Spray Cooling in Terrestrial and Simulated Reduced Gravity

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Abstract. Initial baseline 1-g heat transfer results are reported for an instrumented spray cooling experiment, developed to study effects of electric body forces on spray cooling heat transfer in variable gravity conditions. Heat transfer performance in 1-g for both vertical downward and horizontal spray impingement has been documented for spray volume flow rates of \(4.8 \times 10^{-6} \text{ m}^3/\text{s} \leq Q \leq 9.8 \times 10^{-6} \text{ m}^3/\text{s}\), and heater power levels from 10 W to 70 W using a Thick Film Resistor (TFR) heater. As flow rate is increased at fixed heater power the heat transfer effectiveness increases, as indicated by reduced heater surface temperatures. Heat transfer effectiveness for the vertical downward spray and horizontal spray configurations are nearly identical, but the horizontal spray has somewhat better heat transfer performance when a confining cap is removed at the highest flow rate of \(9.8 \times 10^{-6} \text{ m}^3/\text{s}\). A transparent Indium-Tin-Oxide (ITO) heater consistently has somewhat better performance than the TFR heater. The heat transfer coefficient increases with increased spray flow rate, but is only weakly dependent on the heater power level. Preliminary flow visualization of the spray and liquid film motion for the ITO heater using a high-speed digital video camera and a laser light sheet indicates a highly contorted free surface for the liquid film that forms on the heater surface. Outward radial motion of the wave-like craters and ridges that form on the interface is observed. The liquid droplets that splash off of the heater surface are also observed to move radially, but more slowly than the impinging spray droplets, and are also significantly larger.

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INTRODUCTION

As aircraft and spacecraft become larger and more sophisticated, their thermal management systems must attain higher levels of performance. The Air Force envisions future airborne laser and radar systems that would require heat rejection of between 0.1 MW and 1 MW, and spray cooling is one candidate cooling method under study to meet these increased heat rejection requirements (Mahefkey et al., 2004). A group at the Air Force Research Laboratory (AFRL) under the direction of Yerkes is investigating spray impingement to achieve higher heat transfer performance in a space and weight efficient manner under conditions of variable gravity. Baysinger et al. (2004a) and Yerkes et al. (2006) describe their instrumented spray impingement boiling experiment, which is being flown on the NASA variable gravity research aircraft. Its thermal performance is being documented during both the microgravity (\(\mu g\)) and 1.8-g portions of the flight. The coolant used in the AFRL experiments is FC-72, a nonpolar, dielectric, diamagnetic liquid.

The primary goal of the spray cooling research project at West Virginia University (WVU) is to use electrical body forces to either enhance or balance Earth gravity, thereby simulating the variable gravity effects which result from inertial forces in aircraft and spacecraft without the expense and inconvenience involved in flying experiments on such platforms as sounding rockets or the NASA variable gravity aircraft. A spray cooling experiment has been developed that is identical to the AFRL apparatus, with the addition of provision for generating controlled electric fields near the spray and heater surface. The present work describes initial performance data for this apparatus obtained in terrestrial gravity, where horizontal spray injection has been used to simulate low gravity conditions.

Wolf et al. (1993) have summarized previous boiling jet impingement work. Typical heat transfer rates for single-phase impinging jets are quoted as \(10^5 \text{ W/m}^2\cdot\text{°C}\), with significantly higher heat transfer rates achieved for boiling heat transfer. Spray cooling greatly increases the spatial cooling uniformity relative to liquid jet impingement cooling (Bernardin et al., 1996). Spray cooling has the best performance in terms of large increases in heat transfer coefficient and critical heat flux (CHF) when compared to liquid jet impingement and pool boiling (Chow et al., 1997; Tilton, 1989). With water as the working fluid, spray cooling has achieved a heat flux on the order of 1000 \(\text{W/cm}^2\) in terrestrial gravity (Lin and Ponnappan, 2003). The main applications of spray cooling have been in the quenching of aluminum and steel, in nuclear reactor safety devices, and the steady heat removal from lasers or electronic equipment.

