DEVELOPMENT OF AN EXPERIMENTAL METHOD FOR FLOW-BOILING HEAT TRANSFER IN MICROCHANNELS

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In this paper, we report on an experimental method, supported by a numerical conduction model, to determine the fundamental mechanisms controlling flow-boiling heat transfer processes, such as during bubble ebullition. This is achieved by synchronizing high-speed visualization with surface temperature measurement and using the numerical model to infer the heat transfer coefficient from the surface temperature measurements. A slip coefficient, $S$, is defined and provides a quantitative measure of the effect of conduction heat transfer in typical flow-boiling experiments in microchannels. To demonstrate the method, three high-speed experimental measurements are detailed. Surface temperatures at high frequencies ($O(10 \text{ kHz})$) are obtained with micron-sized thermistors; boiling events are simultaneously visualized and used in conjunction with transient temperature measurements and the $S$ coefficient to infer processes controlling heat transfer in a microchannel. The results demonstrate that microdomains formed by high-thermal-conductivity substrates, including silicon and copper, cannot be used to reveal transient processes at the microscale. Even results obtained using low-thermal-conductivity materials such as Pyrex and Benzocyclobutene (BCB) require conduction numerical analysis in the solid structure to decouple the convection and conduction processes.

KEY WORDS: boiling and evaporation, measurement and instrumentation, nano-/microscale measurement and simulation, NEMS/MEMS, two-phase/multiphase flow

1. INTRODUCTION

Phase change heat transfer in microdomains is a very active research area. However, much is still unknown about the fundamental mechanisms controlling the various processes, such as heat transfer during bubble ebullition, slug flow, thin film evaporation during convective boiling, and critical heat flux conditions. This is primarily because experimental measurements are challenging at the microscale and are not fully mature.

Transient processes associated with the bubble ebullition cycle, slug frequency, and the like, accelerate with diminishing length scales, bringing up characteristic heat transfer and fluid flow frequencies. Primary time constants of $O(1 \text{ kHz})$ and even $O(10 \text{ kHz})$ are typical in microdomains. Conduction/convection-conjugated effects are significant and can hinder attempts to accurately quantify convection heat transfer. Heat losses are also a major concern, and proper measures need to be taken to minimize their effect. For these reasons, flow-boiling heat transfer in microdomains is not fully understood.

At the core of this issue is the thermal conductivity and thermal diffusivity of the material used to form microdomains. Experimental studies to date, with a few exceptions, have used high-thermal-conductivity materials (e.g., silicon, copper,) to obtain important heat transfer characteristics, such as heat transfer coefficient, onset of nucleate boiling, and critical heat flux conditions. Although heat losses have been noted and estimated in many studies, there has been limited consideration, or at least explicit mention, of the full conjugated effects. As demonstrated in this paper, only spatial and time average heat transfer coefficients and heat flux can be measured with microchannels made from high-thermal-conductivity materials. Local, transient measurements using commercially available low-thermal-diffusivity and low-thermal-conductivity materials are possible with limitations.
NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$D_{h2}$</td>
<td>Diameter corresponding to the area undergoing a change in the Nusselt number (m)</td>
</tr>
<tr>
<td>$D_h$</td>
<td>Hydraulic diameter (m)</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient (W/m²·K)</td>
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<tr>
<td>$h_1$</td>
<td>Heat transfer coefficient before the step change ($t &lt; 0$) (W/m²·K)</td>
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<tr>
<td>$h_2$</td>
<td>Heat transfer coefficient after the step change ($t &gt; 0$) (W/m²·K)</td>
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<tr>
<td>$k_f$</td>
<td>Thermal conductivity of fluid (W/m-K)</td>
</tr>
<tr>
<td>$k_h$</td>
<td>Thermal conductivity of heater (W/m-K)</td>
</tr>
<tr>
<td>$k_s$</td>
<td>Thermal conductivity of substrate (W/m-K)</td>
</tr>
<tr>
<td>$L_{char}$</td>
<td>Characteristic length scale (m)</td>
</tr>
<tr>
<td>$Nu_1$</td>
<td>Nusselt number at $t &lt; 0$ (-)</td>
</tr>
<tr>
<td>$Nu_2$</td>
<td>Nusselt number at $t &gt; 0$ (-)</td>
</tr>
<tr>
<td>$Nu_{2meas}$</td>
<td>Nusselt number obtained directly from experiments without use of numerical results (-)</td>
</tr>
<tr>
<td>$q''$</td>
<td>Heat flux (W/m²)</td>
</tr>
<tr>
<td>$S$</td>
<td>Slip coefficient (-)</td>
</tr>
<tr>
<td>$t$</td>
<td>Time after step change in heat transfer coefficient (s)</td>
</tr>
<tr>
<td>$T_f$</td>
<td>Fluid temperature (K)</td>
</tr>
<tr>
<td>$T_s$</td>
<td>Surface temperature (K)</td>
</tr>
<tr>
<td>$\alpha_h$</td>
<td>Thermal diffusivity of heater (m²/s)</td>
</tr>
<tr>
<td>$\alpha_s$</td>
<td>Thermal diffusivity of substrate (m²/s)</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>Temperature difference between surface and fluid (K)</td>
</tr>
<tr>
<td>$\Delta T_{sub}$</td>
<td>Subcooled temperature of fluid (K)</td>
</tr>
<tr>
<td>$\tau_s$</td>
<td>Time constant (s)</td>
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Here we report on an analysis demonstrating the inability of common high-thermal-diffusivity materials to infer important transient heat transfer processes on a microscale heated surface. Other, more suitable materials, such as Pyrex and benzocyclobutene polymer (BCB), are also examined and their potential advantages and shortcomings are detailed. Experimental results are then introduced and methods that resolve high-frequency heat transfer events are demonstrated.

