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Abstract

Spatiotemporally resolved wall heat transfer measurements can provide valuable insight into the fundamental mechanisms affecting flow boiling in microchannels. Operating at the microscale, necessitates resolving changes in local and instantaneous heat transfer characteristics on the order of 100 μm and 1 kHz, respectively. Straightforward interpretation of transient temperature measurements is often challenging due to the conjugate conduction effects in the substrate, which can dampen the measured and inferred heat transfer quantities. These damping effects are described using a slip coefficient (S), which represents the fraction of the change in the local heat transfer that is registered by a sensor (negligible thermal mass) located on a given substrate. Using S , arguments are presented that the conduction patterns in the substrate are predominantly 1-D (i.e. into the substrate) at suitable spatiotemporal-scales. Building on these fundamental considerations, a numerical procedure is adopted to allow a time varying estimate of the local convective heat flux and heat transfer coefficient from transient temperature measurements. Examples of this framework are showcased with experimental results and discussions for interactions observed during flow boiling of HFE 7000 in a single microchannel (hydraulic diameter = 370 μm). At the relatively low mass flux of 200 $\text{kg/m}^2\text{-s}$ reported in this work, liquid evaporation was found to dictate the local heat transfer trends. High rates of heat transfer were observed to accompany the growth of bubbles and evaporation of the liquid film under vapor slugs. Local dryout was routinely observed in the bubbly and slug flow regime and found to initially enhance heat transfer (i.e. at the creation and subsequent propagation of the three-phase contact line) and present near-zero heat transfer rates in the dried-out domain.

Nomenclature

C	specific heat [J/kg-C]	f	fluid
D	diameter [m]	H	hydraulic
h	convective heat transfer coefficient [W/m ² -K]	i	instantaneous
k	thermal conductivity [W/m-K]	in	inlet
L	characteristic length [m]	m	mean
$\dot{m}$	mass flow rate [kg/s]	$meas$	measured
Nu	Nusselt number [-]	s	surface
$\dot{Q}$	heat rate [W]	sat	Saturated/saturation
q''	heat flux[W/cm ²]	sub	Subcooled/subcooling
S	slip coefficient [-]	Superscripts	
T	temperature [°C]	ss	steady-state
t	time [s]	Greek	
Subscripts		Δ	differences
$conv$	convection	α	thermal diffusivity
$elec$	electric	σ	standard deviation

1 Introduction

Understanding the nature of fundamental phase-change heat transfer mechanisms in microchannels and microgaps is crucial for the development of miniature heat exchangers for microelectronics and other allied industries. Despite the large number of available experimental and computational studies pertinent to flow boiling, mechanisms governing bubble growth, thin-film evaporation, and the dynamics of the three-phase contact line are poorly understood. This is partly because experimentally resolving transient and local changes in the heat transfer process is oftentimes limited by the conjugate conduction effects in the substrate. Several experimental [1–3] and computational [4–7] studies have reported flow boiling events with spatiotemporal characteristics on the order of 1 kHz and 100 μm . Knowledge at these length- and time-scales is essential to develop reliable and robust predictive capabilities that will enable design of the next generation of microchannel evaporators. Additionally, with the recent advances in multiphase Computational Fluid Dynamics (CFD) techniques, the availability of transient local heat transfer characteristics and temperature measurements can serve as a validation to current and emerging algorithms.

For pool boiling, several researchers have developed advanced transient experimental and data processing techniques to address conjugate heat transfer. Transient wall temperature measurements using microheater arrays [8–14], infrared cameras [15–21], and microsensor arrays [22–29] have been used to reveal the intricate physics that dictate bubble nucleation and departure. More recently, researchers have also made inroads in exploring transient dynamics in flow boiling [30–32]. Building on the microfabrication, experimental, and numerical techniques discussed in our previous works [33–36], high frequency spatiotemporally resolved heat transfer measurements and synchronized flow visualization are reported here. The current article seeks to

address a key limitation of our previous publications by presenting a time varying estimate of the local heat transfer coefficient at time-scales relevant to microchannel flow boiling (on the order of 1 kHz) as opposed to just peak values. These estimates are derived under the assumption that at suitable length and time-scales, dictated by the boiling phenomena (e.g. bubble size and residence time), the conjugate conduction patterns are predominantly 1-D. Validation of these arguments, quantitative measures of the suitable spatiotemporal-scales, and associated uncertainties are discussed in detail. Under these conditions, the point measurement of surface temperature is used to extract instantaneous estimates of the local convective heat flux and heat transfer coefficient.

