Numerical and Experimental Investigations of Boiling Enhancement in Buoyancy-Driven Microchannels

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NUMERICAL AND EXPERIMENTAL INVESTIGATIONS OF BOILING ENHANCEMENT IN BUOYANCY-DRIVEN MICROCHANNELS

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ABSTRACT
In this study, confinement-driven boiling enhancement trends and experimental data from narrow parallel plate channels are presented and analyzed via comparison with numerical simulations of buoyancy-driven boiling and two phase flow using the commercially-available Fluent CFD software package. An Euler-Euler multiphase approach, known as the volume of fluid (VOF) method, is employed, as bubbles sizes are on the order of the channel dimensions. Numerical results suggest that enhanced natural convection already accounts for a large portion of the unconfined pool boiling heat flux. While the increased buoyancy from large vapor fractions in narrow channels may lead to an order of magnitude increase in channel mass flux, confinement-driven convective enhancement is found to increase the unconfined boiling heat flux by less than 10%. Further, simulated convective enhancement is found to be a maximum for intermediate size channels, in direct contrast to experimental data which show maximum enhancement (500%) for the smallest channels investigated. Experimental results for different channel wall materials suggest an enhancement mechanism highly dependent on boiling surface characteristics.

KEY WORDS: heat transfer, bubble, VOF, multiphase, confinement, Bond number

NOMENCLATURE
\( A \) heat transfer area, \( m^2 \)
\( B_o \) Bond number, Eq. (1)
\( c_p \) specific heat, \( J/kg\cdot K \)
\( D \) diameter, \( m \)
\( f \) bubble departure frequency, \( Hz \)
\( G \) mass flux, \( kg/m^2s \)
\( g \) gravitational acceleration, \( 9.81 \, m/s^2 \)
\( h \) heat transfer coefficient, \( W/m^2K \)
\( h_{fg} \) heat of vaporization, \( J/kg \)
\( \delta \) modified Jacob number, Eq. (5)
\( k \) thermal conductivity, \( W/m\cdot K \)
\( L \) channel length, \( m \)
\( m \) mass flow rate, \( kg/s \)
\( N \) number of bubble nucleation sites
\( P \) pressure, \( Pa \)
\( Pr \) Prandtl number, \( c_p\mu/k \)
\( q'' \) heat flux, \( W/m^2 \)
\( R \) heat transfer enhancement ratio, Eq. (2)
\( R_a \) centerline average surface roughness, \( m \)
\( T \) temperature, \( ^\circ C \) or \( K \)
\( t \) time, \( s \)
\( W \) channel width/depth, \( m \)

Greek symbols
\( \beta \) thermal expansion coefficient, \( 1/K \)
\( \delta \) channel spacing, \( m \)
\( \phi \) surface-liquid interaction parameter, Eq. (9)
\( \mu \) dynamic viscosity, \( Pa\cdot s \)
\( \rho \) mass density, \( kg/m^3 \)
\( \Theta \) roughness parameter, Eq. (10)
\( \theta \) contact angle, \( ^\circ \)
\( \sigma \) surface tension, \( N/m \)

Subscripts
b bubble
f liquid phase
g vapor phase
g bubble growth
h heated surface
in inlet
sat saturated
w waiting

INTRODUCTION
Boiling channels in which bubble sizes are on the order of the channel dimension are expected in thermal management applications ranging from direct liquid cooling of IC die stacks to microchannel heat sinks to fuel cell heat exchangers. Vapor generated during boiling activity on heated channel walls can increase fluid buoyancy and circulation, and enhanced thermal performance relative to undisturbed pool boiling is often observed. Unlike traditional forced flow boiling situations, in the case at hand channel flow arises from density differences between the two phase mixture in the channel and the ambient liquid. In the literature it is almost exclusively assumed that forced flow boiling heat transfer is governed by a combination of forced convection and nucleate boiling mechanisms [1]. When applied to buoyancy-driven flows, the lack of \textit{a priori} knowledge of the mass flow rate makes application of forced flow correlations problematic. Further, the majority of forced flow studies employ high mass flow rates and highly...
subcooled liquid. Empirical and theoretical investigations of buoyancy-driven microchannel flows have considered additional enhancement mechanisms of evaporation of the thin film between compressed bubbles and the heated wall as well as transient conduction to the liquid phase in the bubble slug wake. While it is difficult to separate and evaluate numerous enhancement effects experimentally, the dependence of channel mass flux on a variety of heat transfer mechanisms may all be investigated individually and in concert via numerical simulation.