2. BACKGROUND AND THEORETICAL CONSIDERATIONS

To better understand experimental challenges at the microscale, consider an infinitesimally thin constant power heater positioned on a substrate having thermal diffusivity $\alpha_s$ and thermal conductivity $k_s$, which is in direct contact with a fluid. A minuscule, perfect (no thermal mass) and precise thermistor (i.e., one that instantaneously measures the local surface temperature) is placed directly on top of the heater. The flow undergoes a sudden temporal change that lasts for a period of $\tau_s$, affecting an area with a diameter $D_{h2}$ directly above the thermistor. This corresponds to a rapid change in the local heat transfer coefficient.

The convective heat transfer equation,

$$ q'' = h_i (T_s - T_f), $$

suggests that for a constant heat flux heater, $q'' = \text{constant}$, on a perfect insulator (i.e., $k_s = 0$), all of the heat is directly dissipated into the fluid. If the fluid temperature, $T_f$, is known, the instantaneous heat transfer coefficient, $h_i$, can then, without reservation, be directly calculated using the surface temperature measurement, $T_s$, obtained through the thermistor.

For a substrate with high thermal conductivity, a direct link, without accounting for conduction, between the thermistor measurements and the heat transfer coefficient cannot be performed—conduction heat transfer must be accounted for before the heat transfer coefficient can be inferred. If the change is confined to a small region (i.e., $D_c$ is small), the heat is conducted effectively to or from (depending on the relative magnitude of the heat transfer coefficients) the region adjacent to, but not under, the area undergoing the heat transfer coefficient change. Under this
condition, the local heat losses do not allow the thermistor to register the full effect of the change. Because of the high thermal diffusivity, the change propagates quickly (i.e., much faster than the convective process).

The scenario just described is common in microchannels carved from high-thermal-conductivity materials. In fact, the transient signature of many typical processes cannot be effectively captured in microdomains made of material such as silicon and copper—high-thermal-conductivity materials are inadequate for local [i.e., O(10 \, \mu m) and even O(100 \, \mu m)] heat transfer measurements. For this reason, these materials cannot be used to measure important transient heat transfer phenomena associated with processes such as bubble nucleation.

Low-thermal-conductivity materials, such as Pyrex and BCB, can remedy the situation just presented to a certain extent. The opportunities and limitations of such an approach are examined next.

