Foam Spray Cooling Heat Transfer Performance Study

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ABSTRACT

Previous studies have shown that spray cooling heat flux enhancement may be attained using enhanced surfaces. However, most enhanced surface spray cooling studies have been limited to extended and/or embedded surface structures. This study investigates the effect of foam on spray cooling heat flux. The foam used was graphite Poco Foam. The foam piece was attached to a copper block with a cross-sectional area of 2.0 cm^2 using high thermal conductivity epoxy as the thermal interface material (TIM). Measurements were also obtained on a heater block with a flat surface for purposes of baseline comparison. A 2x2 nozzle array was used with PF-5060 as the working fluid. Thermal performance data was obtained under nominally degassed conditions (chamber pressure of 41.4 kPa). Results show that the highest heat flux (CHF) attained was 113 W/cm^2 using the graphite Poco Foam.

INTRODUCTION

Spray cooling heat transfer is a multiphase convective heat transfer process that has gained much attention in recent years as a viable thermal management solution to high heat flux problems within the electronics, aerospace and naval industries. Review of spray cooling literature shows that a great deal of research has been conducted to gain a better understanding of the phenomena associated with the spray cooling heat transfer process. Previous studies performed have parametrically examined the effect of secondary gas atomizers vs. pressure atomizers (Sehmbey et al., 1995; Yang et al., 1993), mass flux of ejected fluid (Estes and Mudawar, 1995; Yang et al.,, 1996), spray velocity (Chen et al., 2002; Sehmbey et al., 1992), droplet impact velocity (Chen et al., 2002; Healy et al., 1998; Sawyer et al., 1997), surface roughness (Sehmbey et al., 1995; Sehmbey et al., 1992; Bernadin and Mudawar, 1999; Pais et al., 1992), ejected fluid temperature, chamber environmental conditions, and spray footprint optimization on the effective heat flux across the heater surface (Mudawar and Estes, 1996). Other topics studied include the effect of surfactant addition (Qiao and Chandra, 1997; Qiao and Chandra, 1998), and secondary nucleation (Sehmbey et al., 1995; Mesler, 1993; Rini et al., 2002).

Most studies that have examined enhanced surfaces have done so primarily from the perspective of surface roughness (Sehmbey et al., 1995; Pais et al., 1992) and micro-structured surfaces (Hsieh and Yao, 2006). However, as interest in spray cooling grows, enhanced surface spray cooling investigations emphasizing structure enhancements beyond the surface roughness level are also increasing in number. In the study by Silk et al. (2006), the effects of enhanced surface structures beyond the surface roughness range on spray cooling heat transfer were investigated. The surface enhancements consisted of cubic pin fins, pyramids, and straight fins machined on the top surface of heated copper blocks with 2.0 cm^2 cross-sectional areas. Measurements were also obtained on a heated flat surface for data comparison. PF-5060 under nominally degassed conditions (chamber pressure of 41.4 kPa) was used as the working fluid. Spray volumetric flux (0.016 m^3/m^2s) and nozzle to heater distance (17 mm) were held constant throughout each test. The spray temperature was 20.5°C. The study showed that the straight fins had the largest heat flux enhancement relative to the flat surface, followed by the cubic pin fins and the pyramid surface. Each surface had an increase in evaporation efficiency at CHF compared to the flat surface. The authors determined that the straight finned surface had the most efficient use of area added for additional heat transfer relative to the flat surface. They also determined that heat flux enhancement observed with the use of enhanced surfaces is a function of surface area added and liquid management on the heater surface.

Coursey et al. (2005) investigated spray cooling of high aspect ratio open micro-channels with a full cone nozzle. The study included five heat sinks each having a projected surface area of 1.41x1.41 cm^2, a channel width of 360 µm and a fin width of 500 µm. Fin heights tested ranged between 0.25 mm and 5.0 mm. A flat surface with the same projected surface area as the enhanced surfaces was also tested for data comparison purposes. PF-5060 at T_{sat}=30°C was used as the working fluid. Studies were performed with nozzle pressures ranging 137.8 kPa ↔ 413.4 kPa. The study showed that heat flux performance for each of the enhanced surfaces improved relative to the flat surface's performance. The authors determined that longer fins outperform shorter ones in the single phase regime and that the addition of fins resulted in multi-phase effects at lower wall temperatures compared to the flat surface. The studies also showed that the use of enhanced surfaces increased the phase change efficiency in comparison to the flat surface case. Fin heights between 1.0 mm and 3.0 mm provided optimum heat flux performance for the test conditions studied.