In this study numerical simulations have been pursued to explore to what degree enhanced convection of the liquid phase may contribute to experimentally observed heat transfer enhancement in buoyancy-driven microchannels. Given that bubble motion can affect the motion of the liquid phase as much as the liquid phase affects bubble motion, an Euler-Euler multiphase approach, such as the volume of fluid (VOF) method [2][3] is required. While the literature contains many direct numerical simulation studies of idealized individual bubble nucleation and motion in unconfined pools (see [4]), examples of confined nucleate boiling problems are quite scarce [5]. While high degrees of accuracy are unreasonable to expect at this time, multiphase numerical simulations may be used to probe the order of magnitude of various effects and explore experimentally-observed trends.

Figure 1 shows an illustration of the vertical parallel plate channel geometry considered. The channel length, $L$, is taken to be parallel to gravity. The channel spacing, or the distance between the channel walls, $\delta$, is the key dimension driving confinement effects. For simplicity, only asymmetrically heated channels—heat is dissipated from only one channel wall with the other adiabatic—will be considered, though the modeling approach could be readily extended to symmetrically heated channels.

![Fig. 1 Vertical Parallel Plate Channel Geometry with Nomenclature](image)

**EXPERIMENTAL STUDIES**

Confined boiling has garnered considerable experimental attention over the past half decade. Since the landmark study of Ishibashi and Nishikawa [6], numerous researchers have explored heat transfer enhancement in boiling channels, tubes, and other confined geometries. A detailed literature review of natural convection boiling in vertical, parallel plate channels is presented by [5]. While the body of work is still quite varied, some general observations and conclusions may be drawn and summarized as follows:

- At large channel spacings, boiling behavior proceeds as in unconfined pool boiling (i.e. the limit as $\delta \rightarrow \infty$).
- At progressively smaller channel spacings, increasing heat transfer enhancement is seen in the low heat flux region of the nucleate boiling curve, while the high flux region and, in particular, CHF remain unchanged.
- With still smaller spacings, and/or channel aspect ratios larger than 10, deteriorated heat transfer (relative to pool boiling) is observed at high heat fluxes, and CHF tends to decrease with channel aspect ratio ($L/\delta$) in a roughly hyperbolic fashion [7]. Enhancement continues to increase at low heat fluxes.
- If channel spacing is reduced further, low heat flux enhancement would be expected to peak, after which further reductions in channel spacing would eventually lead to deteriorated heat transfer relative to unconfined pool boiling across the entire heat flux range.

Many researchers have employed the Bond number to relate channel spacing to bubble departure diameter [7].

$$\text{Bo} = \delta^2 \left( \frac{g(\rho_f - \rho_g)}{\sigma_f} \right)$$

(1)

Subtle variations in the usage of this nondimensional parameter grouping include the reciprocal of Eq. (1) [8], its square root [9][10], and the combination square root of the reciprocal [11], though occasionally it is based on the hydraulic diameter ($2\delta$ for a parallel plate channel) instead of the channel spacing, $\delta$. Numerous researchers, including [8–12], identify $\text{Bo} \approx 1$ (as defined above) as generally delineating the transition between confined and unconfined behaviors.

Geisler and Bar-Cohen [12] constructed an experimental system to facilitate the investigation of pool boiling heat transfer in various parallel plate channels, focusing on sub-millimeter spacings. Boiling experiments were conducted for the electronics cooling liquid FC-72 at atmospheric pressure for silicon (20 × 20 mm and 20 × 30 mm) and aluminum (20 × 20 mm) heaters with varying surface roughness. Channel spacing was varied from 20 mm to 0.3 mm, providing channel aspect ratios (length-spacing) from effectively zero to 67. Experiments were performed with uniform, equal heat flux on both channel walls (symmetric heating) as well as with one side heated and the other unpowered (asymmetric heating).
Figure 2 shows boiling curves from [12] for asymmetrically heated polished silicon channels. As channel spacing is decreased below 2 mm (Bo = 7.6), and the heated surfaces begin to interact, boiling curves shift subtly to the left, signifying lower superheats and slightly enhanced heat transfer. With further decreases in channel spacing to 0.7 mm and 0.3 mm (Bo = 0.93 and 0.17, respectively), heat transfer at low heat fluxes is greatly enhanced, while boiling data in the high flux region show deteriorated performance. Geisler and Bar-Cohen [12] did not investigate channel spacings small enough to show a drop-off in enhancement and deteriorated heat transfer in the low flux region.

