CORRESPONDING IR THERMOGRAMS AT DIFFERENT TIME INSTANCES. (a) AND (b) ARE FOR LIQUID Re = 110; (c) IS FOR LIQUID Re = 55.

the heated channel has a definite impact on the local thermal footprint. With long bubbles, the portion of heated zone covered by low thermal capacity fluid (gas-phase) increases as the residence time of the bubble phase inside the channel is larger. Consequently, wall temperature will increase to a much higher value and enhanced mixing created by the bubble wake, becomes rather ineffective. This is clearly observed in the Figure 2 (d) and (e). The contrast between Figure 2(a)-(c) and Figure 2(d)-(e), in terms of regaining the wall temperature corresponding to the steady-state liquid flow value, must be noted. Therefore, it can be inferred that if series of smaller gas bubbles are injected in the heated zone which divides the liquid-phase into small liquid slugs of sufficiently smaller lengths (as compared to the length of the heated zone) will be beneficial in enhancement of heat transfer.

Further, it is interesting to observe the axial variation of heater wall temperature during isolated bubble passage, as shown in Figure 3 (a-c), for different time instances with its corresponding IR thermograms (on the right). Figure 3 (a), (b) correspond to Figure 2 (a), (b), respectively, while Figure 3(c) corresponds to Figure 2 (d). These temperature variations have also been compared with axial temperature variation of steady state single-phase liquid flow well before the bubble insertion ($t < 0$) and at sufficiently after of the bubble has left the heated channel ($t = t_\infty$).

It is self-illustrative that when small bubble (Figure 3 (a) and (b)) slips through the liquid flowing in heated zone, it significantly lowers the wall temperature in comparison to its corresponding steady state temperature value in the wake, as explained earlier. On the other hand, when larger bubble goes in the heated zone, liquid flow steady-state value gives the minimum wall temperature. It is clear that thermal field is relatively less disturbed at the front of the bubble but at the rear of it, heated wall is significantly cooled due to the wake of the tail. This indicates that in comparison with the tail wake, the disturbances in the liquid flowing in front of the bubble have lesser impact on the enhancement of local heat transfer. This situation is qualitatively analogous to external flow over a solid bluff body (if we see from the frame of reference set onto the bubble). In such a situation, major disturbances in the flow field are only seen into the near wake region of the body; the sharp temperature drop at the rear of the bubble is attributed to the local thermal disturbance.

SUMMARY AND CONCLUSIONS

In the present work, thermal behavior has been observed for an isolated Taylor bubble flow inside a square mini-channel. It has been shown that when single-phase liquid flow is perturbed by a bubble slipping through, it changes the flow field significantly in the wake of the bubble. This results in substantial temperature drop of the heated wall. This lower temperature zone exists for quite some time even after the bubble has passed. Thermal field is relatively less altered in the front of the bubble. If the length of the bubble is comparable to that of the heated zone, then the advantage of wake in terms of enhancing heat local transfer cannot be gained. However, tail wake behavior significantly affects the local heat transfer phenomena in smaller bubbles. It is challenging to experimentally quantify the local spatial distribution of heat transfer coefficient, unless local fluid temperature in the liquid region is also simultaneously measured non-intrusively. This, in general, can only be possible by optical means, the work on which is presently underway.

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