Flow Regimes and Convective Heat Transfer of Refrigerant Flow Boiling in Ultra-Small Clearance Microgaps

Mohamed H. Nasr,^a Craig E. Green,^a Peter A. Kottke,^a Xuchen Zhang,^b Thomas E. Sarvey,^b
Yogendra K. Joshi,^a Muhannad S. Bakir,^b Andrei G. Fedorov^{a,*}

^aGeorge W. Woodruff School of Mechanical Engineering

^bSchool of Electrical and Computer Engineering

Georgia Institute of Technology,

Atlanta, GA 30332, USA

*Corresponding Author: *AGF@gatech.edu*

Abstract

Understanding two-phase convective heat transfer under extreme conditions of high heat and mass fluxes and confined geometry is of fundamental interest and practical significance. In particular, next generation electronics are becoming thermally limited in performance, as integration levels increase due to the emergence of ‘hotspots’ featuring up to ten-fold increase in local heat fluxes, resulting from non-uniform power distribution. An ultra-small clearance, 10 μm microgap, was investigated to gain insight into physics of high mass flux refrigerant R134a flow boiling, and to assess its utility as a practical solution for hotspot thermal management. Two configurations - a bare microgap, and inline micro-pin fin populated microgap - were tested in terms of their ability to dissipate heat fluxes approaching 1.5 kW/cm^2 . Extreme flow conditions were investigated, including mass fluxes up to $3,000 \text{ kg/m}^2\text{s}$ at inlet pressures up to 1.5 MPa and

exit vapor qualities approaching unity. Dominant flow regimes were identified and correlated to two phase heat transfer coefficients obtained using model-based data reduction for both device configurations. The results obtained were compared to predictions using correlations from literature, with the maximum heat transfer coefficient reaching $1.5 \text{ MW/m}^2\text{K}$ in the vapor plume regime in the case of the finned microgap.

Keywords: Microgap; Pin fins; Boiling; Two-phase heat transfer coefficient; Flow regime; Pressure drop.

1. Introduction

Two-phase flow and heat transfer in microgaps have received significant attention due to their ability to dissipate high heat fluxes and maintain small wall temperature gradients, while being less prone to flow instabilities than microchannel geometries [1-8]. Recent studies have shown increasing two-phase heat removal rates in microgaps with decreasing gap height, with thin film convective boiling being the dominant heat transfer mechanism [3-4]. Prior to the advancement of microfabrication processes, studies were conducted with mini-channels ($D_h \geq 1\text{mm}$), which differed from microgaps because nucleate boiling was the dominant heat transfer mechanism [2, 9-11]. With departure of the observed behavior from predictions of the macroscale convective boiling theory and a potential for high heat flux dissipation, there is a need to investigate microgap flow boiling in ultra-small clearance geometries.

Boiling heat transfer in microgaps is strongly linked to flow boiling regime, fluid properties and key operating parameters, such as quality, mass flux and heat flux [2, 12, 17, 18]. Thin film boiling has been reported to yield large heat removal rates because of the low thermal resistance of the liquid film wetting the microgap surface [2,12,17]. As the microgap hydraulic

diameter decreases, this boiling mechanism becomes more prevalent due to increased bubble confinement [2]. In the absence of fundamental understanding of two-phase flow in microgaps, empirical correlations are generally used to predict heat transfer coefficient, CHF and pressure drop, often for a limited range of operating conditions that cannot be extrapolated to variations in geometry or coolant [14, 17, 19-23]. Heavy reliance on empirical correlations for predicting two-phase behavior in microgaps is a major limitation in this field of study. While correlations have shown increased robustness when developed for a specific flow regime, the absence of a universally accepted standard that quantitatively defines well established flow regimes makes it difficult to apply flow boiling correlations outside the operating range they were developed for.

The primary flow regimes observed in two-phase flow through miniature horizontal gaps are bubbly, intermittent, annular and stratified [14, 17,18]. Bubbly flow is defined by the flow of spherical vapor bubbles surrounded by liquid. Intermittent flow is defined by elongated vapor plugs formed by the agglomeration of bubbles that are surrounded by liquid. The stratified flow regime is only observed in horizontal flow, and refers to a stratification of liquid and vapor layers, with the vapor phase flowing on top of the liquid. Annular flow is defined by a thin liquid film, which covers the top and bottom surfaces of the gap, with a vapor core in between. In the annular flow regime, thermal resistance due to conduction in the liquid film wetting the microgap surface is substantially reduced as the liquid film becomes thinner with increases in vapor quality. Thinning of the liquid film reduces the distance over which heat needs to conduct or, alternatively, increases the effective heat transfer coefficient. Annular flow was shown to be the most prevalent flow regime in microgap geometries according to several studies [2, 4, 14, 17]. High heat removal rates are achieved in convective thin film evaporation regimes such as annular flow due to the reduction in conductive resistance that occurs when the-liquid-vapor

interface is brought closer to the heated wall of the microgap and the acceleration of the liquid film, driven by the expanding vapor phase through shear stresses at the liquid-vapor interface.

