SPRAY COOLING HEAT TRANSFER ENHANCEMENT AND DEGRADATION USING FRACTAL-LIKE MICRO-STRUCTURED SURFACES

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ABSTRACT
In the present study, spray cooling curves are presented for two micro-structured surfaces and are compared to smooth surface results. The micro-structured surfaces consisted of bio-inspired fractal-like geometries, denoted as grooves or fins, extending in a radial direction from the center to the periphery of a 37.8 mm circular disc. Depending on the location on the surface, dimensions of groove widths and heights varied from 100 to 500 µm, and 30 to 60 µm, respectively. Fin width and height dimensions remained constant over the surface at 127 and 60 µm, respectively. Results are presented as heat flux versus the surface-to-exit spray temperature difference at each of five volume flux conditions ranging from 0.54 to 2.04 x 10^{-3} m³/m²-s. Convection heat transfer coefficients are also presented for each case as a function of heat flux. Results indicate that at low and high volume fluxes, an improvement in heat transfer occurs in the single phase regime for the fin geometry. Enhancement in the single phase regime does not occur at the intermediate volume flux condition. In the two phase regime for the fin structure significant enhancements, up to 50%, are observed. Whereas the groove structure performs similarly to the smooth surface in the single phase regime and exhibits large degradation in the two phase and critical heat flux regimes, up to 50%. Critical heat flux for the fin surface compares well to that of the flat surface, with a slightly increase at high volume flux conditions.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbols</th>
<th>Greek Symbols</th>
<th>Subscripts</th>
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<tbody>
<tr>
<td>L</td>
<td>Length</td>
<td>μ</td>
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<tr>
<td>m</td>
<td>Mass Flow Rate</td>
<td>Subscripts</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
<td>atm</td>
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<tr>
<td>q''</td>
<td>Heat Flux</td>
<td>Cu, Copper</td>
</tr>
<tr>
<td>Q''</td>
<td>Volume Flux</td>
<td>d, Disc</td>
</tr>
<tr>
<td>R</td>
<td>Thermal Resistance</td>
<td>i, Inner</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds No.</td>
<td></td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>o, Orifice</td>
</tr>
<tr>
<td>V</td>
<td>Velocity</td>
<td>out, Outer</td>
</tr>
<tr>
<td>We</td>
<td>Weber No.</td>
<td>v, Vapor</td>
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<tr>
<td>w</td>
<td>Wall</td>
<td>w</td>
</tr>
<tr>
<td>32</td>
<td>Sauter Mean</td>
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INTRODUCTION
Spray cooling occurs when a liquid stream is broken up into droplets, such as by a pressured swirl or gas atomization nozzle, and then ejected out of an orifice to impinge on a heated surface. Droplets exit the nozzle with a characteristic velocity, size and rate, ultimately forming a three-dimensional spray with specific characteristics unique to the operating conditions. Mean velocities, diameters and droplet number or volume fluxes are typically used in spray cooling to characterize the spray. For a given nozzle, spray characteristics can be a function of fluid properties, pressure drop across the nozzle, and distance from the nozzle exit. Depending on the spray angle and nozzle to surface distance, the droplet impact area can be controlled.

Research is actively being pursued aimed at improving the use of spray cooling in the boiling regime. For example high power electronic require increasing heat dissipation which has led to research in cooling methods other than the conventional...
forced air convection. Spray cooling of electronics is still a developing technique but has been applied to supercomputers and the space shuttles open loop flash evaporator system. Spray cooling is also being used during laser radiation treatment for removal of port wine stains to reduce thermal damage of the epidermis. Future uses may include cooling of high-powered electronic components for lasers, hybrid vehicles and various military applications.

The advantages associated with spray cooling include exceptionally high heat transfer coefficients at high heat flux and relatively low wall temperatures. While similar heat transfer coefficients have been achieved with jet impingement, the advantage of spray cooling is its extremely efficient use of liquid and the spatio-temporal uniformity of temperature at the impingement surface.