Much work has been performed using porous foam and microporous surfaces as a heat flux enhancement technique for forced convection (Shih et al., 2006; Jamin and Mahmoud, 2007; Salas and Waas, 2007), thermosyphons (Coursey et al., 2005), pool boiling (Kim et al., 2002; Parker and El-Genk, 2006; Li and Peterson, 2007) and thermal energy storage (Pauken et al., 2007) applications.

Jamin and Mahmoud (2007) performed a forced convection study to quantify heat transfer enhancement for Poco HTC foam (manufactured by Poco Graphics, Inc.) in cross flow. Measurements were taken for the heat transfer rate and pressure drop for a heated vertical pipe with and without the porous foam. The copper pipe had an O.D. of 15.875 mm and a length of 152.2 mm. These dimensions were constant for all the cases studied. Tests were performed over a range of velocities (≈ 8 ↔ 45 m/s) using air as the working fluid. The studies showed that the HTC foam enhanced heat transfer by factors between 1.3 to 3.05 relative to the bare copper surface's heat transfer performance. However, use of the foam resulted in increased pressure drop across the test section. The authors determined that this was due to the low permeability of the foam which created a large resistance to fluid flow inside the porous media.

Coursey et al. (2005) investigated thermal performance of a graphite foam thermosyphon evaporator. The graphite foam used in their study was Poco Foam (manufactured by Poco Graphics, Inc.). Multiple foam samples having variations in height, width, and density, were bonded to copper heaters each having a cross-sectional area of 1.0 cm^2. Evaporator performance was examined using FC-72 and FC-87 as working fluids. Effects of liquid fill level and condenser temperature as well as foam height, width and density were studied. The study showed that the heat transfer performance of the two working fluids was similar. However, the liquid fill level, condenser temperature, geometry and density of the graphite foam had a significant affect upon thermal performance. The authors determined that the boiling process was surface tension dominated. The highest heat flux attained in this study was 149 W.

In the study by Parker and El-Genk (2006), orientation effects upon nucleate boiling were investigated for a porous graphite and a smooth copper surface. Inclination angles for the orientations investigated ranged from 0° inclination (heater normal facing upwards) to 180° inclination (heater normal facing downward). The projected area for heat exchange was 10 mm x 10 mm for both heaters. The porous graphite and flat copper surface were attached to the heating elements on their test section using high thermal conductivity epoxy. Degassed FC-72 was used as the working fluid. Nucleate boiling heat flux, the nucleate boiling heat transfer coefficient and CHF were reported and compared. The study showed that the highest CHFs were attained for both the porous and the flat surfaces in the upward facing orientation (i.e., 0° orientation) and had values of 30 W/cm^2 and 18 W/cm^2 respectively. The study also showed that the porous graphite surface had consistently higher CHF values than the flat copper surface for all orientations tested.

Prior enhanced surface spray cooling studies (Silk et al., 2006; Coursey et al., 2007; Silk, 2008) have shown that spray cooling of enhanced surfaces (extended fins and porous tunnels) results in a corresponding heat flux enhancement. Since spray cooling is a multiphase convective process, and the use of porous foam has shown heat flux enhancement for nucleate boiling and forced convection processes, spray cooling of porous foam may also be a viable technique for providing spray cooling heat flux enhancement. The present work investigates spray cooling heat flux performance using porous foam as a surface enhancement. Heat flux as a function of volumetric flux is reported for a single Poco Foam sample. Heat flux data attained using the Poco Foam as a surface enhancement is also compared to that of a flat copper surface having the same projected heat exchange surface area. It was found that the Poco Foam surface provided a heat flux enhancement relative to the flat surface data. However, superheat levels at the copper/foam interface were noticeably greater than those observed at the liquid/solid interface for the bare copper surface.

TEST SETUP AND PROCEDURE

The experiments were conducted using a closed fluid loop system. The test rig (schematic shown in Fig. 1) consisted of an environmental test chamber, liquid pump, flow meter, micro-filter and a condenser. Chamber temperature and pressure were measured via a T-type thermocouple and a pressure sensor. Temperature and pressure sensors were also placed in the liquid line upstream of the nozzle for fluid and supply line temperature and pressure measurement.