For dense sprays, a liquid film forms on the heater surface, and at sufficiently high heat rejection rates vapor bubbles are formed in the film at nucleation sites; see the review by Chow et al., 1997. These vapor bubbles tend to be swept
away from the heater surface by the radial motion in the liquid film, resulting in efficient re-wetting of the heated surface. This mechanism for rewetting of the heater surface is absent in pool boiling. As more of the impinging liquid boils, eventually the vapor leaving the heated surface prevents smaller liquid droplets from reaching the heater surface; this effect is thought to be a critical factor in the onset of CHF.

Tilton (1989) speculated that one reason spray cooling may be ideal for spacecraft is a lack of dependence upon buoyancy forces for vapor removal and liquid supply. Yoshida et al. (2001) found that spray cooling in the nucleate boiling regime was more effective in simulated microgravity than in terrestrial gravity when water was used, but when FC-72 was used only a slight improvement was seen. They simulated microgravity conditions by rotating the apparatus 180 degrees, changing the direction of the heated surface from upward facing to downward facing. The degree to which changing surface orientation accurately simulates µg is not clear. Also, it is noted that their Weber numbers were significantly smaller than in the present work. Kato et al. (1994) also used orientation to simulate microgravity conditions, although they used a horizontal spray to simulate conditions of µg, and found data that was comparable to that of Yoshida et al. (2001). Baysinger et al. (2004a) found flow management to be a critical issue in successful µg operation of their spray cooling experiment.

The goal of the work reported herein is to obtain initial performance data for the WVU spray cooling apparatus under terrestrial gravity, using horizontal spray (Kato et al., 1994) to simulate microgravity effects. Also, preliminary flow visualization of droplet impingement and splashing and the dynamics of the liquid film on the heater surface are reported. Future work is ongoing to add the electric Kelvin force effects, and microgravity tests are also planned.

**EXPERIMENTAL APPARATUS**

The present apparatus has been designed to match the AFRL spray cooling experiment (Baysinger et al., 2004a) as closely as possible, using identical spray nozzles, heaters, and sump geometry, as well as using the same working fluid of FC-72. A detailed description of the current apparatus has been given in the thesis by Hunnell (2005). A schematic of the spray nozzle, heater surface, pedestal, and sump is shown in Fig. 1. The nozzle and heater surface have been housed in a spray chamber that has been fitted with view ports for flow visualization. The geometry of this portion of the present apparatus is identical to the AFRL apparatus, except for electrical penetrations in the WVU spray chamber for an ongoing study of the effects of electrical forces on spray cooling performance.

At an FC-72 flow rate of 9.5x10^-6 m³/s the Spraying Systems full cone 1/8G-1 nozzle generates a dense spray of droplets with a nominal droplet diameter of 48 µm at a nominal velocity of 12 m/s (Baysinger et al., 2004a). Nozzle-heater spacing for the present results is 13 mm, and heater diameter is 16 mm. This spacing was chosen so that the outer edge of the cone coincides with the circumference of the heater. Two different types of heaters have been used in the present work: an optically transparent Indium-Tin-Oxide (ITO) heater, and a more robust ceramic Thick-Film Resistor (TFR); these same heater types are being used in the AFRL work. The heaters are bonded to the top of an optically transparent cylindrical glass pedestal (Fig. 1); the pedestal is installed in a sump that is used to collect the excess liquid FC-72 for recirculation. Usually the sump has been fitted with a hollow conical “cap” that is used to redirect the excess liquid into the sump, to aid in liquid flow management; results have also been obtained without a cap fitted to the sump (called “unconfined flow” herein). The transparent pedestal allows viewing the flow on the surface of the ITO heater from below.

The spray chamber housing the nozzle and sump has been fabricated from a stainless steel flanged 15 cm ID short pipe nipple fitted with two smaller-diameter flanged side ports: one for the spray nozzle supply piping, and a second for the heater/pedestal/sump assembly. The spray chamber is sealed by 2.5 cm thick polycarbonate transparent blanks bolted to the flanges; these ports are used for visualization of the spray. This spray chamber has been pressure tested to 1.4 MPa. It is this portion of the apparatus that is essentially identical to the AFRL design. For the present work, the spray chamber has been mounted to an experiment base that houses the needed pumps, valves, heat exchangers, and instrumentation. The spray chamber and all its components will be used in a planned series of future µg flights, while the present experiment base will only be used at WVU for 1-g baseline studies.