The disadvantages of spray cooling include high pressure drop requirements, liquid and vapor fluid collection difficulties, abnormally large number of parameters that effect performance, and inconsistent reported results amongst seemingly identical conditions. The latter two have led to limited progress in predictive capabilities such as correlations or analytical/numerical models, with the vast majority of spray cooling research being experimental in nature.

Spray cooling curves contain three distinct regions of heat transfer; single phase, two phase and critical heat flux (CHF). During single phase heat transfer, the surface is usually flooded with liquid and it is thought that heat is transferred via forced convection with possible free surface evaporation, with the former dominating at high liquid flow rates, and the latter becoming more important at low liquid flow rates. In addition, droplet impingement at the liquid free surface can result in droplet rebound, break up and coalescence with other drops that disrupt the thermal boundary layer and thereby may improve heat transfer.

During two phase heat transfer the heated wall temperature is above the saturation temperature of the fluid and the circumstances are complicated by a thinning of the liquid layer and vapor bubble generation during boiling. As the wall temperature increases the liquid layer thins causing a reduction in the conductive resistance and increase of evaporative heat transfer at the free surface of the liquid layer. Vapor bubbles may or may not be generated at the wall, depending on the spray conditions. With high spray volume flux conditions the impinging droplets can puncture the liquid film, rupture, and sweep away microscopic nucleate bubbles causing a suppression of bubble formation. Conversely, with low volume flux conditions either the droplets do not have enough kinetic energy to penetrate through the liquid layer into the vapor bubbles, or the bubble formation rate and number is significantly greater than the droplet impingement.

At sufficiently high heat flux and/or wall temperature critical heat flux (CHF) tends to ensue at the heated surface periphery and propagates inward. The surface experiences the formation of a vapor layer that is essentially impenetrable to the impinging droplets. At this point a rapid increase in wall temperature and possible decrease in heat flux occurs.

The use of structured surfaces with spray cooling has been reported in a few studies. Silk et al. [1] sprayed PF-5060 at a high mass flux onto cubic pin fin, pyramid and straight fins surface structure geometries with heights of 1mm. Results of the study, when compared to a flat surface, showed a critical heat flux enhancement of 31%, 43%, and 58% for the pyramid, cubic pin fin, and straight fin geometries, respectively. Hsieh and Yao [2] used water at very low mass fluxes sprayed onto multiple square micro-studs of varying height, width and spacing dimensions ranging from 333 to 455, 160 to 480 and 120 to 360 microns, respectively. Relative to a flat surface, heat flux in the two-phase regime was enhanced on the order of 25%, however the single phase regime showed no discernible difference. Coursey et al. [3] sprayed PF-5060 onto five different straight fin geometries with width of 500 microns, fin spacing of 360 microns, and fin heights varying from 0.25 to 5 mm. All surface structures showed a heat flux enhancement in the single phase, two phase regime having a 300% increase in some cases. Sodtke and Stephan [4] sprayed water onto three pyramid geometries with height of 75, 150, and 225 microns. Significant enhancements in heat flux were observed for all three surface structures, with the greatest enhancements occurring for low coolant fluxes.

In the current study the effects of volume flux on spray cooling heat transfer performance are investigated for a flat surface and two different micro-structured surfaces denoted as fins and grooves. The fin and groove surface structures consist of fractal-like branching flow paths; a geometry exhibiting symmetry with the preferential direction of the coolant flow. Fin height and groove depth range between 30 and 60 microns depending on the location; a scale that is smaller than that of the spray Sauter mean diameter which was estimated to be 90 microns. Flat copper and nickel surfaces were also compared to study the effects of surface material.

**EXPERIMENTAL SETUP**

A semi-closed (test chamber open to the atmosphere) flow loop shown in Fig. 1 was used for the spray cooling experiments. De-ionized water was heated to 69 °C in a Hart Scientific (model 7320) hot water bath and drawn through a magnetic drive gear pump (Micro-Pump Series GB) pressurizing the water. Fluid flow rate, temperature, and
pressure at the nozzle were measured with a rotameter, K-type thermocouple, and pressure transducer, respectively. The flow meter was located between the pump and nozzle, and the thermocouple and pressure transducer were located just upstream of the nozzle. A full cone spray pattern was generated using a pressure-swirl atomization nozzle (Spraying Systems TG0.3) with an orifice diameter of 0.51 mm, and a maximum free passage diameter of 0.41 mm. A filter inside the nozzle assembly prevented passage of particles larger than the maximum free passage diameter from entering the swirl chamber. Vapor generated during the cooling of the impingement surface was condensed and combined with the liquid collected in the test chamber. The water flowed back to the water bath using gravity, completing the loop.