### 2.1 Slip Coefficient

To determine the relative magnitude of a true change in the heat transfer coefficient and the relative magnitude registered through the thermistor, a slip coefficient, \( S \), is defined:

\[
S = \frac{\text{Nu}_{\text{meas}} - \text{Nu}_1}{\text{Nu}_2 - \text{Nu}_1}
\]

where \( \text{Nu}_{\text{meas}} \) is the Nusselt number deduced directly from the thermistor’s temperature measurement, \( T_s \), according to

\[
\text{Nu}_{\text{meas}} = \frac{h_{\text{meas}} L_{\text{char}}}{k_f} = \left[ \frac{q''}{(T_s - T_f)} \right] \left( \frac{D_h}{k_f} \right)
\]

\( \text{Nu}_1 \) and \( \text{Nu}_2 \) are the true Nusselt numbers before the change in the heat transfer coefficient (\( @t > 0 \)) and after the change (\( @t > 0 \)), respectively. Note that for a thermal conductor substrate, the heat flux to the fluid may differ from that generated by the heater, \( q'' \), because a fraction of the heat is locally conducted into or out of the surface region.

The measured Nusselt number, \( \text{Nu}_{\text{meas}} \), adopts a naïve approach in which all of, and only, the heat generated in the heater is locally transferred to the fluid. Because conduction comes into play, \( \text{Nu}_{\text{meas}} \) may significantly differ from the true Nusselt number associated with the change at \( t = 0 \) and the one being sought (\( \text{Nu}_2 \)). In other words, the slip coefficient gives the ratio between the experimental measured change in the Nusselt number and the true change. It varies from 0 to 1—an \( S \) value approaching unity allows a direct link between the thermistor measurement and the Nusselt number through Eq. (3); low values require careful attention to the conduction/convection-conjugated effects in order to resolve the true Nusselt number. Equation (2) can be used for such instances.

### 2.2 Numerical Model: Slip Coefficient

To estimate the slip coefficient for a range of scenarios pertinent to microchannel flow, a transient numerical model using ANSYS was developed (Fig. 1) and executed for candidate substrate materials silicon (\( k_s = 148 \, \text{W/m-K}; \alpha_s = 8.921 \times 10^{-5} \, \text{m}^2/\text{s} \)), Pyrex (\( k_s = 1.4 \, \text{W/m-K}; \alpha_s = 7.466 \times 10^{-7} \, \text{m}^2/\text{s} \)), BCB (\( k_s = 0.202 \, \text{W/m-K}; \alpha_s = 1.545 \times 10^{-7} \, \text{m}^2/\text{s} \)), and air (\( k_s = 0.026 \, \text{W/m-K}; \alpha_s = 2.223 \times 10^{-5} \, \text{m}^2/\text{s} \)). A solid medium with dimensions of 400 \times 400 \times 500 \mu m was discretized into 115,000 finite elements. (A grid independence evaluation was carried out and it was found that the number of finite elements used in this study was sufficient for accurate calculation.) The model consisted of a 100-nm-thick titanium (\( k_h = 21 \, \text{W/m-K}; \alpha_h = 8.929 \times 10^{-6} \, \text{m}^2/\text{s} \)) heat generation element in direct contact with the fluid boundary. A constant temperature boundary condition was set on the lower (horizontal) surface of the domain, the value of which was obtained by auxiliary calculations involving heat exchange by natural convection. The lateral surfaces of the computational domain were subjected to symmetry (insulation) boundary conditions. For time \( t < 0 \), a steady-state calculation was executed that involved a fully developed (thermally and hydraulically, with \( \text{Nu} = 5.384 \)) flow in a microgap with a constant heat flux boundary condition on a single (lower) side. At time \( t = 0 \), the heat transfer coefficient over an area of diameter \( D_{h2} \) (\( D_{h2} \) varied 100–500 \mu m) was changed to \( h_2 \) and the transient surface temperature, \( T_s \), at the center of the segment was obtained. The measured Nusselt number was then calculated through Eq. (3), which was then introduced into Eq. (2) to calculate the slip coefficient, \( S \). The initial heat transfer...
coefficient, $h_1$, was set to 12,490 W/m$^2$·K, and $h_2$ was set at multiples of $h_1$ such that $h_2 = 1.5h_1$, $2h_1$, $5h_1$, $7h_1$, and $10h_1$.