![Fig. 2: Boiling Curves for Asymmetric 20 × 30 mm Polished Silicon Heater Channels [12]](image)

At heat fluxes ranging from 40 to 60 kW/m² for symmetrically heated channels and 50 to 70 kW/m² for asymmetrically heated channels, boiling surface temperatures increased beyond those observed for unconfined pool boiling, signifying a transition to deteriorated heat transfer performance with channel confinement. CHF decreased with decreasing channel spacing in agreement with the previously-stated trends evident in the literature.

![Fig. 3: Enhancement Ratios, based on Eq. (2), for the Asymmetric Channel Data of Fig. 2](image)

Confinement-driven boiling enhancement (and degradation) is shown as a function of wall superheat in Fig. 3, based on the asymmetric channel boiling curves of Fig. 2. The enhancement ratio shown is defined as the ratio of the channel heat flux to the unconfined pool boiling heat flux at the same surface superheat i.e.

$$ R = \frac{q''(\delta)}{q''(\delta \rightarrow \infty)} = \frac{h(\delta)}{h(\delta \rightarrow \infty)} \quad (2) $$

It is clear from Figs. 2 and 3 that confinement-driven enhancement occurs at wall superheats ranging from 5°C to 18°C. The additional heat transfer contribution above unconfined boiling due to confinement effects is quite significant—greater than 100% over much of this range.

### Numerical Modeling of Convective Enhancement

The VOF method as implemented in the Fluent CFD software (ver. 6.2.16) was used to simulate buoyancy-driven two phase flow and heat transfer in boiling microchannels. Since the focus of this study is the effect of confinement on two phase flow and channel heat transfer, rather than fundamentals of the nucleate boiling process, bubble generation is approximated by a small vapor inlet located in the heated surface. Furthermore, mass transfer between the phases is not included in the analysis, allowing convective effects to be investigated in isolation. Analyses are based on the FC-72 fluid properties shown in Table 1 [13][14]. The density of the vapor phase is assumed constant, while the Boussinesq approximation is used for the liquid phase. The boiling surface is taken to be polished silicon, and the empirical boiling curves shown in Fig. 2 are assumed to apply.

### Table 1: Saturated FC-72 Properties at 1 atm [13][14]

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Saturation Temperature, $T_{sat}$</td>
<td>56.6°C</td>
</tr>
<tr>
<td>Heat of Vaporization, $h_{fg}$</td>
<td>84500 J/kg</td>
</tr>
<tr>
<td>Vapor Phase</td>
<td></td>
</tr>
<tr>
<td>Density, $\rho_g$</td>
<td>13.4 kg/m³</td>
</tr>
<tr>
<td>Thermal Conductivity, $k_g$</td>
<td>0.02 W/m·K</td>
</tr>
<tr>
<td>Specific Heat, $c_{p,g}$</td>
<td>500 J/kg·K</td>
</tr>
<tr>
<td>Dynamic Viscosity, $\mu_g$</td>
<td>0.0000181 Pa·s</td>
</tr>
<tr>
<td>Liquid Phase</td>
<td></td>
</tr>
<tr>
<td>Density, $\rho_l$</td>
<td>1620 kg/m³</td>
</tr>
<tr>
<td>Thermal Conductivity, $k_l$</td>
<td>0.0522 W/m·K</td>
</tr>
<tr>
<td>Specific Heat, $c_{p,l}$</td>
<td>1098 J/kg·K</td>
</tr>
<tr>
<td>Dynamic Viscosity, $\mu_l$</td>
<td>0.0004536 Pa·s</td>
</tr>
<tr>
<td>Thermal Expansion Coefficient, $\beta$</td>
<td>0.001639 1/K</td>
</tr>
<tr>
<td>Surface Tension, $\sigma$</td>
<td>0.00827 N/m</td>
</tr>
<tr>
<td>Contact Angle, $\theta$</td>
<td>1°</td>
</tr>
</tbody>
</table>
Simulation Parameters
Key bubble parameters include the growth time \( t_g \), departure diameter \( D_b \), departure frequency \( f \), and nucleation site density \( N/A \). Given this information, equivalent vapor mass flow rates may be calculated. In general, accurate prediction of these parameters is rather elusive. Available correlations are often limited in their applicability, and statistical distributions of nucleation site and bubble characteristics are rarely considered. However, for the purposes of exploring the order of magnitude of various effects, it is assumed that approximate bubble parameters will suffice.