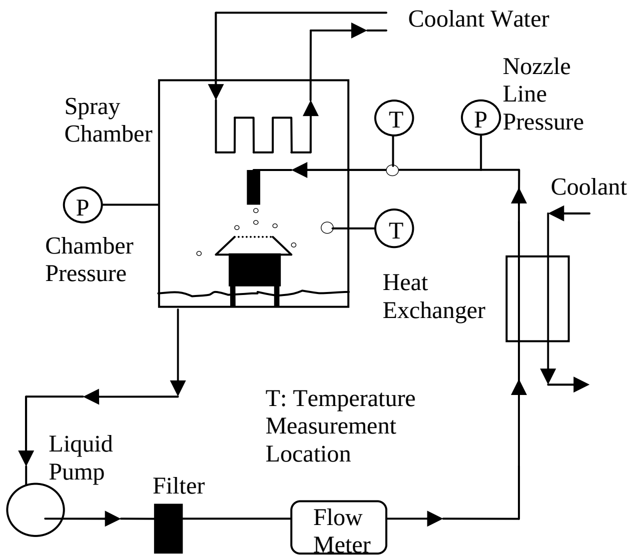

Fig. 1. Spray Cooling Test Rig Configuration

Heat was supplied to the test article using a 500 W cartridge heater. The test article was placed within the interior of the chamber, but was separated from the excess liquid by an enclosure consisting of a polycarbonate housing and an alumina bisque ceramic top flange (Fig. 2). The upper section of the copper block was epoxied to the ceramic flange. Temperature measurements in the copper blocks were taken via five T-type thermocouples mounted in the upper section of each block (shown in Fig. 3). Assuming steady state 1-D conduction through the upper portion of the block, the

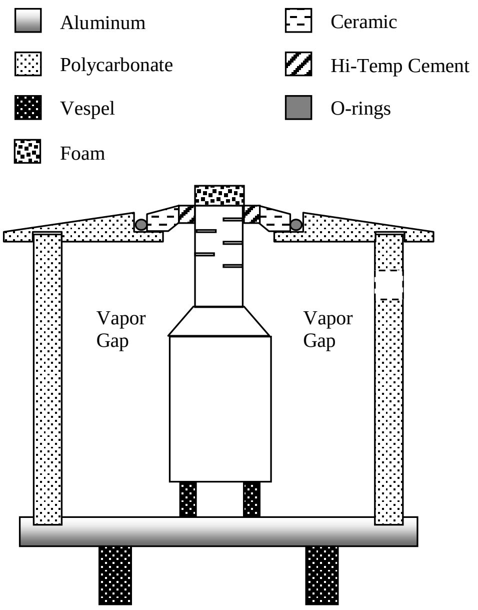

Fig. 2. Copper Block Housting Schematic

heat flux was calculated using Fourier's Law. Reported heat flux was determined as the average value from multiple pairs of thermocouples (TC1 through TC5). Surface temperature was determined via linear extrapolation using TC1 and TC2.

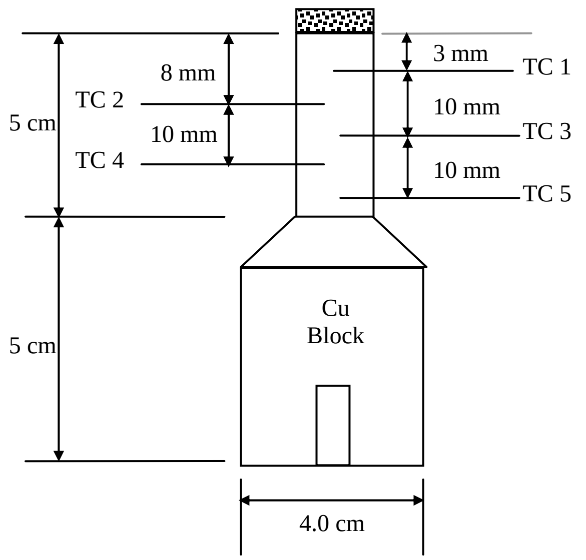

Fig. 3. Copper Block Housing Schematic with TC Locations (not to scale)

Before each test, the spray chamber and fluid loop were charged with PF-5060 and a vacuum was repeatedly applied to the chamber until a pressure of 41.4 kPa (470 ppm gas concentration) was reached. The chamber was allowed to attain equilibrium prior to conducting the tests. Test conditions are shown in Table 1.