2 Experimental apparatus

2.1 Microdevice

To perform transient flow boiling experiments, a microchannel with three heaters and six temperature sensors (thermistors) was designed and fabricated, as described in references [33–35,37]. The titanium heater (100 nm) and thermistor (75 nm) were connected to contact pads by 1 μm thick aluminum vias. A 1 μm layer of silicon dioxide (SiO_2) was deposited over the entire heater to insulate the thermistor signal from the DC power source and prevent contact with the working fluid. The thermistor was also coated with a 500 nm layer of SiO_2 for preventing electrical coupling via the working fluid. The heaters and thermistors were formed by deposition of titanium and aluminum, followed by a sequence of lithography and wet etching steps. **The width of an individual thermistor was designed to be 15 μm (i.e. across the direction of the flow).**

The assembled microdevice (Figure 1) comprises three components: two processed Pyrex wafers — P_1 and P_2 (20 mm $\times$ 28 mm $\times$ 500 μm each) and a double sided vinyl sticker that was used to bond the wafers together. The microchannel was created along a predefined pattern (1.5

mm wide and 22 mm long), which was cut into the sticker using a laser cutter. The height of the microchannel was defined by the thickness of the vinyl sticker, which was measured to be 211 μm . Two holes were drilled on P_1 to create an inlet and outlet for the fluid. To enable access to the contact pads, 18 holes were drilled through the second Pyrex wafer— P_2 . The liquid (HFE-7000) was introduced into this microchannel, and with the embedded titanium heaters, the flow was heated with a controlled heat flux. The thermistors located on top of the heater layer enable the measurement of the wall temperature, which were used to infer the spatial and temporal variation in the boiling heat transfer coefficient. The width of the heater was chosen to be 1 mm despite a 1.5 mm channel to mitigate potential warping of the vinyl layer at elevated temperatures. By performing measurements away from the heater edge (i.e. at the thermistor), the assumption of a constant heat flux is justified.

Figure 1: a) Schematic of the microdevice used to perform the transient measurements. b) Location of the thermistors with reference to the inlet (figure reproduced from Rao et al. [33]). All results presented in this work pertain to temperatures as measured by thermistor 4 ($x = 15.50$ mm).

A fluidic block, as described in prior works [33], was used to house the microdevice and act as an interface for the electrical and fluidic connections. This test section was connected to

the test loop (detailed in Section 2.2) by means of Swagelok® quick connectors. Electrical connections to the heaters and thermistors on the microdevice were achieved by use of spring loaded probe pins, which were soldered onto a printed circuit board (PCB). Header strips on the PCB were connected to ribbon cables, which were then routed through a breakout board to the data acquisition unit (DAQ) and direct current (DC) power supplies (see Section 2.3). To enable optical access to the flow, a rectangular window was cut out in the center of the PCB.

2.2 Flow loop

A micropump-driven closed-loop flow boiling rig (Figure 2) was constructed for experiments with HFE-7000. A rotameter was used to measure the flow rate through the system. Prior to experiments, the rotameter was calibrated to ensure accuracy. A 5 μm Swagelok® particle filter was placed upstream of the rotameter to prevent blockage of the microdevice. To control inlet sub-cooling, a resistive preheater was installed downstream of the rotameter. The voltage required to achieve a desired degree of subcooling for a given mass flow rate through the system, was empirically determined prior to experiments. All tubing and fixtures were thermally insulated downstream of the preheater section. The heated fluid then entered an inlet restrictor to suppress flow boiling oscillations in the microchannel. A pressure transducer (OMEGA® – PX303) and T-type thermocouple (OMEGA®) were positioned just upstream of the test section (i.e., the fluid block with the MEMS device) to measure the inlet temperature and pressure.

Figure 2: Illustration of the closed loop flow boiling rig used in this study.