Bar Cohen et al [14] have described an M-shaped trend of two-phase heat transfer coefficient over a broad range of vapor qualities, ranging from subcooled boiling and approaching unity at the exit, for flow boiling in microgaps. The M-shaped trend contains inflection points corresponding to flow regime transitions, and reflects thermophysical phenomena of two-phase flow in microgaps. The initial increase in heat transfer coefficient in the subcooled domain corresponds to an acceleration of two-phase flow with onset of nucleate boiling. Bubble agglomeration limits increases in heat removal rates, and transition to the intermittent flow regime leads to a reduction in heat transfer performance due to periodic wall dryout from vapor slugs. At moderate qualities (15-40%), the transition to annular flow leads to an increase in heat transfer coefficient, as thin film convective boiling dominates the heat transfer mechanism. Maximum heat fluxes are achieved at high qualities (50-75%), as the liquid film in annular flow becomes thinner, and is followed by a decline in heat transfer coefficient with the onset of local dryout

In this study, flow boiling of R134a was experimentally investigated in a 10 μm microgap to assess its capability to dissipate ultra-high heat fluxes. Additionally, flow boiling in a microgap enhanced with micro pin fins was studied for the same geometry and test structure dimensions. To the best of our knowledge, the geometry investigated represents an extreme in the dimensions and flow conditions, which has not been previously examined. These include heat fluxes up to $\sim 1.6 \text{ kW/cm}^2$, coolant mass fluxes up to $3,000 \text{ kg/m}^2\text{s}$ at inlet pressures up to 1.5 MPa, and exit qualities approaching unity. Our goal is to understand the fundamental physics of two-phase heat transfer under these extreme operating conditions. Flow boiling visualizations

for each device are presented, and dominant regimes are interpreted in terms of their underlying flow and heat transfer mechanisms. A numerical model was developed to extract two-phase heat transfer coefficient, quality and heat flux to provide a comprehensive evaluation of microgap thermal characteristics. Two-phase heat transfer coefficient results were also compared to predictions using correlations found in literature.

2. Experimental Testbed

An overview of the bare microgap and pin fin microgap test structures is shown in Fig. 1 supplemented by cross section views and SEM images. Subcooled fluid enters the device through a 200 μm diameter inlet port and flows through the inlet plenum, which is 50 μm deep to minimize parasitic pressure drop at the fluid entry/exit domains. The microgap test section is 300 μm long x 200 μm wide x 10 μm tall and is located in the middle of the device with a platinum resistance heater deposited on the back side. The heater generates a controlled heat load through Ohmic heat generation and also serves as a resistance temperature detector (RTD) for temperature measurement. Three additional RTDs are located on each side of the heater (six total), orthogonal to the coolant flow direction, solely for temperature measurement in the vicinity of the heater (Fig 1-a). The RTDs are 80 μm x 55 μm spaced at ~ 17.5 μm between each other.

Pyrex glass seals the top of the microgap and allows for flow visualization. Air trenches are 40 μm wide and etched 180 μm deep into silicon from the back (heater) side of the device to reduce conduction heat spreading in the bulk silicon. The pin fin device contains an array of 32x20 inline cylindrical pin fins, which are 10 μm tall (i.e., no clearance to the top of the microgap), 4 μm in diameter and 10 μm apart contributing a surface area enhancement factor of

2.19. Device features including micro pin fins, air trenches, ports and plenums were etched in silicon using Bosch process with high precision and accuracy. The microfabrication process flow for these devices was documented by Green et al [24].