Spray Nozzle Array. A Parker Hannifin spray nozzle consisting of a 2x2 spray cone array was used for each of the tests (a close-up of the individual spray cones is shown in Fig. 4.). Given the nozzle-to-heater surface

Table 1. Test Case Conditions

Spray Cooling Parameters	
Parameters	**Values**
P_{sat} (kPa)	41.4
T_{sat} (^{o}C)	31
T_l (^{o}C)	20.5
h_{fg} (kJ/kg)	92
Gas Content (ppm)	470

Fig. 4. Spray Manifold Close-up of 2x2 Nozzle Array

distance used, the heater surface area and the spray configuration for the present tests, the volume flux at the center of the heater surface was twice as much as the average for the entire heater surface. Towards the perimeter of the heater surface the volume flux reduced to as much as 40% of the average area value. Thus, the spray can be considered a non-uniform center biased spray. For a detailed explanation of the volume flux experiment performed, please see previous work by Silk et al. (2006).

Poco Foam. The porous graphite foam used in the present study was Poco Foam which is manufactured by Poco Graphite, Inc. The Poco Foam was machined into the shape of a cylinder of height 5.9 mm and an approximate diameter of 16 mm (size scale photo shown in Fig. 5.). This gave the top and bottom sides of the foam cylinder a cross-sectional area of 2.0 cm^2. Table 2 provides a listing of the properties of the Poco Foam.

Prior to attachment of the foam to the copper block, the top surface of the copper block was slightly roughened (using scotch-brite) for better adherence of the bonding material. The porous foam test piece was then bonded to the top surface of the copper block using a thin layer (≤ 0.1 mm) of high thermal conductivity epoxy

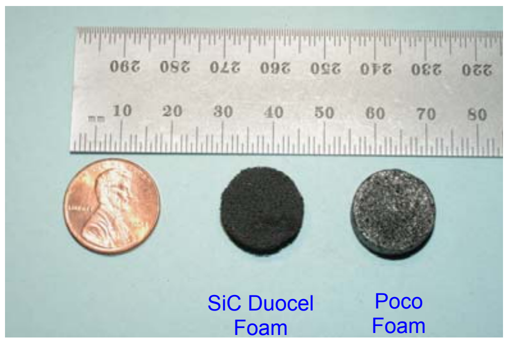

Fig. 5. Photos of Foam Structures

Table 2. Summary of Foam Properties

Property	Poco Foam
Average Pore Diameter	≈ 350 μm
Open Porosity	96%
Total Porosity	75%
Density (ρ)	0.55 g/cm^3
Specific Heat (c_p)	0.7 J/g-K
Planar Conductivity (k_r=k_θ)	45 W/m-K
Out of Plane Conductivity (k_z)	135 W/m-K

(Stycast 2850 FT/Catalyst 11 by Emerson and Cummings, Inc.). The thermal conductivity of the epoxy was approximately 1.3 W/m-K. The copper/foam TIM was then cured at 120°C for one hour prior to testing.

Liquid Passage Through Poco Foam. During the heat flux experiments, the porous foam experienced liquid flow across its top surface. Assuming that a portion of the liquid sprayed onto the top surface of the Poco foam traveled through the interior of the foam, knowledge regarding the amount of liquid internal to the foam at a given instant is of value in the assessment of heat transfer and fluid dynamics phenomena internal to the porous structure during heating. One way to quantify how much liquid was internal to the foam at a given instant was to measure the liquid volumetric flow through the foam (top to bottom) while being sprayed at a constant volumetric flux. With the aid of a stopwatch, graduated cylinder, and a specially fabricated holder for a foam sample identical to the one bonded to the copper heater, the volume flow rate through the foam was measured at each of the flow rates used for heat flux performance testing. The aluminum holder (shown in Fig. 6) constrained the liquid flow internal to the foam to the axial direction while allowing free passage of liquid through the bottom. Liquid flow measurements for the foam were performed with the apparatus shown in Fig. 6. placed internal to the spray chamber at the same partially degassed saturation conditions and spray manifold height used for the heat flux performance tests.

MEASUREMENT UNCERTAINTY

The primary quantity of interest for these experiments is the heat flux. The heat flux calculation has three contributions to the uncertainty: the conductivity, the thermocouple locations, and the error in the temperature measured. The conductivity value used was 389 W/m-K with an estimated error of 1%. The error in the thermocouple temperature measurements was estimated as ±0.5°C. The error in the thermocouple location was determined to be ±0.56 mm. Equation 1 was used to calculate the error for the heat flux values reported. The uncertainty in the heat flux was determined to be 5.6% at 80 W/cm^2. Calculations indicated that heat losses within the upper neck of the copper block were less than 1% of the total heat input at CHF for the flat surface case. Spray cooling heat flux demonstrated repeatability within 1%