2.3 Data acquisition (DAQ) and synchronization

A total of 9 pairs (3 heaters and 6 thermistors) of electrical connections from the PCB, were fed into a breakout board. The breakout board was used to split the heater power lines and the thermistor signals to preserve signal integrity and minimize electromagnetic coupling. Temperatures were inferred from resistance measurements of the thermistors. Each thermistor was connected to an external excitation module (NI SCXI-1581), which produced exactly 100 μA of current. The induced potential drop across each thermistor was independently measured by an NI SCXI-1125 isolation amplifier. The NI modules were integrated into a SCXI-1000 chassis with a SCXI-1600 communications module to interface with a computer. To facilitate the synchronization between analog measurements and the high-speed visualization, a trigger TTL (transistor-transistor logic) signal from a pulse generator was fed into the analog input trigger port on the SCXI-1600. A shunt network was utilized to simultaneously power the heaters and measure the electrical power dissipated. The potential drop across the heater and shunt resistor were measured for each active heater during an experiment. Individual heaters were powered using the Agilent E3645 digital DC power supply. An inverted microscope (Ziess Observer – Z1) and a high-speed camera (Phantom 4.2) were used for flow visualization. A high precision

pulse generator (Quantum Composers Plus – Model 9518) was used to trigger both the temperature DAQ and the high-speed camera with a TTL pulse.

2.4 Thermistor calibration

Instantaneous wall temperatures were inferred from the resistance measurements of the thermistors. The effect of Joule-heating from the source current of the excitation module was evaluated to be negligible compared to the thermal energy input from the heater assembly. The resistance-temperature relation for the thermistors were determined via a calibration process, wherein the microdevice was packaged in a specially designed test section made of oxygen-free high conductivity (OFHC) copper. Four calibrated thermocouples were then mounted onto test fixture, one of which was placed directly under the microdevice to monitor the device temperature. The test section was then placed in an insulated, temperature controlled oven and the set-point was incrementally changed from room temperature (23 °C) to a maximum of 120 °C. Conditions were sought where the temperature recorded from the thermocouples was the same and remained unchanged for 30 minutes to validate the assumption that the microdevice and the on-board thermistors were at this same temperature. The thermistor response was always found to present a linear relation to the temperature with typical sensitivity of 1.5 °C/ Ω (see Figure 3).

Figure 3: Typical resistance – temperature trend for thermistors used in this study. Thermistor calibration was performed in a thermally insulated constant temperature oven.

3 Experimental details and uncertainty

In a typical experiment, the heat flux provided by the titanium heaters on the device was parametrically varied for the target flow rate, $\dot{m}$ (kg/s). From the electrical power dissipated by the heater (Q_{elec}), the steady-state heat loss term (Q_{loss}^{SS}) was subtracted to calculate the heat load dissipated to the fluid. The heat loss term, Q_{loss}^{SS} , was experimentally estimated by subjecting the fluidic block-microdevice assembly to evacuated conditions (close to vacuum). In the absence of any natural convective heat transfer (evacuated conditions), at steady-state the heat generated by the heater is equal to the steady-state heat loss (ignoring radiative heat transfer). The heat loss term was correlated to the average surface temperature measured from the different thermistors. The flexibility of three discrete heaters allowed for non-uniform heat flux boundary condition across the length of the channel. The heat loss term was established for each unique configuration to account for the change in the conduction patterns (Fig. 4).

Figure 4: Experimentally measured heat loss of heater 3 was correlated to its average surface temperature. The average heater 3 temperature was defined as $(T_3 + T_4 + T_6)/3$, where T_i are measurements from individual thermistors.

In order to calculate the time averaged ($\bar{h}$) and the transient heat transfer coefficient (h_i), the steady-state mean temperature of the fluid (T_f) was defined as,

$$T_f = T_{in} + \frac{(Q_{elec} - Q_{loss}^{SS})}{\dot{m}C} \quad 1$$

where C (J/kg-C) is the specific heat of the fluid and T_{in} is the inlet fluid temperature. Quasi-steady temperature readings were used to determine if steady-state boiling was achieved before the high-speed transient experiment was triggered. If the estimated T_f was greater or equal to the saturation temperature of the fluid, T_{sat} , the latter was used instead. Typical temperature acquisition was performed at 10 kHz and the synchronized footage was recorded between 3 – 10 kHz.

The magnitude of the unbiased uncertainty in measured local temperatures by the thermistors was estimated using an uncertainty propagation analysis [38]. Error in the inferred temperature measurement arises from three sources: the excitation current (± 50 nA max), the

voltage measurement module ($\pm 150 \mu\text{V}$ max), and the thermocouple used for device calibration ($\pm 0.5^\circ\text{C}$). With typical thermistor resistance varying between $250 - 500 \Omega$, the error in temperature measurements was estimated to be $\pm 0.6 - 1.2^\circ\text{C}$.