The FC-72 is pumped from a reservoir by a positive-displacement gear pump to the spray nozzle, where flow rate is controlled by setting the pump speed, and a rotameter monitors flow rate. The spray impinges on the heater surface, and the excess liquid is collected in the sump, while the vapor is condensed on the chamber walls. A positive-displacement diaphragm pump sends the sump fluid through a liquid-air heat exchanger back into the reservoir. Spray chamber temperature is controlled by water flowing through tubes cemented to its outside cylindrical surface.
This flow is created by a separate flow loop, consisting of a water reservoir, centrifugal pump, rotameter, and liquid-air heat exchanger. Both flow loops contain filters, as well as pressure and temperature instrumentation.

The glass pedestal onto which the heater has been mounted has seven type E thermocouples installed in the same locations as those used for the AFRL apparatus (Baysinger, 2004b). The analysis developed by Yerkes, et al. (2006) has been used to compute the heater surface temperature from the measured temperature 0.5 mm below the heater surface in contact with the FC-72. All thermocouples have been calibrated against precision mercury-in-glass thermometers with 0.1 °C resolution in a stirred water bath from 20 to 60 °C. Flow meter repeatability is estimated as ±3 to 5%, heater power accuracy is estimated as ±0.5 to 1W, and temperature resolution is ± 0.1 °C, with an estimated accuracy of ±0.2 °C (Hunnell, 2005). Error bars shown for the dimensional results have been computed using these uncertainty estimates.

These dimensional results have also been analyzed in the non-dimensional form developed by Yerkes et al. (2006). The heat flux has been non-dimensionalized as the parameter, \( \Delta \) (Yerkes et al, 2006), defined as:

\[
\Delta = \frac{\text{Heater Power}}{\pi b T_{\infty,\text{wall}} k_{\text{fur}}},
\]

where \( b \) is the radius of the heater, \( T_{\infty,\text{wall}} \) is the average temperature of the liquid FC-72 as it travels through the sump, and \( k_{\text{fur}} \) is the thermal conductivity of the heater material. The temperature difference, \( (T_s - T_{\infty,\text{top}}) \), has been non-dimensionalized by dividing by \( T_{\infty,\text{wall}} \). The reference temperature \( T_{\infty,\text{top}} \) is the average temperature of the FC-72 liquid film on the heater surface, taken as the average of the initial liquid spray temperature and the temperature of the spray as it leaves the heater surface. \( T_s \) is the heater surface temperature computed from the measured interface temperature at the bottom of the heater substrate. For the present results, the heat transfer model developed by Yerkes et al. (2006) has been extended to include the effect of the insulating glass layer on the top side of the TFR ceramic heater in the calculation of the heater surface temperature, \( T_s \). This results in a reduction of the heater surface temperature of approximately 5 °C at a heater power of 60 W.

Computed heat transfer coefficients have been non-dimensionalized as the Nusselt number, \( \text{Nu} \), defined as:

\[
\text{Nu} = \frac{h b}{k_{\text{fluid}}},
\]

where \( k_{\text{fluid}} \) is the liquid thermal conductivity and \( h \) is the heat transfer coefficient.

\[ \text{FIGURE 1. Schematic of the Nozzle, Pedestal, Heater, Spray, Sump, and Cap.} \]

**RESULTS**

Spray cooling performance results for the present apparatus have been presented by Hunnell (2005) under conditions of terrestrial gravity for both vertically-downward directed and horizontal sprays, for FC-72 volumetric flow rates of \( 4.8 \times 10^{-6} \text{ m}^3/\text{s} \leq Q \leq 9.8 \times 10^{-6} \text{ m}^3/\text{s} \), and heater power levels between 10 W and 70 W primarily using the TFR heater. These results were obtained at a nozzle-to-heater spacing of 13 mm, for three different sump geometries: an
unconfined flow (no cap installed on sump), and two confined flows (straight-walled cap, identical to the AFRL geometry, and curved-wall cap). Heat flux has been computed as the ratio of the heater power divided by the heater surface area. No correction has been made for heat lost down the pedestal; Baysinger (2004b) showed that this effect was only from 1 to 3% of the heater power. The heater surface temperature, $T_s$, has been computed from the measured temperature 0.5 mm beneath the heater surface, as described above. Since no attempts have been made to control the amount of dissolved air in the liquid FC-72, it has been assumed that the FC-72 used herein was saturated with air.