Bubble Correlations. Tong et al. [15] explored the suitability of a variety of bubble correlations for highly-wetting liquids, including FC-72. They determined that the Cole and Rohsenow [16] departure diameter model best fit available experimental data:

\[
D_b = \sqrt{\frac{\sigma_h E}{g(\rho_f - \rho_g)}}
\]

where

\[
E = \left(0.000465 \cdot Ja^{7/4}\right)^2
\]

and

\[
Ja' = \frac{c_p \rho_f T_{sat}}{\rho_g h_{fg}}
\]

with the saturation temperature specified in absolute degrees. Tong et al. [15] modified the Cole and Rohsenow [16] correlation to include the wall temperature dependence of departure diameter by evaluating the surface tension in Eq. (3) at the heated surface temperature.

Also following [15], the bubble departure frequency is evaluated using the Malenkov [17] correlation:

\[
f \cdot D_b = \frac{U_b}{\pi} \left[ 1 - \frac{1}{\left( 1 + \frac{\rho_f h_{fg}}{\rho_g q} \right)^2} \right]
\]

where

\[
U_b = \sqrt{\frac{gD_b(\rho_f - \rho_g)}{2(\rho_f + \rho_g)}} + \frac{2\sigma}{D_b(\rho_f + \rho_g)}
\]

Further, it is assumed that the bubble growth time, \( t_g \), is one-quarter of the overall bubble departure period \( (1/f) \), with the waiting time, \( t_w \), equal to the remainder [18].

The Benjamin and Balakrishnan [19] nucleation site density correlation is employed, following [20].

\[
\frac{N}{A} = 218.8 \Pr_f^{1.63} \frac{\Delta T_{sat}^3}{\varphi \Theta^{0.4}}
\]

where the surface-liquid interaction parameter is given as

\[
\varphi = \sqrt{\frac{g(\rho_f \cdot k)_{h,0}}{g(\rho_f \cdot k)_{i,0}}}
\]

and the dimensionless roughness parameter is

\[
\Theta = 14.5 - 4.5 \frac{R_i P}{\sigma} + 0.4 \left( \frac{R_i P}{\sigma} \right)^2
\]

In Eq. (10), \( R_i \) is the centerline average surface roughness, assumed equal to 1 \( \mu m \) following [12], and \( P \) is the system pressure (101 kPa).

Bubble Parameter Estimates. Total boiling heat flux is often assumed to consist of additive contributions from various heat transfer mechanisms. Taking this view, the boiling surface in the numerical model is defined with a constant temperature boundary condition. By changing the channel geometry, confinement-driven heat transfer enhancement from various mechanisms may then be evaluated relative to unconfined pool boiling performance.

In order to preserve as much similarity as possible between the 2-D simulations and 3-D experiments, it was deemed most important to maintain the same bubble departure diameter to appropriately capture channel spacing effects while, at the same time, also generating the same volume of vapor in the channel to provide an equivalent buoyancy force. Thus, the point on the pool boiling curve corresponding to the generation of a total vapor volume (number of bubbles times the bubble departure volume) equivalent to the volume of a single 2-D (cylindrical) bubble of the same diameter was chosen as the operating point for the simulations. This point may be expressed mathematically as

\[
\frac{4}{3} \pi \left( \frac{D_b}{2} \right)^3 \frac{N}{A} WL = W \pi \left( \frac{D_b}{2} \right)^2 \Rightarrow D_b \frac{N}{A} L = \frac{3}{2}
\]
For saturated FC-72 at atmospheric pressure and the unconfined boiling curve shown in Fig. 2, Eq. (11) sets the operating point at a wall superheat of 12.3°C and heat flux of 17.7 kW/m² (1.77 W/cm²). The predicted bubble departure diameter is 0.778 mm, with a departure frequency of 59.3 Hz and bubble period and growth time of 16.9 ms and 4.2 ms, respectively. These values are congruent with experimentally-observed measurements reported in the literature for FC-72 and similar highly-wetting organic fluids [21–24].