2.2.1 Slip Coefficient: Silicon

Silicon has often been used as a substrate material for microdevice configurations in the study of both steady-state and transient phase change phenomena (Tsai and Lin, 2002; Chen and Garimella, 2006; Wang et al., 2007; Sathyamurthi and Banerjee, 2009; Krishnamurthy and Peles, 2010). Results for silicon with $h_2 = 2h_1$ are shown in Fig. 2. Clearly, and as expected, the $S$ value is very low (less than 7.5%) even when the change spans a large area (e.g., $D_{h_2} = 500\, \mu$m) for large time intervals (i.e., at steady state, which occurs at 1–10 ms). Note that for a characteristic frequency of $O(1 \, \text{kHz})$—a typical bubble ebullition frequency in microdomains—the conduction process does not reach steady state and the $S$ value is even smaller. This is because temperature gradients in the solid are larger than they are during steady state. The increase in the $S$ value with diameter is a result of the diminishing dominance of radial conduction (i.e., conduction of the fluid from regions of low heat transfer coefficients to regions of high heat transfer coefficients) radial conduction increases roughly linearly with diameter, but the heat transfer rate increases as $L_{\text{char}}^2$.

![FIG. 1: Schematic of the computational domain used for calculating the slip coefficient.](image1.png)

![FIG. 2: Slip coefficient for silicon for diameters, $D$, of 100–500 $\mu$m.](image2.png)
For a 100-μm bubble with a frequency of O(1 kHz), which is typical of bubbles in microchannels during nucleate flow boiling, an $S$ value of ~5% is predicted. As an example, consider the following conditions, which are characteristic of water flow boiling in a microchannel: $q'' = 100 \text{ W/cm}^2$, $h_1 = 20,000 \text{ W/m}^2\text{-K}$, and $h_2 = 40,000 \text{ W/m}^2\text{-K}$. The heat transfer coefficient that is registered is only $h_{2,\text{meas}} = 21,000 \text{ W/m}^2\text{-K}$ (i.e., 5% of 20,000 W/m²·K plus $h_1 = 20,000 \text{ W/m}^2\text{-K}$). This corresponds to a surface-to-fluid temperature measurement of 50°C before the change in the heat transfer coefficient and 47.6°C after the change (i.e., $\Delta T = 2.4^\circ\text{C}$). For a perfect insulator, the temperature should have been 25°C (i.e., $\Delta T = 25^\circ\text{C}$). An uncertainty of $\pm 1^\circ\text{C}$ propagates to an uncertainty in the heat transfer coefficient of 42% for silicon, assuming that all other measurements are precise. For a perfect insulator, the heat transfer coefficient uncertainty for this case is only 4%.

2.2.2 Slip Coefficient: Pyrex

Pyrex is a practical substrate that has been used in several studies on flow boiling in microchannels (Demiray and Kim, 2004; Myers et al., 2005; Wang and Chen, 2008; Houshmand and Peles, 2013, 2014a,b; and Rao et al., 2014). It is employed primarily because it has a relatively low thermal conductivity and a low thermal diffusivity and therefore the $S$ coefficient is much higher than for silicon (Fig. 3). However, for periodic events with a characteristic frequency of O(1 kHz), the $S$ value can still be quite low.

Information about a sudden change in the heat transfer coefficient gradually propagates into the solid medium. For short time periods, the solid temperature varies only in the immediate vicinity of the change, and the thermistor temperature is unaffected by the diameter, $D_{h_2}$. As time elapses, information about the change propagates outside the region confined by $D_{h_2}$, deeper into the substrate. As a result, the $S$ value begins to depend on $D_{h_2}$. For small $D_{h_2}$, the conduction into or out of the region confined by $h_2$ is more considerable than that for large $D_{h_2}$; thus, the $S$ value is larger for larger $D_{h_2}$.