4 Numerical processing

4.1 Theoretical considerations

Conjugate conduction - convection effects must be taken into account to interpret the spatiotemporally resolved wall heat flux and the boiling heat transfer coefficient. Despite the small thermal mass of the sensor, the measured response is dictated by a finite area surrounding the thermistor. In earlier works [34–35], the framework for interpreting this transient response using a non-dimensional variable, the slip coefficient S , was presented. An ideal sensor (zero thermal mass), when located on an infinitesimally thin constant power heater and a perfectly insulating substrate, would record a step temperature response to a step change in the heat transfer coefficient ($h_1 \rightarrow h_2$ or $Nu_1 \rightarrow Nu_2$), as shown in Figure 5. Here, h_1 (or Nu_1) represents the initial quasi-steady heat transfer coefficient of the system and h_2 (or Nu_2) represents the value after the step change. The constant heat flux dissipated by the heater (q'') is wholly transferred to the fluid, allowing the direct estimation of the instantaneous heat transfer coefficient from the surface temperature measurements ($T_{s,i}$) as,

$$h_i = \frac{q''}{(T_{s,i} - T_f)} = \frac{q''}{\Delta T_{ideal,i}} \quad 2$$

Figure 5: Sensor temperature response accompanying a step change in local heat transfer coefficient (Nu) is a strong function of the thermal properties of the substrate material (i.e., k_s and α_s). A non-dimensional number, S , is defined to correlate the temperature drop measured by the sensor located on a real substrate to that obtained with an ideal substrate (i.e. undamped). For all discussions in this subsection it is assumed that $Nu_2 > Nu_1$, i.e. a local increase in the heat transfer rate.

For an actual sensor, the temperature decay following a step change in h (or Nu) is damped compared to the ideal case and the extent of which is dictated by the thermal conductivity (k_s), diffusivity (α_s), and the area (represented as D_{h2} — the characteristic diameter of said area) under the influence of the new heat transfer coefficient — h_2 (or Nu_2) . The slip coefficient is defined as the fraction of the actual change in the local heat transfer quantity (i.e. $\Delta Nu = Nu_2 - Nu_1$) that can be captured by a sensor – substrate pair. Mathematically,

$$S_i = \frac{Nu_{meas,i} - Nu_1}{Nu_2 - Nu_1} \quad 3$$

where, $Nu_{meas,i}$ is calculated based on the experimentally measured surface temperature ($T_{s,i}$) and the constant heat flux assumption as,

$$Nu_{meas,i} = \frac{h_{meas,i} L_h}{k_f} = \left[\frac{q''}{(T_{s,i} - T_f)} \right] \left(\frac{L_H}{k_f} \right) \quad 4$$

In Eq. 4, L_h and k_f represent the hydraulic diameter and the thermal conductivity of the fluid respectively. If $S(t)$ is known *a-priori*, it offers a way to relate the damped measurement to the

actual change in the local heat transfer coefficient h_2 (or Nu_2). This treatment thus eliminates the need to characterize the time varying convective heat flux across the solid interface. For a perfectly insulating substrate (k_s and $\alpha_s = 0$), devoid of any steady/transient conduction effects, $S = 1$ at all times. For a thermal superconductor (k_s and $\alpha_s = \infty$), all of the transient response is instantaneously muted leading to $S = 0$ for all $t > 0$. It is interesting to note that owing to the steady-state heat-loss (or heat-gain, depending on relative magnitudes of Nu_1 and Nu_2), S at steady-state (i.e. $t \rightarrow \infty$) does not equal to unity.

The slip coefficient was computed using a 3-D finite element model for a range of Nu_1 , Nu_2 , and q'' values. Details of the model as well as several useful discussions pertaining to the transient interactions are presented in prior works [34,35]. Results indicate (Figure 6) that the transient sensor dynamics is heavily driven by the area affected by Nu_2 (i.e. D_{h2}). The trend lines for different D_{h2} , branch-off from one another depending on the time taken for the thermal signal to diffuse into or out of the area defined by D_{h2} . As D_{h2} increases, the thermistor, which is located at the center of this domain is less influenced by the effects at the periphery and thus remains unaffected from spreading effects (in-plane) for a longer time interval. Additionally, the evolution of S shows a very weak dependence on magnitude of the change in the heat transfer rates (i.e. $\Delta Nu = Nu_2 - Nu_1$) for the time-scales relevant to flow boiling studies – i.e. in the > 10 kHz domain.

Figure 6: Computed slip coefficient for Pyrex. a) for a range of areas exposed to the change in the heat transfer coefficient ($Nu_2 = 5Nu_1$). b) for different magnitudes of the change in the Nusselt number.