At this operating point, the predicted nucleation site density is 96354 1/m², or approximately 58 nucleation sites on a 20 × 30 mm (L × H) heater. The volume of a single 0.778 mm diameter bubble is 2.47 × 10⁻¹⁰ m³. The volume of 58 such bubbles is equivalent to the volume of a single, 30 mm deep, 0.778 mm diameter cylinder.

Based on observations in the literature that bubble diameter tends to increase proportionally to time raised to an exponent varying between 1/3 and 1/2 [25], a constant volumetric flow rate over the bubble growth time is assumed. The average vapor mass generation rate over the bubble growth time is

\[
\dot{m} = \pi \left(\frac{D_b}{2}\right)^2 W \frac{\rho_v}{t_g}
\]

(12)

The vapor inlet representing the nucleation site in the CFD model was taken to be 0.1 mm in size. This dimension is not representative of expected nucleation sites but instead represents a compromise between computation resources and bubble behavior—i.e. 0.1 mm is large enough to maintain a reasonable number of computational cells, while it is, at the same time, sufficiently smaller than the bubble departure diameter. Combining this with Eq. (12) yields an expression for the average vapor mass flux over the bubble growth time.

\[
G = \pi \left(\frac{D_b}{2}\right)^2 \frac{\rho_v}{s t_g}
\]

(13)

where s is the size of the vapor inlet, 0.1 mm. For the chosen operating point, Eq. (13) evaluates to 15.18 kg/m³s over the bubble growth time, \(t_g = 4.2\) ms. In order to get the bubble to detach properly from the wall, it was necessary to produce a very short discharge of liquid (less than 0.01 mm thick—much too small to influence channel hydrodynamics) at the nucleation site at the end of bubble growth. It is presumed that if the nucleate site was modeled as a very small inlet, more closely representing actual nucleation site geometries, the availability of surrounding liquid and growth force of the bubble would be sufficient to detach the bubble from the wall and rewet the nucleation site. The total nucleation site mass flow rate averaged over the bubble period is less than 2% of the channel mass flow rate.

**CFD Model**

The model geometry, shown in Fig. 4, corresponding to the experimental boiling channels of Geisler and Bar-Cohen [12] is 20 mm long (in the flow direction) and has a variable channel spacing. The computational domain has been extended 10 mm above and below the channel exit and inlet. These regions have been made wider than the channel itself, to approximate the experimental configuration.

Experimentally, unconfined behavior was observed for channel spacings less than 2 mm by Geisler and Bar-Cohen [12]. For the baseline analysis representing unconfined pool boiling, a channel spacing of 5 mm was chosen for convenience. Most of the computational domain is meshed using a uniform 0.1 mm cell size. A smaller cell size in the vertical direction (0.02 mm) was used to provide five cells across the vapor inlet representing the bubble nucleation site. The vapor inlet is centered along the vertical length of the channel, representing an average location. The mesh of the 5 mm model consisted of 40,200 cells. Channel spacings of 0.7 and 0.3 mm were also analyzed. In the case of the 0.3 mm channel, a slightly different meshing approach was taken to provide six 0.05 mm wide fluid elements across the channel spacing. Cell sizes in the vertical direction were consistent for all simulations.

A detailed study of mesh sensitivity was outside the scope of the current work, however, a test model of a 0.7 mm channel was solved using one fewer cells across the channel gap. While some noticeable differences were observed in the absolute values of the solution results, trends and relative effects were unchanged.
Before running the two phase VOF simulation, a preliminary laminar, steady-state, single phase, natural convection simulation is required to establish the flow field, providing initial conditions to the subsequent transient simulation. During the single phase simulation, the boundary defining the vapor inlet is included as part of the wall, with a constant temperature of 342.1 K, corresponding to the 12.3°C temperature rise of the surface above saturation. A fixed 0.1 ms time step is used for the transient VOF simulations. Given the high vapor velocity during bubble growth, the $k$-$\varepsilon$ turbulence model is employed. Complete, detailed model and solution settings are provided in [5]. It was determined via trial and error that setting the maximum number of iterations per time step at 40 was sufficient. Residuals did not drop below $1 \times 10^{-6}$ for all time steps, but they did tend to level off well within 40 iterations. In addition, monitored quantities, such as heat fluxes and mass flow rates, were observed to level off well within 40 iterations.