The test chamber was fabricated from Plexiglas® enclosing the nozzle assembly, a vapor condenser, a water collection plenum, and a heater assembly. The spray nozzle was mounted to a vertical translation stage capable of varying the nozzle height from 0 to 10 cm with a resolution of 0.2 mm. Vapor generated during the cooling of the impingement surface was condensed and combined with the liquid collected in the test chamber. The water flowed back to the water bath using gravity, completing the loop.

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TABLE 1. NOZZLE, SPRAY AND AMBIENT CONDITIONS DURING TESTING

<table>
<thead>
<tr>
<th>Nozzle Flow Conditions</th>
<th>Spray Conditions</th>
<th>Ambient Conditions</th>
</tr>
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<tbody>
<tr>
<td>( m_0 ) (g/s)</td>
<td>( \Delta P_o ) (kPa)</td>
<td>( \alpha_o ) (deg.)</td>
</tr>
<tr>
<td>( \Delta P_o ) (kPa)</td>
<td>( d_{32} ) (µm)</td>
<td>( V_o ) (m/s)</td>
</tr>
<tr>
<td>( \alpha_o ) (deg.)</td>
<td>( T_{atm} ) (°C)</td>
<td>( P_{atm} ) (kPa)</td>
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<tr>
<td>4.53</td>
<td>434.4</td>
<td>68.4°</td>
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Table 1: Nozzle, Spray and Ambient Conditions during Testing

Table 2: Test Matrix

<table>
<thead>
<tr>
<th>( m_0/m_{tot} )</th>
<th>( Q'' \times 10^3 ) (m³/m²-s)</th>
<th>Nozzle-to-Surface Distance (mm)</th>
<th>( q'' ) (W/cm²)</th>
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<tr>
<td>0.15</td>
<td>0.54</td>
<td>63.3</td>
<td>17 to CHF</td>
</tr>
<tr>
<td>0.25</td>
<td>0.90</td>
<td>53.9</td>
<td>17 to CHF</td>
</tr>
<tr>
<td>0.37</td>
<td>1.36</td>
<td>47.6</td>
<td>17 to CHF</td>
</tr>
<tr>
<td>0.46</td>
<td>1.69</td>
<td>44.4</td>
<td>17 to CHF</td>
</tr>
<tr>
<td>0.56</td>
<td>2.04</td>
<td>41.2</td>
<td>17 to 89*</td>
</tr>
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*CHF was not reached.

DATA REDUCTION

The two vertical columns of thermocouples in the neck of the copper, one located 3.2 mm from the center (designated as inner set) and the other located 15.9 mm from the center (designated as outer set), were used to determine the surface heat flux and the surface temperature. Using an area averaged approach (Eqn. 3), the heat flux was calculated as:

\[ q_w = \left( \frac{k A_i}{L_{1-3}} \Delta T_{1-3} + \frac{k A_{out}}{L_{4-6}} \Delta T_{4-6} \right) / A_d \]  

(3)

where \( k \) is the thermal conductivity of copper, \( A_i \) is the inner surface area of the copper defined by the mean radius between the inner and outer set of thermocouples, \( A_{out} \) is the outer surface area making up the rest of the copper area, and \( A_d \) is the surface area of the disc. The thermocouples are numbered as 1 through 3 for the inner set and 4 through 6 for the outer set. Lengths between thermocouples 1 and 3 and 4 and 6, \( L_{1-3} \) and \( L_{4-6} \), were measured directly using calipers. The temperature differences used here are between thermocouples 1 and 3, \( \Delta T_{1-3} \), and between 4 and 6, \( \Delta T_{4-6} \).