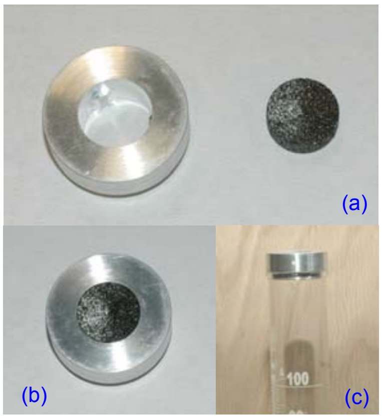

Fig. 6. Foam liquid flow rate test apparatus; (a) aluminum holder and Poco foam sample, (b) Poco foam inside aluminum holder, (c) Poco foam and holder at top of graduated cylinder

for multiple tests under identical test conditions. Pressure values had an uncertainty of ±3 kPa. Flow meter measurements had an error of ±1 ml/min.

$$\delta\dot{q}'' = \pm\sqrt{\left(\frac{\partial\dot{q}''}{\partial x}\delta_x\right)^2 + \left(\frac{\partial\dot{q}''}{\partial k}\delta_k\right)^2 + \left(\frac{\partial\dot{q}''}{\partial(\Delta T)}\delta_{\Delta T}\right)^2} \quad (1)$$

RESULTS AND DISCUSSION

Heat flux performance as a function superheat and volumetric flux for the flat and Poco Foam surfaces are shown in Figs. 7 and 8 respectively. The calculated heat flux is based on the projected heater surface area (2.0 cm^2) for each of the cases.

Flat Surface. In each of the flat surface study cases, tests where performed up to CHF. Volumetric fluxes tested ranged 0.010 m^3/m^2-s (120 ml/min) to 0.016 m^3/m^2-s (200 ml/min). The heat transfer variation for all volumetric fluxes is linear in the low heat flux regime (which is indicative of single phase convection). For the lower volumetric flux cases (i.e., volumetric fluxes of 0.010 m^3/m^2-s and 0.011 m^3/m^2-s), multiphase effects become pronounced (denoted by the increase in slope of the heat flux curves) at ΔT_{sup} ≈ 14°C. For the other flow rate cases, multiphase effects become pronounced between ΔT_{sup} ≈ 19°C and 21°C. As the volumetric flux increases from 0.010 m^3/m^2-s to 0.016 m^3/m^2-s, there is a noticeable increase in heat flux throughout the spray cooling curves. However, the spray cooling curves for the 0.015 m^3/m^2-s and 0.016 m^3/m^2-s cases are nearly identical (i.e., well within the experimental uncertainty), indicating that for the test conditions used, volumetric fluxes greater than 0.015 m^3/m^2-s provide diminishing returns upon the heat flux. The maximum CHFs (79 W/cm^2 and 80 W/cm^2) occurred for the 0.015 m^3/m^2-s and 0.016 m^3/m^2-s cases respectively.

Poco Foam Surface. Similar to the flat surface test cases, volumetric fluxes tested ranged 0.010 m^3/m^2-s (120 ml/min) to 0.016 m^3/m^2-s (200 ml/min). The superheat was defined as the difference between the

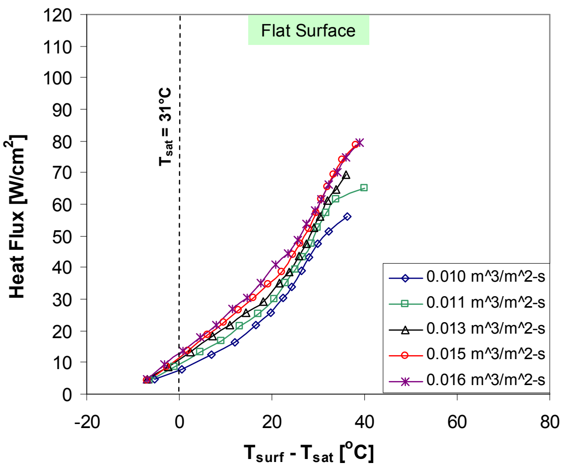

Fig. 7. Heat Flux as a Function of Superheat and Volumetric Flux for Flat Surface