Several practical observations about the applicability of nucleate and convective flow-boiling measurements can be inferred from Fig. 3. For events like partial periodic dry-out, our observations, discussed later in this paper and presented in detail in Rao et al. (2014) and Peles and Rao (2014), suggest that the characteristic time constant is O(100 Hz) with a typical dry area of 200–500 μm. Thus, a thermistor can register approximately 50% of the phenomenon. For higher frequencies of 1 kHz and above, such as near a growing bubble [e.g., 50–200 μm at frequencies

![Graphs](https://via.placeholder.com/150)

**FIG. 3:** $S$ coefficient as a function of time for Pyrex: (a) for range of areas undergoing a change in the heat transfer coefficient; (b) for different changes in the Nusselt number. For a short time following the change, the effect is confined to a small region and the diameter of the area undergoing the change does not affect thermistor response. This is because information about changes occurring in the heat transfer has not propagated deep enough into the substrate. Also, the slip coefficient is marginally dependent on the Nusselt number.
of O(1 kHz)], only a small fraction of the event is captured by a perfect thermistor. It is interesting to note that the $S$ value is weakly dependent on the change in the heat transfer coefficient, as shown in Fig. 3, making the results applicable to a large variation in heat transfer coefficient changes (e.g., the $S$ value for $h_2 = 1.5h_1$ can be used to infer results for $h_2 = 5h_1$).

2.2.3 Slip Coefficient: BCB

A less explored material with even lower thermal conductivity and diffusivity is BCB (Moghaddam et al., 2007). Because of its low $k_s$ and $\alpha_s$ and the limitation of Pyrex discussed previously, BCB is a desirable material for fundamental heat transfer studies.

Results for $S$ are shown in Fig. 4. The $S$ value is modestly higher for BCB than for Pyrex. For a slow event (e.g., 100 Hz), the temperature recorded by the thermistor captures almost the entire event, but for fast processes (e.g., larger than 1 kHz) a smaller fraction of the event is registered at the wall.

2.2.4 Slip Coefficient: Air

To explore the limits of the influence of material properties, calculations were performed with air as the substrate material. Such a configuration can be potentially realized as a thin thermistor suspended on an air gap in the microchannel. (Note that heat is transferred only by conduction through the air gap because advection is suppressed by the low Rayleigh number.) The predictions for $S$, shown in Fig. 5, are significantly higher at time scales relevant to the physics, dictating the phase change heat transfer at the microdomains (10 kHz and lower). In fact, the $S$ value approaches unity for most of the scenarios commonly encountered at the microscale.

2.2.5 Summary of Material Analysis

Silicon, copper, and the like, are not practical materials for transient and local measurements in microchannels. Temperature measurements with a Pyrex substrate can provide valuable insight into various heat transfer processes, but conduction-/convection-conjugated effects need to be carefully analyzed. Without proper consideration of conduction effects, results can be misleading, at least quantitatively. To some extent, BCB is a better material than Pyrex, but it can introduce additional practical problems, among them microfabrication complexity and an inability to tolerate partial dry-out (quick burnout). The configuration of a thermistor suspended in an air gap demonstrates an almost ideal performance for transient thermal measurements in a microchannel with the slip coefficient approaching unity.

![Graph of $S$ vs. time for different D values](image)

**FIG. 4:** $S$ coefficient for BCB. Because the $S$ value is higher for BCB than for Pyrex, it is a better material for decoupling conduction from convection.
FIG. 5: $S$ coefficient for air. Extremely large $S$ values are predicted with air as the substrate. The low thermal conductivity and diffusivity of air allow decoupling of conduction and convection phenomena at the microscale. However, effectively forming a heater on an air gap is a challenging problem that has yet be solved.

the kHz time scale. However, significant complexities can be introduced with such configurations, including the issue of mechanical robustness.

3. EXPERIMENTAL RESULTS

The numerical results are now leveraged to demonstrate a method in which measurements of transient heat transfer coefficients for flow-boiling experiments can be obtained. Both fast processes (e.g., bubble ebullition) and slow processes (e.g., thin liquid film evaporation accompanying slug flow) are demonstrated.