The surface temperature corresponding to the inner thermocouple set, \( T_{w,i} \), was determined using the thermal resistance analogy given by

\[ T_{w,i} = T_3 - q_{w,i} \sum R_j \]  

(4)

where the summation term, \( \sum R_j \), is given:

\[ \sum R_j = \frac{L_{1-3}}{k_d} + \frac{L_{Cu}}{k_{Cu}} + R''_{TIM} \]  

(5)

In Eqns. (4) and (5), \( R''_{TIM} \) is the thermal resistance of the thermal interface material (TIM) and \( q''_{w,i} \) is the heat flux calculated from Fourier’s law using \( \Delta T_{1-3}, L_{1-3}, \) and \( k \). The same procedure was performed for the outer wall temperature, \( T_{w,o} \), with the average of the two used as the reported wall temperature, \( T_w \). Nine repetitions of each measurement were taken with the mean used for calculation of the reported values. Uncertainties for heat flux, wall temperature and the corresponding convection coefficient were calculated using the method of propagation of errors. Thermocouples were calibrated using a NIST standard over a temperature range of 25 to 500 °C, resulting in a calibration uncertainty that ranged between +/− 0.6 and 0.9 °C. The uncertainty in thermocouple location was taken as the radius of the thermocouples (0.26 mm). The thermal conductivity of the nickel disc and the copper between thermocouples and their uncertainties were taken from tabulated data in [7].

Figure 3. Image of radially grooved surface (left) the disk center is on the left side of the image; side view of the end of the grooved surface (right); groove spacing is 0.45 mm.

Figure 4. Image of radially finned surface (left) the disk center is on the left side of the image; side view of the end of the grooved surface (right); fin spacing is 0.45 mm.

and side view images are shown for the groove and fin structured surfaces in Figures 3 and 4, respectively. Additionally, a flat copper disc (> 99.4% pure) was tested at the volume flux of 0.56 m³/m²-s to assess material composition on. Both the flat nickel and the copper surface were prepared identically by sanding with progressively finer grit sandpaper to control the effect of roughness on heat transfer.
RESULTS

To facilitate interpretation of the figures Table 3 shows the symbol type and symbol shading used for each test case. Here, *us*, *fs* and *gs* represent the unstructured (or smooth) surface, finned surface and grooved surface, respectively. For uncertainty values that change little between data points, only a few representative error bars are shown in the figures. Fluid temperature, flow rate, and pressure drop at the nozzle are fixed for all cases at 63.5 ± 0.5 °C, 4.53 ± 0.05 g/s, and 63 ± 3 psi, respectively. The Sauter mean diameter of the droplets is assumed constant and estimated to be 90 μm. A global energy balance was done for the *fs* surface at the highest volume flux and at a copper base temperature of 510 °C (corresponded to a heat flux of 53.2 W/cm² and an area averaged heat flow of 595 W) to help verify heat flux calculations. The time averaged power entering the heaters was measured to be 687 W. Therefore the thermal losses between the ceramic heaters and the copper neck region were on the order of 13%.

Table 3. Symbol type and shading representation

<table>
<thead>
<tr>
<th>( Q' \times 10^3 ) (m³/m²s)</th>
<th>Nickel</th>
<th>Copper</th>
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<tr>
<td>( 0.54 )</td>
<td>[●]</td>
<td>[◇]</td>
</tr>
<tr>
<td>( 0.90 )</td>
<td>[□]</td>
<td></td>
</tr>
<tr>
<td>( 1.36 )</td>
<td>[◇]</td>
<td>[△]</td>
</tr>
<tr>
<td>( 1.69 )</td>
<td>[◇]</td>
<td>[□]</td>
</tr>
<tr>
<td>( 2.04 )</td>
<td>[◇]</td>
<td></td>
</tr>
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</table>

Volume Flux Effects

In this section, heat flux and convection coefficient are presented versus volume flux for each surface. Surface heat flux, \( q'' \), versus the wall-to-exit fluid temperature difference, \( \Delta T_w \), is presented in Fig. 5 for the unstructured surface, *us*, for all five volumetric fluxes. Single phase, two phase and CHF regimes are identified for reference. Similar regimes are observed for all other figures in their range of heat fluxes. In the single phase regime, the heat flux increases with a fairly constant slope, indicating a constant heat transfer coefficient. The heat flux values in this regime ranged from 18 – 24 W/cm² and 17 – 34 W/cm² for the lowest and highest volumetric flux cases of 0.54 and 2.04 x 10⁻³ m³/m²-s, respectively. Uncertainty of heat flux in the single phase regime is between 5 and 10%.