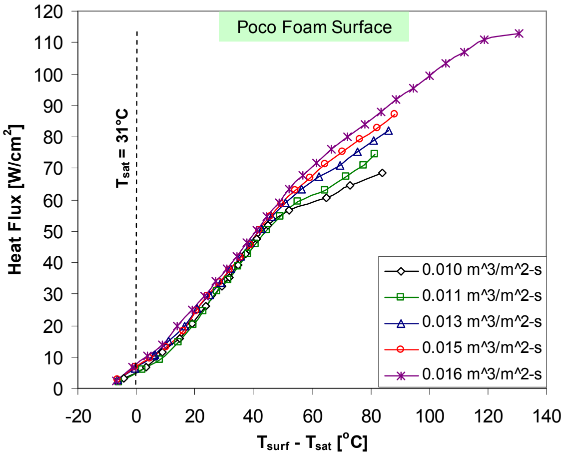

Fig. 8. Heat Flux as a Function of Superheat and Volumetric Flux for Porous Foam Surface

copper surface temperature at the copper/foam bondline and the working fluid saturation temperature. Due to concerns regarding possible failure of the copper/foam TIM at elevated temperatures, as well as initial concerns regarding possible dissociation of harmful Fluorine gas from the PF-5060 working fluid (Arnold et al., 2007), tests for the lower volumetric fluxes (i.e., 0.010 m^3/m^2-s to 0.015 m^3/m^2-s) were performed up to a temperature of ≈115°C for the copper surface at the copper/foam interface. Heat flux testing at the largest volumetric flux (0.016 m^3/m^2-s) was performed up to CHF. The study showed that variation in heat flux for the different volumetric fluxes was negligible (i.e., within the experimental uncertainty of the measurements) at $\Delta T_{sup} \leq 45$°C. In the low heat flux regime (i.e., $\dot{q}''$ <15 W/cm^2) the overlaying spray cooling curves have a linear heat transfer variation with superheat. At ΔT_{sup} ≈15°C, the slope of the spray cooling curves increases slightly, yet the heat flux to the copper surface superheat relationship remains linear up to 45°C superheat. At superheats greater than 45°C, the spray cooling curves separate and the heat flux increases with volumetric flux. Maximum heat flux (113 W/cm^2) for the volumetric fluxes tested occurred for the 0.016 m^3/m^2-s case.

EFFECTS OF POROUS FOAM

Flat vs. Poco Foam Comparison. In order to properly determine the viability of Poco Foam as a heat transfer enhancement technique for spray cooling applications, heat flux performance must be examined relative to that of the flat surface cases. Table 3 has a summary of the data for the flat and Poco Foam surfaces. Included in the table is the volumetric flux onto the heater surface, CHF (or maximum heat flux attained), maximum surface temperature observed and the multiphase evaporation efficiency. As shown in Table 3, use of the Poco Foam provided a noticeable improvement in maximum heat flux performance (as well as multiphase evaporation efficiency) relative to that of the flat surface at comparable volumetric fluxes. However, review of Figs. 7 and 8 show that heat flux performance for each of the flat surfaces exceeded that of the Poco Foam surface at comparable superheat levels throughout their spray cooling curves (up to CHF for the flat surfaces). The Poco Foam cases attained heat fluxes greater than that of the flat surface cases only after having reached extensive superheat levels (i.e., ΔT_{sup} > 45°C). In addition, heat flux variation as a function of volumetric flux was noticeable in the spray cooling curves for the flat surface cases throughout the single phase and multiphase regimes whereas in the Poco Foam cases there was negligible heat flux variation for superheat levels below 45°C. From a thermal resistance network perspective, addition of the Poco Foam onto the copper heater surface increases the overall resistance to heat flow between the copper heater and the working fluid. Thus, the elevated superheat observed for the copper surface at the copper/foam interface may be considered a byproduct associated with use of the Poco Foam in the present configuration (notwithstanding possible TIM resistance at the bondline). Nonetheless, additional thermal resistance provided by the Poco Foam does not address the overlay of the spray cooling curves at ΔT_{sup} ≤ 45°C. However, investigation of the heat transfer and

Table 3. Summary of Heat Flux Test Data

Surface Description	$\dot{V}''$ (m^3/m^2-s)	$\dot{q}''_{max}$ (W/cm^2)	ΔT_{max} (°C)	$\eta_{2\text{-}\Phi}$ (%)
‡ Flat Surface	0.010	56†	36.1	32.6
‡ Flat Surface	0.011	65†	40.0	34.4
‡ Flat Surface	0.013	69†	36.0	30.9
‡ Flat Surface	0.015	79†	38.0	30.6
‡ Flat Surface	0.016	80†	40.0	29.1
Poco Foam	0.010	68§	84.0	39.6
Poco Foam	0.011	74§	81.5	39.1
Poco Foam	0.013	82§	86.0	36.4
Poco Foam	0.015	87§	88.1	33.6
Poco Foam	0.016	113†	130.6	40.8

† CHF was attained
§ Maximum heat flux taken at T_{surf} ≈ 115°C
‡ CHF previously reported by Silk (2006)

fluid flow interior to the Poco Foam may provide some insights into this phenomenon.