Heat flux results are plotted versus the difference between the heater surface temperature and the average temperature of the liquid film on the heater surface, $(T_s - T_{\infty, \text{top}})$, for unconfined flow at $6.5 \times 10^{-6}$ m$^3$/s and $9.8 \times 10^{-6}$ m$^3$/s in Fig. 2a; results for the straight-walled and curved-wall caps look similar. Uncertainty estimates for the heat flux and temperature difference are shown as error bars in the figure. For the unconfined geometry, the horizontal spray is slightly more efficient than the vertical spray at a flow rate of $9.8 \times 10^{-6}$ m$^3$/s. This is consistent with trends observed by Kato et al. (1994) and Yoshida et al. (2001). However, at $6.5 \times 10^{-6}$ m$^3$/s, a slight opposite trend is observed for the unconfined flow.

Heat transfer performance of the two different heater types (TFR and ITO) is shown in Fig. 2b for a spray flow rate of $9.8 \times 10^{-6}$ m$^3$/s. The ITO heater is consistently more efficient (cooler at equal heat fluxes); this is believed to be due
to the differences in heater geometry. The ITO heater is formed by deposition of a thin ITO resistive film on the glass surface, and is thus in direct contact with the impinging FC-72 spray. The TFR heater has its resistive film sandwiched between a bottom substrate and a thin glass top layer in contact with the FC-72.

The spray droplet Reynolds, Weber, and capillary numbers have been computed (Hunnell, 2005), based on FC-72 properties, and the droplet diameter and velocity measurements obtained by Baysinger et al. (2004a) for an identical spray nozzle. Weber and Reynolds numbers were 1140 and 1650 respectively. The relatively large values of the Weber and Reynolds numbers indicate that droplet momentum dominates over both surface tension and viscous forces. The capillary number \( \left( \frac{\mu V}{\sigma} \right) \) was 0.7. A capillary number of 1 indicates that surface tension forces are comparable to viscous forces.

The current non-dimensional results appear in Figs. 3, 4, and 5. Not much difference is observed between the vertical and horizontal sprays at the same FC-72 flow rate and heater power for the unconfined flow (Fig. 3a) or for the straight cap geometry (Fig. 3b). The unconfined spray shows some improvement for the horizontal spray at the

![Graph](image-url)

(a) Unconfined flow.

![Graph](image-url)

(b) Straight cap geometry.

**FIGURE 3.** Non-dimensional Heat Flux vs Nondimensional Surface Temperature Difference.
The highest flow rate of 9.8x10^{-6} m³/s (Fig. 3a). The greater heat transfer efficiency of the ITO heater compared to the TFR heater for the vertical spray remains in the non-dimensional presentation (Fig. 4). Nusselt number (Fig. 5) increases versus FC-72 flow rate, but there is not much difference between results for the vertical and horizontal sprays. Nusselt number increases only slightly versus nondimensional heater power, GΔ. These Nusselt number results are for the TFR heater, and are about 30 to 40% lower than results for an ITO heater presented by Baysinger (2004b); this is consistent with the present comparison in Fig. 2b.

**FIGURE 4.** Vertical Spray Non-dimensional Heat Flux vs. Non-dimensional Surface Temperature Difference for ITO and TFR Heaters.

**FIGURE 5.** Nusselt Number vs. Non-dimensional Heat Flux for Vertical Downward Spray and Horizontal Spray with Straight Cap.