The transition to the two phase transition regime in Fig. 5 begins at a \( \Delta T_w \) between 31 and 36 °C for the high and low volume flux cases, respectively. The slope of the curves are not constant throughout this regime with higher slopes corresponding to higher volume flux conditions. A sharp rise in slope occurs during the transition from the single phase to the two phase regime. This is due to the wall temperature being above the fluid saturation temperature of 100°C, resulting in the beginning of nucleate vapor bubble formation. The value of the wall temperature where transition occurs varies with each volume flux case. The transition occurs at lower \( \Delta T_w \) for higher flow rates but at much higher heat flux values. This indicates that nucleation can occur at lower \( \Delta T_w \) provided the surface heat flux is sufficiently high. A second steeper slope persists in the two-phase regime until the wall temperature approaches the value where CHF occurs. CHF is approached for all cases corresponding to a value of \( \Delta T_w \) near 50°C.

The *fs* surface has qualitatively similar trends with regards to wall temperature and volumetric flux effects on heat transfer and is shown in Fig. 6. Transition to the two phase region occurs at approximately the same value of \( \Delta T_w \). However, CHF values generally occur at a lower \( \Delta T_w \) value, near 45 °C.

The spray cooling curves for the grooved surface, *gs*, is shown in Fig. 7. Note that the scales are different from those in Figs. 5 and 6. The single phase regime for the *gs* surface is similar to that of the other surfaces. Within levels of uncertainty, the single phase slopes are the same with similar quantitative and qualitative variation in heat flux, wall temperature and volumetric flux, with higher volumetric flux shifting the curves to lower values of \( \Delta T_w \). The two phase transition, two phase regime and CHF all show significant differences compared with the other surfaces. The two phase transition and two phase regime have much lower increases in heat flux with increasing \( \Delta T_w \). The CHF value is considerably reduced for the grooved surface compared with unstructured and finned surfaces. The mechanisms causing this trend are not now known, but it is hypothesized that vapor entrapment within the micro-scale grooves, may be prohibiting the cooler liquid

![Figure 5. Heat flux versus wall temperature difference for the unstructured nickel surface.](image-url)
Figure 6. Heat flux versus wall temperature difference for the fin structures surface.

from penetrating into these areas of the surface.

The convection heat transfer coefficient, \( h \), is plotted for the unstructured surface, \( u_s \), in Fig. 8 versus heat flux, \( q'' \), for the range of volumetric flux studied. In the single phase regime \( h \) is relatively high for the lowest heat flux of 17 W/cm\(^2\) and reaches a minimum where the two phase transition region begins, somewhere between 22 and 32 W/cm\(^2\) with increasing values corresponding to larger volumetric flux values. There is a high sensitivity to volume flux in this low heat flux region. There is a much more rapid decrease of \( h \) with increasing heat flux for the higher volumetric flux cases.

Beyond the two phase transition regime, indicated near the minimum \( h \) value, there is a systematic rise of \( h \) towards CHF. In the two phase region, the slope of \( h \) versus heat flux appears to be less dependent on volumetric flux. A possible reason for this is that a higher volume flux can act to suppress growth and sweep away nucleating bubbles, suppressing the relative phase change contribution to heat transfer.

As CHF is approached, the convection coefficient approaches its maximum with a sudden drop signifying the onset of CHF. The convection coefficient values at CHF are highest for the highest volumetric flux conditions and occur at significantly higher values of the surface heat flux. Qualitative trends in convection coefficient for the \( f_s \) surface are similar to the \( u_s \) surface in each of the heat transfer regimes, the results are given in Fig. 9.