Simulations of the 5 mm channel were allowed to run for nearly 80 bubble cycles, until steady conditions were achieved. Narrow channel VOF simulations reached steady conditions significantly sooner than the 5 mm channel and were terminated after 0.60 s of simulated time. Approximately 2 hours of solution time are required for each bubble period (16.9 ms of simulated time), running on a 3 GHz Pentium-based PC running the Windows XP operating system. Thus, run times were on the order of one week.

### NUMERICAL RESULTS AND DISCUSSION

Table 2 provides a summary of simulation results for both single and two phase flow and heat transfer in 5 mm, 0.7 mm, and 0.3 mm channels. These data are discussed in detail below.

#### Single Phase Simulations

Complete single phase results, including temperature and velocity contour plots are presented in [5]. In general, the 5.0 mm and 0.7 mm channels were shown to behave as unconfined vertical plates, while the 0.3 mm channel approaches fully developed channel flow conditions. Single phase heat transfer coefficients calculated from the numerical results are on the order of 100 to 200 W/m²K and agree with analytical predictions within 5% [5].

The data provided in Table 2 include channel mass flow rate, mass flux, and average channel inlet (liquid) velocity. Total mass flow rate is greatest for the 5 mm channel, however mass flux (based on the channel cross-sectional area) and, therefore, fluid velocities are greatest for the 0.7 mm channel. While the liquid flow in the 5 mm channel is clearly unconfined, the thermal boundary layer in the 0.7 mm channel interacts somewhat with the opposing unheated wall. In the 0.7 mm channel, heat transfer from the bottom half of the heater increases 9% compared to the 5 mm channel due to the increased mass flux. At the same time, heat transfer from the top half of the heater is reduced by 18% because of the thermal boundary layer interaction/sensible heat rise of the liquid. The net result is an overall difference of less than 2% in total heat transfer compared to the unconfined (5 mm) case.

The 0.3 mm channel, however, experiences both reduced mass flux and greatly increased sensible heat rise compared to the 0.7 mm channel. This yields significantly deteriorated total heat transfer compared to the wider channels (nearly half). This behavior is evident in the data of Table 2, where it is shown that 89% of the total heat is dissipated from the bottom half of the heater.

### Table 2: Summary and Comparison of Single and Two Phase Numerical Simulation Results

<table>
<thead>
<tr>
<th></th>
<th>Single Phase</th>
<th>Two Phase</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel Spacing, (\delta) (mm)</td>
<td>5 0.7 0.3</td>
<td>5 0.7 0.3</td>
</tr>
<tr>
<td>Channel Mass Flow Rate per Unit Depth (kg/m·s)</td>
<td>0.0169 0.0079 0.0021</td>
<td>0.28 0.10 0.026</td>
</tr>
<tr>
<td>Channel Mass Flux (kg/m²s)</td>
<td>3.4 11 7.0</td>
<td>56 143 87</td>
</tr>
<tr>
<td>Average Channel Inlet Liquid Velocity (m/s)</td>
<td>0.0021 0.0070 0.0043</td>
<td>0.0346 0.0882 0.0535</td>
</tr>
<tr>
<td>Heat Flux (kW/m²)</td>
<td>top 2.09 1.72 0.32</td>
<td>9.96 9.63 6.99</td>
</tr>
<tr>
<td></td>
<td>bottom 3.12 3.40 2.47</td>
<td>4.86 8.45 8.38</td>
</tr>
<tr>
<td></td>
<td>average 2.60 2.56 1.39</td>
<td>7.41 9.04 7.69</td>
</tr>
<tr>
<td>Average Heat Transfer Coefficient (W/m²K)</td>
<td>211 207 113</td>
<td>600 732 622</td>
</tr>
</tbody>
</table>
**Multiphase Simulations**

As previously stated, the single phase steady state solutions were used as initial conditions for the two phase transient simulations. Single phase liquid velocities were less than 0.01 m/s in the vicinity of the nucleation site. Thus early in the transient multiphase simulations, bubbles move less than 0.2 mm in the vertical direction during the time preceding the next bubble generation event. This distance is small compared to the size of the bubbles, and, as a result, the first few bubbles coalesce, until the buoyancy of the vapor is able to accelerate the flow in the channel. Once the leading bubble exits the computational domain (at approximately $t = 0.25$ s), steady (though periodic) conditions are slowly established, and a stream of individual bubbles is observed.