Preliminary flow visualization of spray impingement and droplet ejection or splashing and the time-dependent behavior of the liquid film on the heater surface has been obtained using a Redlake model HG-LE high-speed digital video camera and laser light sheet illumination (Hunnell, 2005). An example video image of the droplet impingement and ejection as viewed from the side of the impingement region is shown in Fig. 6, while an image of the liquid film behavior, viewed from the underside of the transparent ITO heater, is shown in Fig. 7. Ejected liquid drops appear to be much larger than the 48 µm diameter impinging spray droplets, and move much more slowly (Fig. 6). Liquid film motion (Fig. 7) is also much slower than spray droplet velocity (~1 m/s, versus ~10 m/s). The liquid film surface appears to be highly contorted (Fig. 7), with a series of deep craters and ridges apparently formed by the spray impingement and the ejection of the larger droplets.
Motion of these larger droplets, as well as the motion of the craters and ridges observed on the surface of the liquid film, is always in the radial direction.

**FIGURE 6.** Interaction of Spray and ITO Heater at $Q = 5.7 \times 10^{-6}$ m$^3$/s and 17 W/cm$^2$ (1000 fps).

**FIGURE 7.** Liquid Film on Heated Surface Viewed through Bottom of Pedestal at $Q = 5.7 \times 10^{-6}$ m$^3$/s and 17 W/cm$^2$ (3000 fps).

**CONCLUSIONS**

Initial baseline 1-g heat transfer results are reported for an instrumented spray cooling experiment that uses a full-cone spray nozzle to cool a circular resistive heater, using both ceramic Thick Film Resistor (TFR) and optically-transparent Indium-Tin-Oxide (ITO) heaters. A conical cap surrounding the heater area that directs the excess liquid spray into an annular sump to be recirculated facilitates flow management. Heat transfer performance in 1-g for both vertical downward and horizontal spray impingement has been documented for spray volume flow rates of $4.8 \times 10^{-6}$ m$^3$/s $\leq Q \leq 9.8 \times 10^{-6}$ m$^3$/s, and heater power levels between 10 W and 70 W using the TFR heater. As flow rate is increased at fixed heater power the heat transfer effectiveness increases, as indicated by reduced heater surface temperatures and increased heat transfer coefficients. Heat transfer effectiveness for the vertical downward spray and horizontal spray configurations are nearly identical. This may indicate that horizontal spray injection cannot adequately simulate behavior in μg. The horizontal spray has slightly better heat transfer performance for the unconfined flow at the highest flow rate of $9.8 \times 10^{-6}$ m$^3$/s. The ITO heater consistently has somewhat better performance than the TFR heater. The heat transfer coefficient increases with increased spray flow rate, but is only weakly dependent on the heater power level.

Preliminary flow visualization of the spray and liquid film motion using a high-speed digital video camera and laser light sheet illumination indicates a highly contorted free surface for the liquid film that forms on the heater surface. Outward radial motion of the wave-like craters and ridges that form on the interface is observed. The liquid drops that splash off of the heater surface are also observed to move radially, but more slowly than the impinging spray droplets; these drops are also significantly larger.
NOMENCLATURE

\( b \) = radius of the heater (m)
\( G \Delta \) = (Heater Power) / (\( \pi b T_{o,wall} k_{htr} \)), the non-dimensional heater power
\( h \) = heat transfer coefficient (W/m\(^2\)-°C)
\( k_{fluid} \) = thermal conductivity of the liquid (W/m-°C)
\( k_{htr} \) = thermal conductivity of the heater (W/m-°C)
\( \text{Nu} \) = Nusselt number
\( Q \) = liquid volume flow rate (m\(^3\)/s)
\( T_s \) = heater surface temperature (°C)
\( T_{sat} \) = liquid saturation temperature (°C)
\( \Delta T_{sub-cooling} \) = difference between heater surface and liquid temperatures (°C)
\( T_{\infty, top} \) = average liquid temperature of on heater surface (°C)
\( T_{\infty, wall} \) = average liquid temperature of the liquid in sump (°C)
\( V \) = droplet velocity (m/s)
\( \mu \) = liquid viscosity (N-s/m\(^2\))
\( \Theta_s \) = (\( T_s - T_{\infty, wall} \))/\( T_{\infty, wall} \) = non-dimensional heater surface temperature
\( \Theta_{\infty, top} \) = (\( T_{\infty, top} - T_{\infty, wall} \))/\( T_{\infty, wall} \) = non-dimensional liquid film temperature on surface of heater
\( \sigma \) = liquid surface tension (N/m)

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