4.2 1-D model

With the S parameter it is possible to refine the methodology presented in Rao et al. [33] to estimate the transient wall heat flux and heat transfer coefficient. Arguments from the preceding section indicate that as the area of influence (i.e. D_{h2}) of the transient heat transfer coefficient increases, the resultant change in the surface temperature is less dependent on the in-plane effects and dominated by the conduction in the out of plane direction (into the substrate). In the extreme limit, this effect results in a situation where the surface temperature change in response to changes in the transient heat transfer coefficient becomes independent of D_{h2} . In this limit, the temperature change accompanying the change in the heat transfer coefficient can be adequately captured as a 1-D thermal system.

In order to validate this hypothesis, a 1-D finite element model was created as shown in Figure 7. Boundary conditions identical to the 3-D model were prescribed at the nodes corresponding to the fluid – solid interface ($x = 0$), heater ($x = 1 \mu\text{m}$), and the bottom edge of the Pyrex wafer ($x = 501 \mu\text{m}$). Appropriate material properties and thermal boundary conditions were prescribed to enable a direct comparison of the S parameter for Pyrex calculated from the 3-D model where $Nu_2 = 2Nu_1$.

Figure 7: Schematic of the 1-D heat transfer domain used to study the sensor dynamics in the limit of a very large D_{h2} . Boundary conditions were imposed at the surface node (fluid – solid interface), heater node and the outer edge of the wafer. Element sizes were refined to achieve grid independence of the results.

The trend for S calculated from the 1-D model displays the same behavior as the other cases calculated from the 3-D model (Figure 8). The dynamics of the system are driven initially by thermal capacitance (relatively flat S) and subsequently by diffusion (gradually increasing $S(t)$ to steady state). Similar to the curves for the larger values D_{h2} , S calculated from the 1-D model branches off from the $D_{h2} = 500 \mu\text{m}$ curve after allowing sufficient time for thermal diffusion in the in-plane direction from the peripheries to the thermistor located at the center. The S for the 1-D system presents the highest value amongst considered D_{h2} 's. This is because of the heat

transfer being constrained to just one dimension, eliminating the scope for any thermal diffusion into the domain (as $Nu_2 > Nu_1$) to reduce S . Together, these results indicate that under suitable circumstances corresponding to a large D_{h2} and intermediate times, the transient response can be adequately described by a 1-D approximation.

Figure 8: The slip coefficient calculated with the 1-D approximation is compared to the results from the 3-D model. Diverging trends are observed only after sufficient time for thermal diffusion (~ 0.01 s).

Operating under the 1-D assumption offers to simplify the data processing technique. Though the S parameter provides physical insight into the nature of the conjugate conduction – convection heat transfer, its application to estimate the transient heat transfer coefficient requires *a-priori* knowledge of $S(t)$. Additionally, parametric calculations are required to estimate $S(t)$ at different heat flux (q'') values and impacted areas (D_{h2}). Furthermore, the implicit assumption in using the slip coefficient requires all changes in the Nusselt number to be treated as step change, which is typically not the case. If the 1-D model (Figure 7) can accurately describe the thermal interactions, then the same model can be used to calculate the transient convective heat flux at

the solid – fluid interface. The recorded temperature data (T_i) can be used as a time varying Dirichlet boundary condition at the surface node to solve the 1-D model for the experimental heat load ($Q_{elec} - Q_{loss}^{SS}$) and fluid temperature (T_f or T_{sat}). The temperature distribution in the 1-D domain allows the calculation of transient convective heat flux, q_{conv}'' , as

$$q_{conv}''(t) = -k_s \left. \frac{dT_{1-D}}{dx} \right|_{x=0}^{t=t} \quad 5$$

Where $T_{1-D}(x,t)$ is the solution from the computational model and k_s represents the thermal conductivity of the substrate. The transient heat transfer coefficient can then be calculated as

$$h_i = \frac{q_{conv,i}''}{(T_i - T_f)} \quad 6$$

The longest event (or minimum frequency) that can be resolved using a 1-D assumption for a given D_{h2} was identified as the time-step where the predictions from the 3-D and the 1-D models diverge (Figure 9). Thus, for all events with a greater characteristic frequency, the 1-D and the 3-D models predict the S identically. As an example, consider the heat transfer associated with a vapor slug. Vapor bubbles on the order of 400 μm in diameter were often encountered in this study. The minimum frequency corresponding to a characteristic area for $D_{h2} = 400 \mu\text{m}$ in Figure 9 is ~ 100 Hz—allowing the use of the 1-D assumption to calculate the heat transfer coefficient. Additional trend lines are also presented in Figure 9, for 5% (---●---) and 10% (---●---) deviation from the 3-D model to facilitate analysis in situations where longer events need to be analyzed.