3.1 Experimental Setup

A dedicated experimental setup to measure the transient heat transfer coefficient during flow boiling was built. In addition, a microchannel was microfabricated and instrumented with an array of built-in thermistors.

3.1.1 Microdevice

To study nucleate flow boiling, a microchannel with multiple heaters and temperature sensors was designed and fabricated (Fig. 6). This configuration was identical to the one used in Rao et al. (2014) and Peles and Rao (2014), which adapted several key design elements from previous work by Houshmand and Peles (2013, 2014a,b). Subcooled liquid was allowed into a 210-μm-deep, 1.5-mm-wide, 22-mm-long microchannel with embedded titanium heaters and thermistors to dissipate heat to the working fluid and to measure the local wall temperature. The thermistors were located on top of the heater layer and enabled temperature measurement within the substrate, which was then used to infer the spatial and temporal variations of the local boiling heat transfer coefficient.

The microdevice consisted of two processed Pyrex wafers ($P_1$ and $P_2$) and a double-sided adhesive-coated vinyl layer that bonded them together. The microfabrication process resulted in three 100-nm-thick, 1-mm-wide resistive heaters on the 500-μm Pyrex substrate ($P_2$). Six 75-nm-thick thermistors were formed on top of a 0.5-μm layer of silicon dioxide, which was deposited directly on the heater. A data acquisition (DAQ) system and power supplies were wired to the thermistors via gold-plated, spring-loaded probe pins, and an inlet port and an outlet port, both 1-mm in diameter were drilled. In addition, 18 holes were drilled through the second Pyrex wafer ($P_1$) to enable access to the contact pads on the substrate. Finally, the device was bonded. For complete details of microfabrication and microdevice assembly see Rao et al. (2014).
3.1.2 Experimental Loop

A schematic of the experimental setup is shown in Fig. 7. An open-loop system was used for experiments with water and a closed system was used for coolant HFE-7000. The open-loop system, detailed in Rao et al. (2014), consisted of data acquisition components, a high-speed camera, a pressure tank, a rotameter, a 5-μm filter, a resistive preheater, an inlet restrictor, and auxiliary components. A pressure transducer and a T-type thermocouple were positioned just before the inlet of the microchannel test section to estimate the inlet fluid temperature and pressure.

The test section (Fig. 8) was housed in a special package that allowed the microdevice to interface with the large-scale world. An inverted microscope (Zeiss Observer-Z1) with a high-speed camera (Phantom 4.2) was used to visualize the flow phenomena. Synchronization between temperature (resistance) measurements and the high-speed...
FIG. 8: Package developed to house the microdevice and interface electrical measurements.

camera was achieved by a high-accuracy pulse generator to simultaneously trigger the camera and the analogue measurements.

3.1.3 Experimental Uncertainty

All uncertainties were calculated as described in Taylor and Kuyatt (1994). The unbiased uncertainty associated with the measured temperature was estimated by a detailed uncertainty propagation analysis and estimated to vary ±0.6–±1.2°C.

3.2 Experimental Results and Discussion

Estimation of the transient heat transfer coefficient associated with a single event cycle, such as nucleation or bubble coalescence and growth, is now demonstrated using the slip coefficient presented in this work. Considering a step change in the Nusselt number, if $S$ and $\text{Nu}_{\text{meas}}$ are known through numerical calculations and experimental measurements at time $t$ ($t = 0$ corresponds to the instance of the step change), $\text{Nu}_2$ can be calculated according to

$$\text{Nu}_2 = \frac{\text{Nu}_{\text{meas}} - \text{Nu}_1 (1 - S)}{S}$$  \hspace{1cm} (4)

A more computationally intensive technique to resolve the transient heat transfer coefficient was recently proposed by Rao et al. (2014), who used a finite-element framework to examine variations in the transient heat transfer coefficient based on experimental temperature histories associated with boiling cycles. By matching the heat flux and fluid temperature to experimental data, the convective heat transfer coefficient in the computational model was perturbed in magnitude over suitable spatial and temporal spans to reflect the high-speed visualization accompanying the boiling event. By overlay of experimental and computational temperature histories, predictions were made for different types of bubble-driven events and interactions.