The grooved surface, \( g_s \), has markedly different results for the heat transfer coefficient compared to both the \( u_s \) and \( f_s \) surfaces, shown in Fig. 10 as \( h \) versus \( q'' \). In the single-phase regime the behavior of \( h \) is similar to that of the other surfaces where a decrease occurs as heat flux increases, although the decrease is very small at the lowest volumetric flux. Increases in volume flux improve the heat transfer coefficient, with the largest sensitivity occurring in the single-phase regime.

In contrast to the \( u_s \) and \( f_s \) surfaces, the \( g_s \) surface at the three highest volumetric fluxes has a decreasing heat transfer coefficient in the two phase and CHF regimes as the heat flux increases. The lowest volume flux case shows a very slight improvement in the two phase transition regime. The variation of \( h \) versus \( q'' \) with changes in volumetric flux may be due to the heat transfer rate being inhibited by vapor trapped within the micro-grooves. The very low convective coefficient in the single phase regime for low volume fluxes results in the demonstrated increase once transitioning into the two phase regime. However, at the higher volume fluxes the convective

Figure 7. Heat flux versus wall temperature difference for the grooved structured surface.

Figure 8. Convective heat transfer coefficient versus heat flux for the unstructured nickel surface.
heat transfer in the single phase regime is very large and once into the two phase regime, vapor may be trapped locally. Further studies into this phenomenon using optical observations or detailed surface measurements are needed.

Surface Structure Effects

For each of the Figs. 11 through 13, heat flux is plotted versus $\Delta T_w$ for the three surfaces at volume fluxes of 0.54, 1.36 and $1.69 \times 10^{-3}$ m$^3$/m$^2$.s. An unstructured copper disc was also tested to study the effect of surface material and is presented in Fig. 11. Static contact angle at room temperature was measured for the smooth copper and nickel surfaces and found to be approximately 78° for both. In the discussion below reference to

Figure 11. Heat flux versus surface temperature difference for each surface with $Q'' = 0.54 \times 10^{-3}$ m$^3$/m$^2$.s.

enhancements or degradations in heat transfer is with respect to the us surface.

In the single phase regime, both structured surfaces have only a minor effect on the heat transfer, resulting in a slight improvement especially at higher wall temperatures (or heat fluxes). The fs surface appears to transition to the two-phase regime at somewhat lower surface temperatures than the other surfaces. When compared at a given surface temperature the fs enhances the heat flux on the order of 25% near this transition. Although the two phase heat transfer is enhanced, the CHF for the fs surface is somewhat lower than that of the us surface by approximately 10%. In addition, CHF appears at a lower wall temperature difference. These trends for the fs surface in the two phase and CHF regimes are similar to those seen in other studies of structures surfaces [1, 2, and 4].

Towards the end of the single phase regime the gs surface does not exhibit the fairly rapid rise of heat flux with increasing surface temperature. In the two phase regime the gs surface performs significantly worse than the us surface, with heat flux values significantly below that of the other surfaces. This trend of heat flux degradation follows into CHF, with CHF occurring nearly 20-25% lower than the us and fs surfaces.

The copper surface, or cs, shows a slight increase in heat transfer in the single phase and two phase transition regimes compared to the us surface, but is consistently slightly below results for the fs surface. In the two phase regime the cs surface performs almost identically to the us surface indicating little or no impact of the different materials used here. The critical heat flux, however, is significantly higher for the cs surface than for the us surface by approximately 15%. It is thought that this increase is due to better wetting characteristics for the water on the cs impingement surface.

Results for the intermediate volume flux case of $1.36 \times 10^{-3}$ m$^3$/m$^2$.s are shown in Fig. 12. Single phase heat transfer, to
within the uncertainty of the data, are identical for all three surfaces. A slight improvement of the \textit{fs} surface occurs at the onset of the two phase regime, which is at a lower surface temperature. However, enhancements in heat flux are much lower than for the lower volume flux case shown in Fig. 11. Critical heat flux for the \textit{fs} and \textit{us} surfaces are identical. The \textit{gs} surface shows heat transfer degradation everywhere once into the two phase regime with no observed increase in the slope compared to the single phase regime. Also, the CHF value is 44\% lower than the \textit{us} and \textit{fs} surfaces.