Figure 5 shows liquid phase volume fraction contours from the 0.3 mm channel simulation after 0.60 s of simulated time. Given that the channel spacing is smaller than the simulated bubble departure diameter, bubbles are flattened until exiting the channel. Additional phase contour plots are presented in [5].

During two phase flow, transient fluctuations take place along the heated surface due to bubble growth and motion. In the 5 mm channels, heat fluxes from the top and bottom halves of the heater were 9.96 and 4.86 kW/m$^2$, respectively, at the end of the simulation. These heat fluxes represent a significant increase compared to the corresponding single phase natural convection results of 2.09 and 3.12 kW/m$^2$, respectively. Unlike during single phase natural convection, when the bottom half dissipates more heat because the boundary layer is thinner, the motion of bubbles along the top half of the heater promotes fluid mixing and yields twice the heat dissipation from the top as the bottom. This disruption of the thermal boundary layer due to passing bubbles is shown clearly in the 5 mm channel contour plot of Fig. 6.

![Fig. 5: Liquid Phase Volume Fraction Contour Plot, $\delta = 0.3$ mm, $t = 0.60$ s](image)

![Fig. 6: Two Phase Transient Temperature Contours](image)
While not directly impacted by bubble motions and fluid mixing, heat transfer on the lower half of the heater surface is enhanced due to the increased flow in the channel. As the channel mass flux increases by an order of magnitude, corresponding liquid velocities are also an order of magnitude higher than those observed in the single phase natural convection solution.

In contrast to the single phase simulations which predict decreasing heat transfer with decreasing channel spacing, the two phase simulation results show maximum heat transfer at a spacing of 0.7 mm. Heat flux results for the 0.3 mm channel are very comparable to the 0.7 mm channel along the bottom half of the heater, as shown in Table 2. However, along the top half of the heater there exist large low-flux regions where flattened bubbles cover the heater surface. These vapor-covered regions are larger in the narrower 0.3 mm channel where bubbles are flattened to a greater degree, leading to significantly reduced heat transfer from the top half of the heater. This behavior is illustrated in the heat flux profiles shown in Fig. 7. Low heat flux regions in Figs. 7b and 7c correspond directly to vapor bubble positions at the times shown.

In the 5 mm channel, bubble growth is accompanied by a sharp drop in local heat flux as vapor covers the surface in the vicinity of the nucleation site, as shown in Fig. 7a. After bubble departure, the large channel spacing provides space for the bubbles to migrate slightly away from the heated surface. Thus, the top half of the heated wall remains wetted with liquid, and the bubbles serve mainly to promote mixing and thin the thermal boundary layer. As a result, heat transfer along the top half of the heated wall is maximized.

It is important to stress that the heat flux values shown in Table 2 and Fig. 7 represent only the convection portion of the total boiling heat flux. Following the boiling curves of Fig. 2, total boiling heat flux with a wall superheat of 12.3°C is expected to be 17.6 kW/m², 35.2 kW/m², and 42.2 kW/m² for 5 mm, 0.7 mm, and 0.3 mm channels, respectively. The maximum two phase convective heat flux observed in the numerical results was 9.04 kW/m² for the intermediate 0.7 mm channel—merely 1.63 kW/m² greater than the convective contribution to the unconfined boiling heat flux. This confinement-driven enhancement represents less than a 10% enhancement of unconfined pool boiling. While modest convective enhancement may be sufficient to explain the subtle enhancement observed in the 1.5 mm and 1.6 mm channel data of Fig. 2, the 100% to 500% enhancement observed experimentally at the smaller spacings is far too large to explain in this way.

**ALTERNATE ENHANCEMENT MECHANISMS**

Clearly the numerical results above are based on a number of very significant simplifying assumptions, however, the order-of-magnitude of effects is difficult to dismiss and suggest that increased convection is not the dominant enhancement mechanism. When investigating buoyancy-driven flows in
narrow vertical channels, numerous researchers have considered evaporation of the thin film between compressed bubbles and the heated wall, transient conduction to the liquid phase in the bubble slug wake, or some combination of these two mechanisms, e.g. [6][26][27]. However, detailed knowledge of many two phase flow parameters (void fraction, film thickness, bubble passage times, etc.) is required for these types of predictions. Even when two phase flow parameters are carefully estimated or even measured directly, these models have met with only limited success [28].