Results from Rao et al. (2014) were compared with estimates of the cycle heat transfer coefficient obtained using the definition of the slip coefficient, $S$, according to Eq. (2). To enable a direct comparison between the two approaches, the slip coefficient was computed for the three different scenarios presented in Rao et al. (2014). The thermal boundary conditions—namely, the thermal energy generation rate and the initial heat transfer coefficient, $h_1$ (or $\text{Nu}_1$)—necessary to calculate $S$ were derived from the same experimental data set to allow a direct comparison between the two predicted $h_2$ values.
3.2.1 Case 1: Single Bubble Nucleation and Growth—O(1 kHz)

Synchronized high-speed video and thermistor temperature measurements for the nucleation, growth, and departure of single bubble at a heat flux of 70 W/cm² and a mass flux of 500 kg/m²·s were recorded. A diameter, $D_{h2}$, of 300 µm was assumed based on the average diameter of the bubble as seen in the high-speed images. The difference between the instantaneous temperature and the initial temperature was represented by $\Delta T$. Figure 9 shows a composite of the recorded high-speed visualization and experimental data (temperature). A rapid change in the surface temperature is registered at time $t = 0.2$ ms (Frame B) corresponding to the nucleation of a bubble in close vicinity to the thermistor (red line). To use the $S$-curve, a characteristic time is required; in this example, the time elapsed between Frame B and Frame E was used (i.e., 0.4 ms). With the aid of Fig. 10, it can be seen that, after 0.4 ms of a step change, the predicted slip coefficient at the experimental heat flux is $S = 0.12$. Using Eq. (4), the heat transfer coefficient ($h_2$) at $t > 0$ calculated using the slip coefficient for bubble nucleation and departure is $91 \pm 15$ kW/m²K. This represents a dramatic increase from the quasi-steady-state value ($h_1$) of $20 \pm 4$ kW/m²K. The estimate idealizes the change in the heat transfer coefficient as a step function, which compares well with Rao et al.’s (2014) peak result of $90 \pm 20$ kW/m²K and an average of $52 \pm 12$ kW/m²K.

3.2.2 Case 2: Postdeparture Bubble Interactions and Growth—O(1 kHz)

Considering a bubble ($D = 300$ µm) that has already departed its nucleation site, Fig. 11 presents the sequence for the approach (relative to the thermistor) and growth of such a bubble in the subcooled nucleate flow-boiling regime. The bubble exhibits an increase in diameter from Frame A to Frame D. The associated temperature measurements at the

![Image of high-speed footage and synchronized corresponding surface temperature inferred from the thermistor represented by a red line in the image sequence. The sudden reduction in recorded temperature is due to the exchange of thermal energy between the heated wall and the nucleating bubble for vapor generation. Experimental conditions: heat flux of 70 W/cm², mass flux of 500 kg/m²·s; $\Delta T_{sub} = 13$°C; temperature acquisition frequency = 10 kHz; video acquisition frequency = 10 kHz.](image)

**FIG. 9:** High-speed footage and synchronized corresponding surface temperature inferred from the thermistor represented by a red line in the image sequence. The sudden reduction in recorded temperature is due to the exchange of thermal energy between the heated wall and the nucleating bubble for vapor generation. Experimental conditions: heat flux of 70 W/cm², mass flux of 500 kg/m²·s; $\Delta T_{sub} = 13$°C; temperature acquisition frequency = 10 kHz; video acquisition frequency = 10 kHz.
FIG. 10: $S$ values for Case 1 (red circle), Case 2 (blue square), and Case 3 (green triangle) corresponding to experimental conditions using a Pyrex substrate with heat flux of 70 W/cm$^2$, $D_{h2}$ = 300 μm, and mass flux of 500 kg/m$^2$s. The time dependence of the slip parameter is assumed to be the same for both cases because the experiments were performed at almost the same heat flux. The inlet subcooling (i.e., mass quality/subcooling of the fluid at the thermistor) was identical in all experimental cases ($\Delta T_{sub} = 13^\circ$C).