Figure 9: The minimum characteristic frequency that can be analyzed with the 1-D assumption is shown as a function D_{h2} . All data points within the shaded area represent conditions in which 1-D assumption is applicable with identical accuracy as a 3-D calculation. Additional trend lines are shown for 5% (---●---) and 10% (---●---) deviation from the 3-D results.

The maximum frequency or time-scales that can be analyzed with this technique can be identified by considering the signal-to-noise ratio (SnR) of the temperature measurements. For all S less than 0.05, the value of the temperature rise/fall in the thermistor is below the latent uncertainties in the temperature measurement, rendering analyses in such domains highly inaccurate. The transient heat transfer coefficient results calculated with the 1-D assumption is presented in the following section for flow regimes typically encountered in microchannels.

4.3 Uncertainty considerations

The uncertainty in the measured temperature presents additional considerations in the use of the 1-D assumption to estimate the transient wall heat flux. Unbiased fluctuations in the experimental data are propagated as large gradients in the boundary conditions owing to the high sampling rate. As an example, consider a standard uncertainty of ± 0.8 °C (i.e. $\pm 2\sigma$), at a sampling rate of 10 kHz, this produces an artificial gradient on the order of 16000 K/s without any

physical basis for the change. The effect of this uncertainty on the calculated transient wall heat flux was minimized using the Savitzky – Golay digital filter [39] to smoothen the raw data. The technique was implemented by fitting a third-order polynomial through five neighboring points. In order to provide a quantitative bound on the levels of uncertainty, experimental data was processed by running the 1-D model for three different sets of boundary conditions: T_i , $T_i+2\sigma$, and $T_i-2\sigma$. Results for $q''_{conv,i}$ from Figure 10 indicate that deviations on the order of 25 % can be expected to arise from the uncertainty in temperature measurements.

Figure 10: Uncertainty in the predicted heat flux arising from uncertainty in temperature measurements. Deviations on the order of 25% were observed. Note the large fluctuations in the first 1.5 ms of the data arising from 1-D initial state of the 1-D model.

From Eq. 6, the sensitivity of the transient heat transfer coefficient (h_i) is driven through its dependence on q'' as well as the temperature difference across the solid-fluid interface (i.e., wall superheat, ΔT_{wall}). Drastic deviations in the predictions of h_i are encountered when ΔT_{wall} tends to diminish. As a rule of thumb, all results are presented for cases where $\Delta T_{wall} \geq 2^\circ\text{C}$. Despite the digital filtering of experimental data, it was observed that results for the derived

parameters ($q_{conv,i}''$ and h_i) were prone to inaccuracies over the first few time steps, for example, opposite trends for $q_{conv,i}''$ in Figure 10 and negative $q_{conv,i}''$ and h_i in Figure 13 and Figure 14. This issue was addressed by choosing the starting points of the analysis to provide a sufficient delay (typically $\sim 2\text{-}3$ ms) before the start of the phenomena under study.

5 Results and discussions

Results are presented in the following section for a variety of flow interactions. The fluid was introduced into the microchannel at an inlet temperature of 24.5°C and a time averaged mass flux of $200\text{ kg/m}^2\text{-s}$. The properties of HFE 7000 were obtained from the material database provided in the Engineering Equation Solver (EES) software, which estimates properties in the subcooled region assuming the fluid to be incompressible. The at the system pressure of 18 kPa, the saturation temperature of the fluid was estimated to be 41°C . In all subsequent figures, the time stamps of the synchronized high speed images are marked on the transient temperature, heat flux and heat transfer coefficient graphs by means of a ★ symbol. A red line is drawn across the physical span of the thermistor in all high speed camera images to indicate the location where the measurements were made. Additionally, to aid direct comparison, the (----) trend line represents the time averaged measurements at the same sensor location.

5.1 Bubbly flow

After the sequence of bubble nucleation, initial growth, and departure, bubbles tend to flow along the channel, growing in size due to vapor addition (assuming a continuous heat flux). This bubble growth impacts the local heat transfer downstream of its nucleation site via advective mixing and evaporative effects. Several models to describe these interactions have been developed at the macroscale. As reported in Szczukiewics et al. [40], several computational

studies [4,41,42] have studied the growth of moving bubbles in microchannels. Due to the challenges involved with decoupling the advective and phase change heat transfer processes as well as the convection-conduction conjugate effects, the availability experimental data is rather limited.