In addition to silicon channels already discussed, Geisler and Bar-Cohen [12] performed companion channel boiling experiments with rough aluminum surfaces. Near-CHF behavior was identical to the silicon channels. However, as the aluminum surfaces possessed a large number of potential nucleation sites with a wide variety of shapes and sizes, unconfined pool boiling superheats were significantly less than obtained for polished silicon, and no low flux enhancement was observed, regardless of channel spacing. Geisler [5] proposed that confined bubbles interact with and enhance nucleation sites on the polished silicon surfaces in a manner that modifies bubble nucleation, growth, and departure, leading to increased heat transfer. This enhancement mechanism would not be available to the aluminum surfaces which are rich in nucleation sites and already provide highly effective nucleate boiling heat transfer rates. It is relatively unusual to publish studies showing a lack of expected enhancement. However, as discussed in detail in [5], channel boiling experiments with methanol by Chien and Chen [29] and R-113 by Feroz MD and Kaminaga [30] also show a lack of expected confinement-driven enhancement at sub-unity Bond numbers on rough metal surfaces.

Nucleation site interaction effects have been well documented in a wide variety of forms. Judd and Chopra [31], for example, studied the ability of growing or passing bubbles to deposit bubble embryos and initiate nucleation at adjacent sites. In this highly detailed, data acquisition-intensive experiment, bubble departure diameters and growth and waiting times were measured at randomly located natural nucleation sites on a large heated surface. Chai et al. [20] theoretically considered additional effects of heat conduction in the boiling surface as well as thermodynamic aspects of bubble fluctuations on the stability of adjacent bubble nuclei. Bonjour et al. [21] investigated the effects of bubble coalescence on bubble growth and heat transfer from artificial nucleation sites with various separation distances and demonstrated that moderate coalescence increases heat transfer. Enhancement was attributed to vaporization of the relatively large supplementary microlayer formed between bubble stems.

These insights reinforce the importance of investigating surface characteristics in addition to two phase flow phenomena in confined boiling systems. Generalized physics-based prediction of low flux confinement-driven enhancement should be expected to be extremely complicated and involve the consideration of a wide variety of fluid, surface, material, and geometric parameters.

OPPORTUNITIES FOR FUTURE WORK
The current study represents a significant step toward developing a multiphase CFD capability for the exploration and prediction of fluid confinement phenomena in nucleate pool boiling. Clearly, detailed mesh sensitivity including the impact of the size of the vapor inlet representing the nucleation site should be evaluated. Thin film evaporation may be incorporated into the simulations via appropriate user-defined mass transfer models. Once mass transfer between the phases is included, mesh density at the wall and its effect on the software’s ability to resolve the phase interface and accurately predict the liquid film thickness will be increasingly important.

It is reasonable to expect that multiple discrete bubbles would have a combined area of influence larger than a 2-D “cylinder,” simply due to the larger bubble area to volume ratio. Further, the myriad hydrodynamic complexities associated with multiple bubble streams has, thus far, been avoided for the sake of simulation simplicity. As such, the ultimate goal of this research is clearly to model an entire heater surface and flow channel, in 3-D, with a variety of randomly-placed nucleation sites. Perhaps such a simulation could be based on a carefully-constructed experiment, where individual sites are located, and bubble parameters (departure diameter, growth time, frequency, etc.) are measured, e.g. [31]. Unfortunately, it will be some time before the required computing power is available. In the meantime, development of the required simulation processes and phenomenological models is a worthwhile pursuit and will continue to provide insight into buoyancy-driven two phase flow and boiling heat transfer in confined geometries.

CONCLUSIONS
Evidence of confinement-driven enhancement of nucleate pool boiling is well established in the literature. VOF multiphase simulations of buoyancy-driven saturated boiling in narrow vertical channels have been performed using the commercially-available Fluent CFD software package to investigate the role of convective heat transfer. Results suggest that enhanced natural convection already accounts for a large portion of the unconfined pool boiling heat flux. While increased buoyancy from large vapor fractions in narrow channels may lead to an order-of-magnitude increase in channel mass flux, enhancement of convective heat transfer is found contribute less than an additional 10%. Further, convective enhancement is found to be greater for 0.7 mm channels than the narrower 0.3 mm channels—in direct opposition to experimentally observed trends. These results, in accordance with experimental data from the literature, suggest an alternate enhancement mechanism, one not based solely on increased convection of the liquid phase but primarily dependent on boiling surface characteristics.

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References