FIG. 11: High-speed footage and synchronized corresponding surface temperature for Case 2. The enhanced heat transfer rate accompanying the growth of a bubble leads to reduction in the measured local surface temperature, as witnessed by measurements in Frames B through D. Experimental conditions: heat flux of 70 W/cm$^2$, mass flux of 500 kg/m$^2$s; $\Delta T_{sub} = 13^\circ$C; temperature acquisition frequency = 10 kHz; video acquisition frequency = 10 kHz.
thermistor (red line) indicate a commensurate reduction in the local surface temperature owing to the local increase in the convective heat transfer rate as the vapor bubble grows in size. The local surface temperature then rises back up as the now enlarged bubble is swept downstream (Frames E through G), returning the local conditions to their quasi-steady value. The slip coefficient, computed just prior to the departure of the bubble (Frame D, 0.8 ms), yields a value of 0.37 (See Fig. 10). Equation (4) predicts a heat transfer coefficient ($h_2$) of $40 \pm 9$ kW/m²K for a single cycle for this slip coefficient value, indicating a significant increase from the quasi-steady value ($h_1$) of $20 \pm 4$ kW/m²K.

3.2.3 Case 3: Thin Film Evaporation—Slug Flow

Figure 12 shows the temperature history of the periodic flow structure accompanying elongated vapor bubbles or the slug flow regime. The recorded temperatures indicate a steady decrease in evaporation of the liquid microlayer pinned under the slug. This phase change process further reduces the thickness of the liquid film ($\delta_{\text{film}}$), leading to an increase in the local heat transfer coefficient ($h \rightarrow k_{\text{film}}/\delta_{\text{film}}$). The slip parameter (Fig. 10) computed for the flow conditions allows calculation of a representative estimate of the heat transfer coefficient for the entire slug flow sequence. According to Eq. (3), an $S$ value of 0.57 is predicted at Frame E (5.8 ms after the approach of the vapor slug or the onset of the step change in the local heat transfer coefficient), which yields $h_2 = 33 \pm 9$ kW/m²K. The large cycle times associated with slug flow and the relative change in the local heat transfer coefficient ($h_1 = 20 \pm 4$ kW/m²K) indicate the effectiveness of the heat transfer process concomitant with this flow pattern.

4. SUMMARY

A new experimental/numerical approach to infer key boiling heat transfer characteristics in microdomains was introduced. The results of a simple conduction numerical model were used to quantify a new parameter termed the slip parameter.

![FIG. 12: Temperature and image sequence for the elongated bubble (slug) flow regime (data markers for temperature measurements are shown for every three measurements). A sharp increase in local heat transfer is observed with the reduction in the liquid microlayer trapped under the vapor slug (Frames B through E). The approach of the liquid slug induces a rise in the local temperature (Frames E through G). Experimental conditions: heat flux of 70 W/cm²; mass flux of 500 kg/m²·s; $\Delta T_{\text{sub}} = 13^\circ$C; temperature acquisition frequency = 10 kHz; video acquisition frequency = 10 kHz.](image-url)
Development of an Experimental Method for Flow-Boiling Heat Transfer

397

coefficient. This coefficient was then used in conjunction with advanced instrumentation methods to infer rapid transient processes typical of boiling at the microscale. It was demonstrated that increasing our knowledge of boiling heat transfer in microdomains requires advanced experimental capacity, including advanced microfabrication capabilities, advanced instrumentation, and careful attention to conjugated conduction/convection heat transfer.

Even with insulating materials like Pyrex and BCB, convection heat transfer cannot be resolved without accounting for conduction processes. To fully insulate the convection process, new microfabrication approaches need to be developed such that the effective thermal conductivity of the substrate forming the microchannel is comparable to that of air. Perhaps this can be achieved by development of an innovative microfabrication device in which an air gap is formed between a very thin heater and the main substrate. In the meantime, substrates with low thermal conductivity, such as Pyrex, can be used in experiments to elucidate the transient characteristics of boiling heat transfer. However, for these substrates the slip coefficient obtained in this study must be used. Alternatively, a numerical modeling such as the one presented in this paper needs to be developed.

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