The thermal interaction accompanying the motion of a single bubble is shown in Figure 11. The sequence of high-speed images, surface temperature, and the evolution of the transient heat transfer coefficient indicate that there is a periodic and repeatable set of interactions following a bubble pass. The fall of the surface temperature and rise in the local heat transfer rate as the leading edge of the bubble reaches the thermistor, indicate the presence of flow structures at the nose of the bubble that tend to promote mixing (0.00 – 0.61 ms for Bubble 1 and 10.47 – 11.39 ms for Bubble 2). When the bubble passes over the thermistor, the trend of increasing heat transfer coefficient continues (0.61 – 2.77 ms and 11.39 – 14.00 ms) as the bubbles grow in size—indicating that the vapor generation process strongly influences the local heat transfer. The temperature was found to increase as the bubbles move away, indicating minimal enhancement effects at the tail end of the bubble. It should be noted that these discussions are derived from the fundamental assumption that the fluid is at a constant temperature (T_m or T_{sat}). The presence of superheated fluid in such situations makes a very strong case for the simultaneous measurement of the temperature of the liquid phase to better understand the local heat transfer. The peak values of the heat transfer coefficient were found to be, on an average, between 2 – 3 times greater than the values for the periodic liquid-dominated conditions between two bubbles (i.e., 18,000 W/m²-K vs. 5,000 W/m²-K). The time averaged heat transfer coefficient computed over the entire data set (~ 10 s), indicated by the dashed line (----) in Figure 11, was estimated to be 7,800 W/m²-K—which is understandably larger than the 5,000 W/m²-K effected by the liquid

phase and much smaller than the short but potent effects of the bubble growth period (18,000 $\text{W/m}^2\text{-K}$).

Figure 11: Transient heat transfer characteristics for the periodic interactions accompanying bubbly flow. The enhancement in the local heat transfer is predominantly influenced by the evaporative growth of the bubbles. Experimental conditions: heat flux = 5.2 W/cm^2 , mass flux = 200 $\text{kg/m}^2\text{-s}$, and $\bar{h} = 7,800 \text{ W/m}^2\text{-K}$.

5.2 Bubble-bubble coalescence

Driven by the stochasticity of the bubble ebullition processes, two bubbles occasionally coalesced. At higher heat fluxes, large vapor slugs (Taylor bubbles) were formed by the same mechanism. To illustrate this phenomena and the impact on the local heat transfer coefficient, experimental results from the same data set as the single bubble pass of Section 5.1 are shown in Figure 12. The trends for the first ~ 6 ms are those accompanying the pass of a single bubble, as described in the preceding case. At the 6.46 ms mark, a bubble enters the flow field with a second bubble in train. As the two bubbles coalesce (8.62 ms) to form a larger bubble, the transient heat transfer coefficient registers a sharp change and the peak value nearly doubles that of a single bubble pass. This enhanced local heat transfer rate is a result of the formation of a local dryout spot. The local conditions leading to dryout, where a high rate of heat transfer occurs across a rapidly evaporating thin liquid layer, forces a sharp decrease in temperature (8.62 – 12.00 ms). As this dryout spot grows and the three-phase contact line sweeps away from the thermistor, the local heat transfer rate reduces drastically and the temperature rises back up to its quasi-steady, liquid-only value as the coalesced bubble is swept downstream (12.32 – 15.86 ms).

The magnitude of the peak transient heat transfer coefficient of $19,000 \text{ W/m}^2\text{-K}$, computed for the bubble pass (Figure 12), matches closely with the prediction made in the preceding section of $18,000 \text{ W/m}^2\text{-K}$. The enhancement caused by the bubble coalescence, augments the peak value of the local heat transfer coefficient to $32,000 \text{ W/m}^2\text{-K}$. The localized evaporation across the presumably very thin (and depleting) liquid film increases the local heat transfer rates to nearly 4 times its time averaged value ($32,000$ vs. $7,800 \text{ W/m}^2\text{-K}$).

Figure 12: Thermal interactions observed following a bubble coalescence event. The coalescence creates a larger bubble and also a thin liquid film underneath. The evaporation of this liquid film leads to a very high local heat transfer coefficient. Experimental conditions: heat flux = $5.2 \text{ W}/\text{cm}^2$, mass flux = $200 \text{ kg}/\text{m}^2\text{-s}$, and $\bar{h} = 7,800 \text{ W}/\text{m}^2\text{-K}$.