Poco Foam Interior Phenomena. As mentioned previously, spray cooling is a multiphase convective heat transfer process. The convective heat flux is the product of the convection coefficient (which is a function of the thermophysical properties of the working fluid and the flow) and the heater surface to liquid temperature difference. Since the foam sample and test conditions were held constant for each of the cases and the heat flux was constant up through $\Delta T_{sup} \approx 45°C$, the heat transfer effects observed are highly dependent upon the flow (either at the top or in the foam's interior). Given the pore size and tortuous void space in the foam, one might speculate that the foam structure impedes liquid passage through it. Since the heat transfer is highly dependent upon the flow, the amount of liquid flow resistance and/or conductance (i.e., permeability) through the foam is of interest and should be quantified. For porous media applications (i.e., packed beds, foam structures, etc.) the Darcy law is often used to determine the permeability of the porous structure (Kaviany, M., 1995). According to the Darcy law, the permeability is defined as having functional dependence upon the pressure gradient across the porous media in question. In the spray cooling process a pressure differential is maintained between the nozzle (or spray manifold) and the evaporator environment in order to have continuous droplet atomization. Upon droplet impingement of the liquid film on the heater surface, the bulk fluid motion of the film is considered inertia driven. Thus Darcy's law is not applicable to the liquid flow being examined in the present test configuration. As an alternative, volumetric flow measurements for liquid passing through the foam (previously described in the Test Setup and Procedure section) were performed. After having completed the volumetric flow rate measurements, the effective liquid velocity through the foam was calculated using eqn. 2.

$$v_{eff} = \frac{\dot{V}_{Foam}}{A_{fs}} \tag{2}$$

where $\dot{V}_{Foam}$ is the volumetric flow rate through the foam. Also calculated were the Reynolds number (ratio of inertia to viscous forces) and the Peclet number (ratio of liquid advection to thermal diffusion through the media) for each of the volumetric fluxes tested. Each of these values are listed in Table 4 along with the percentage of the volumetric flow transiting through the foam relative to the volumetric flow incident on the top surface of the Poco Foam. Definitions for the Reynolds and Peclet numbers are provided in the nomenclature section.

The measurements showed that the amount of liquid passing through the Poco Foam increased with volumetric flow rate impinging the foam's top surface. However, in each of the flow rate cases, the percentage of the volumetric flow passing through the Poco Foam was < 3.5%. For the maximum flow rate case (i.e., volumetric flow of 3.2x 10^{-6} m^3/s) this gave a volumetric flow rate through the foam of 1.12 x10^{-7} m^3/s. In addition, v_{eff} (and subsequently the Re and Pe) increased

Table 4. Summary Data for Liquid Flow Through Poco Foam Sample

$\dot{V}$ x 10^6 (m^3/s)	% of $\dot{V}$ through Foam	v_{eff} (m/s)	Re	Pe
2.0	1.9	0.0053	333	2570
2.2	2.2	0.0067	426	3282
2.6	2.4	0.0075	476	3670
3.0	2.9	0.0104	659	5076
3.2	3.5	0.0139	881	6791

with volumetric flow rate. While the effective liquid velocities through the foam were fairly low (i.e., << 1m/s) the Reynolds numbers were greater than unity,which implied that creeping flow was not occurring. Also, the Peclet number for each of the cases was greater than unity, suggesting that advection had a dominant effect over thermal diffusion. Using the measured volumetric flow rate through the foam for the test cases, the maximum theoretical heat transfer available in both the single and multiphase heat transfer regimes can be calculated using eqns. 3 and 4.

$$\dot{q}''_{1-\phi} = \rho_l \dot{V}'' c_{p,l} \left(T_{sat} - T_{liq}\right) \tag{3}$$

$$\dot{q}''_{2-\phi} = \rho_l \dot{V}'' \left[c_{p,l} \left(T_{sat} - T_{liq}\right) + h_{fg}\right] \tag{4}$$