The results of the higher volume flux case of \(1.69 \times 10^{-3}\) m\(^3\)/m\(^2\)-s is shown in Fig. 13. The \textit{fs} surface yields higher heat flux values at a given surface temperature compared with the \textit{us} surface over the entire range of surface temperatures. Maximum percent enhancement is not as high as what was found for the two lowest volume flux cases, in this case approximately 20\%. But at this higher volume flux CHF occurs at a slightly greater heat flux for the \textit{fs} surface. As with lower volumetric flux the \textit{gs} surface performance shows no increase in slope into the two phase regime with a very low CHF value.

Some insight into the underlying mechanisms determining heat transfer performance can be illuminated through comparisons of convective heat transfer coefficients, given in Figs. 14 and 15 for the volume flux cases of 0.54 and 1.69 \(10^{-3}\) m\(^3\)/m\(^2\)-s for all surfaces, respectively.

Convection heat transfer coefficients for both the \textit{fs} and \textit{gs} surfaces show some improvement over the \textit{us} surface in the single phase regime, with the \textit{fs} surface performing the best. In the two phase regime, the value of \(h\) for the \textit{fs} surface increases rapidly with increasing heat flux at a nearly constant rate and then experiences a rapid decrease near CHF. Enhancements of the convection heat transfer coefficient for the \textit{fs} surface in the single and two phase regimes are on the order of 10\%. For the \textit{fs} surface CHF occurs at a lower surface temperature than for the \textit{us} surface, consequently, at a sufficiently high heat flux the value of \(h\) drops rapidly, and below that for \textit{us}. The \textit{cs} surface has a greater convection coefficient than all of the other surfaces in the single-phase and CHF regimes, but is lower than that of \textit{fs} in the two phase regime, essentially the same as that for \textit{us}.

Results of the higher volumetric flux condition of \(1.69 \times 10^{-3}\) m\(^3\)/m\(^2\)-s are shown in Fig. 15. Similar to the lower volumetric flux case the \textit{fs} surface enhances \(h\) over the entire regime, but here the CHF occurs at a higher heat flux than for \textit{us} resulting in higher values of \(h\) for \textit{fs} even at the highest heat fluxes. The \textit{gs} surface performs significantly worse in the two phase regimes with no increase of \(h\) into the two phase regime as was detected at the lower volumetric flux.

Overall these results show that the effect of increasing volumetric flux on spray cooling can change significantly in value and in basic trends depending on the surface geometry being sprayed. This is especially the case in the two phase regime and in the CHF. Significant degradation occurs of the expected rise in performance into the two phase regime when using the grooved surface compared to both the unstructured and finned surfaces. CHF occurred for the \textit{gs} surface at significantly lowered heat fluxes for all volume flux cases. In contrast, the finned surface showed enhancement, or no degradation, in all heat flux regimes when compared with the unstructured surface, especially towards the low and high extremes of volume flux studied, except for CHF at the lowest volumetric flux.
CONCLUSION

Spray cooling curves were obtained for a range of five volumetric flux conditions, for two different micro-scale surface structures that are based on fractal geometries, one with fins and the other with grooves. These results are compared with those using an unstructured surface. The finned surface structure geometry enhanced heat transfer relative to the unstructured surface by 15 to 50% for many of the volume flux and heat flux cases considered. Compared to the unstructured surface, CHF for the finned geometry was found to be slightly lower at a low volume flux, and slightly higher at a high volume flux. The grooved geometry demonstrated degradation in heat transfer for the bulk of the spray cooling curve with severe reductions in the two phase regime and CHF. Reduction in heat transfer relative to the unstructured surface reached as high as 50% at CHF for high volume flux cases.

The mechanisms behind the observed discrepancy between performances of the two surface structures tested are currently unknown and to determine these will require detailed local measurements or observations. It is hypothesized that vapor trapped within the grooves inhibits liquid from reaching the groove walls, causing a pre-mature occurrence of dryout leading to lower values of heat transfer where liquid-to-vapor phase change exists.

REFERENCES