5.3 Slug flow

The periodic rise and fall in local heat transfer accompanying the flow of vapor slugs is depicted in Figure 13. Starting from the nose of the slug, enhanced heat transfer relative to the average (--- in Figure 13) was observed for the majority of the slug length (0.00 – 4.60 ms and 14.62 – 19.05 ms for slug 1 and 2, respectively). The low thermal resistance across the thin liquid film under the slug appears to be the dominant interaction in this regime. In fact, the local

heat transfer rates are only impeded by the occurrence of a local dryout (5.10 and 19.72 ms). These observations seem to agree with the *three-zone* flow boiling model [43,44] but appear to be contrary to those reported by Freystein et al. [31], who observed a near-zero heat transfer under vapor slugs. While direct comparison across flow boiling experiments is difficult, the differences in operating scales—mini vs. microchannels (this work), could be a possible reason. Experiments aimed at understanding this scale (micro – mini) transition and its effect on local heat transfer in the slug flow regime are currently underway.

Compared to the time averaged heat transfer coefficient of $14,500 \text{ W/m}^2\text{-K}$, at all instances prior to the local dryout, the transient heat transfer coefficient is enhanced by thin film evaporation. Figure 13 also shows the effect of a single bubble (11.22 ms) that occurs between two relatively large vapor slugs. The peak transient heat transfer coefficient of the thin film evaporation process is nearly twice that of the single bubble pass, explaining the higher time averaged heat transfer coefficient ($14,500$ vs. $7,800 \text{ W/m}^2\text{-K}$).

Figure 13: Transient measurements in the slug flow regime. Observations indicate that the heat transfer is dominated by thin film evaporation underneath the vapor bubbles. Continued evaporation leads to the formation of a local dryout which forces the temperature back up to its quasi-steady value. Experimental conditions: heat flux = 6.3 W/cm^2 , mass flux = $200 \text{ kg/m}^2\text{-s}$, and $\bar{h} = 14,500 \text{ W/m}^2\text{-K}$.

5.4 Dryout and onset of CHF

The temperature history accompanying annular flow regime with prolonged periods of wall dryout is presented in Figure 14. Three distinct stages were observed in a typical cycle:

- 1) Liquid rewetting (0.00 – 4.99 ms)
- 2) Liquid film evaporation (4.99 – 7.99 ms)
- 3) Dryout (7.99 – 18.98 ms)

Intermittent rewetting by the liquid phase was found to have a fleeting influence on the local heat transfer. Situations were occasionally encountered where the nucleate boiling was temporarily reestablished (2.66 – 4.99 ms). The bulk of the heat transfer over the entire cycle was found to occur as the liquid film evaporates. As this liquid film dries out (7.99 ms), wall superheats of 15-20 K were routinely encountered. It can be expected that overall heat transfer coefficient is heavily influenced by the duration of the near-zero heat transfer characterizing dryout. The Leidenfrost effect was routinely observed as several fragments of the remnant liquid film were found to skitter across the dry wall (see frames at 14.31, 18.98, and 22.21 ms mark).

Figure 14: Results for the intermittent dryout and rewetting that was observed in the annular flow regime. Experimental conditions: heat flux = 8.3 W/cm^2 , mass flux = $200 \text{ kg/m}^2\text{-s}$, and $\overline{h} = 4700 \text{ W/m}^2\text{-K}$.

6 Summary

Spatiotemporally resolved heat transfer measurements provide valuable insight into the fundamental mechanisms affecting flow boiling at the microscale. A strong case for time resolved temperature measurements of the liquid phase also emerges from considerations presented in this work. Conjugate conduction-convection effects in the substrate are pivotal and dictate the response of the sensor, requiring the use of computational techniques to correlate measured temperatures to transient convective heat flux and heat transfer coefficient data. It was

demonstrated that the sensor dynamics in response to a local change in heat transfer can be modeled accurately as a 1-D system for events at the sub-kHz scale if the area of influence of this local change is sufficiently large. Uncertainty considerations for the data processing technique as well as recommendations to minimize uncertainty were also discussed. Multiple flow regimes including bubbly, slug, and annular flow were studied and the transient evolution of the local heat transfer was examined. Thin film evaporation emerges as an important mechanism that was observed across all flow regimes at a mass flux of $200 \text{ kg/m}^2\text{-s}$. It is interesting to note that local dryout was also observed across the flow quality spectrum and that a condition leading to dryout, i.e., the complete evaporation of the liquid film at the three-phase contact line, was characterized by a high heat transfer coefficient.

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