For the maximum volumetric flow rate transiting through the foam at a given instant (i.e., 1.12 x10^{-7} m^3/s based on the cases studied), it was determined that the single and multiphase heat flux available was 1.1 W/cm^2 and 8.6 W/cm^2 respectively. Each of these values are below the actual experimental heat flux measured for the single phase (7.0 W/cm^2 at ΔT_{sup} = 0°C) and the maximum heat flux in the multiphase regime where the spray cooling curves overlay (e.g., 55.0 W/cm^2 at $\Delta T_{sup} \approx$ 45°C). Since the theoretical heat flux available given the maximum volumetric flow of liquid internal to the foam under any of the test cases studied is only 15% of the measured heat flux for the test cases, the predominant location for heat transfer must be at the liquid/foam interface along the top of the foam. This is in agreement with the determinations of Jamin and Mohamad (2007). In their investigation of forced convection heat transfer (working fluid was air) over a porous foam surface, they concluded that the heat transfer between the foam and the working fluid was limited to the outer surface of the foam at the the gas/foam interface. The concept that heat transfer for porous media is predominantly a surface phenomena has also been reported when using porous foam in pool boiling applications (Coursey et al., 2005). Also, Wong and Dybbs (1975) experimentally showed that the temperature difference between the working fluid interior to a packed bed and the structure particles was approximately zero. Without sufficient temperature gradient, heat transfer in the local measurement region was not possible. Since the experimental heat transfer data in the present study shows that heat transfer is predominantly occurring at the top of the foam (i.e., at the liquid/foam interface), heat flux performance for the Poco Foam's spray cooling curves throughout all levels of superheating may be considered a function of convective phenomena at the top surface, as well as the conductive path leading to the top of the foam surface. General observations during each of the Poco Foam heat flux experiments showed that there was significant droplet rebound occurring from the top of the surface. However, it is unknown as to whether or not this had significant contribution to the heat transfer effects at $\Delta T_{sup} \leq 45°C$. Future work may include the investigation of alternative bonding techniques for determination of their effect upon heat transfer and/or improved in-situ visualization of the liquid/foam interface using sparse sprays.

CONCLUSIONS

Spray cooling heat flux measurements were performed on a Poco Foam enhanced surface as well as a flat surface using PF-5060. Tests were performed under nominally degassed conditions (fluid at 41.4 kPa) for volumetric fluxes ranging 0.010 m^3/m^2-s to 0.016 m^3/m^2-s. The nozzle distance relative to the copper heater surface (17 mm) was held constant for all the tests.

The flat surface showed diminishing returns upon the heat flux above a volumetric flux of 0.015 m^3/m^2-s. The spray cooling curve for the 0.015 m^3/m^2-s and 0.016 m^3/m^2-s cases were nearly identical. Maximum CHF for the flat surface case was ≈ 80 W/cm^2.

The Poco Foam showed heat flux enhancement relative to the flat surface for each of the volumetric flux cases only after reaching extensive superheat levels. The superheat levels attained with the Poco Foam were primarily attributed to the addition of the foam's thermal resistance to the heat flow path. Also, accompanying volumetric flow measurements for the liquid flow through the foam showed that in each of the test cases heat transfer is occurring predominantly at the foam surface experiencing droplet impingement.

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NOMENCLATURE

ρ_l	liquid density, kg/m^3
A	Area
P	Pressure
T	Temperature
TC	thermocouple
c_p	specific heat, J/kg°C
h_{fg}	enthalpy of vaporization, kJ/kg
k	thermal conductivity, W/m-K
$\dot{q}''$	heat flux per unit area
$\dot{V}''$	volumetric flux, m^3/m^2-s
$\dot{V}$	volumetric flow rate, m^3/s
$\dot{V}_{Foam}$	foam interior volumetric flow rate, m^3/s
v_{eff}	effective velocity through foam, m/s
x	distance from heater surface within heater, m
Re	$\rho_l v_{eff} d_h/\mu_l$
Pr	$\mu_l c_{p,l}/k_l$
Pe	Re·Pr
ΔT_{sup}	$T_{surf}-T_{sat}$, °C
$\delta\dot{q}''$	error in heat flux
δ_k	error in conductivity
$\delta_{\Delta T}$	error in thermocouple temperature difference
δ_x	error in thermocouple location
$\eta_{2\text{-}\Phi}$	$\dot{q}''/\rho\dot{V}''\left[c_{p,l}(T_{sat}-T_l)+h_{fg}\right]$

Subscripts

flat	flat surface
fs	foam structure/working fluid interface
k	conductivity
l	liquid
CHF	critical heat flux
max	maximum
sat	saturation conditions
surf	surface
T	temperature
x	thermocouple distance
1-Φ	single phase
2-Φ	multiphase

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