(a) Device Overview

Fig. 1 Overview of microgap testbed: (a) panoramic view of device (b) cross section of entire device (c) SEM images of heater and pin fin microgap (d) pin fin microgap cross section (e) bare microgap cross section. **(2-column fitting image)**

3. Experimental Setup & Procedure

3.1 Experimental Setup

The experimental test loop used in this study is shown in Fig 2. The devices were housed in a machined PEEK package with O-ring seals for the inlet/outlet ports and pressure taps as shown in Fig 3. An Agilent 34970a data acquisition unit was used to record pressure drop, heater

resistance, circuit current, inlet/outlet fluid temperatures and reservoir temperature for various flow rates of the working fluid. A KDS Scientific Legato 270 series syringe pump was used to supply a refrigerant to the test section at a prescribed flow rate. Fluid temperature measurements were obtained with Omega K-type thermocouples. Pressure drop was measured with Omega PX 309 series pressure transducers, which are connected to the pressure ports of the tested devices shown in Fig 1. Power was supplied to the device heaters with an Agilent E3641A power source. A fan cooled WBA series thermoelectric was used to condense vapor coming out of the test section. The reservoir tank was heated by electrical wire heaters with an Omega CN4000 PID controller to pressurize and fill the system with a refrigerant. A Keyence VH-Z100R microscope was used to obtain flow visualization images and videos. Microscope flow visualization images show a top-down view of the microgap test section as shown in Fig 3.

Fig 2 Schematic of experimental test loop. (2-column fitting image)

Fig 3 Exploded view of test section and hermetic package. Flow visualization obtained from a top-down view of the microgap with microscope. Pressure ports and thermocouple ports in package not shown. **(Single-column fitting image)**

3.2 Experimental Procedure

Before starting experiments, device heaters were calibrated in a temperature controlled vacuum oven, showing consistent linear correlation between the RTD resistance and temperature. Experiments started by evacuating the experimental setup to remove most of residual air from the test loop before charging with R134a. The reservoir tank, containing refrigerant, was pressurized by external heating using coil heater (Fig. 2) to ensure complete filling of the experimental loop with liquid. Mass fluxes between 1,000 and 3,000 kg/m²s were tested by setting the syringe pump to the desired flow rate and subcooled R134a was delivered to the test device at 22°C inlet temperature and inlet pressures between 860-1090 kPa. Power was

applied to the heaters in 0.1-0.25W increments, ensuring that steady state temperature and pressure readings were obtained at each power level. Power to the heaters was turned off when local dryout was visually observed in the microgap or when inlet pressure started to approach glass syringe limits to avoid catastrophic failure. Flow visualization was performed using a microscope with up to 700X lens magnification and continuous video capture with 0.067s time resolution.

3.3 Uncertainties

Uncertainty in T-type thermocouple (standard) used for heater calibration was $\pm 1.0^{\circ}\text{C}$, uncertainty in power applied to heaters from Agilent E3641A power source was $\pm 0.011\text{W}$, uncertainty in mass flow rate from syringe pump was $\pm 0.01\text{ mL/s}$, and uncertainty in pressure transducer measurements was $\pm 8.62\text{ kPa}$. Uncertainty in microgap height, air trench depth, and pin fin height is within $\pm 5\%$ of reported dimensions, and uncertainty in heater length and width are within $\pm 1\%$ of reported dimensions. Error propagation was applied to assess the uncertainty in mass flux and heater heat flux, which were found to be $\pm 6.5\%$ and $\pm 1.5\%$ of measured value, respectively.

4. Data Reduction

Direct measurement of heat losses in the microgap testbed was not possible in the limit of measurement uncertainties due to the low heater power ($<3\text{W}$) required to achieve very high heat fluxes from small footprint area heaters and the small microgap size ($10\text{ }\mu\text{m}$ tall) with respect to the silicon layer thickness ($240\text{ }\mu\text{m}$), resulting in significant heat spreading by conduction even in presence of isolating air gaps. With a low power input supplied at the heater and significant losses due to spreading, the heat input to the microgap falls within the measurement uncertainty

and could not be accurately quantified to measure the heat flux to the fluid and heat transfer coefficient. As a solution, a 3D finite element heat transfer model was built in COMSOL 5.2 to extract all thermal characteristics pertaining to the microgap, including the local heat flux, the average two-phase heat transfer coefficient, exit vapor quality, and wall temperature in the microgap from experimental data for the investigated testbeds. To ensure accurate accounting of all heat transfer pathway and losses, the 3D computational domain had to include the microgap, silicon test chips, the printed circuit board (PCB) and the package with all relevant design features. The thermal interfaces and variation of thermal conductivity of different materials with temperature were also included. After careful validation of the model using benchmark experiments with an evacuated device, the model was applied to the cases of refrigerant boiling in different two-phase flow regime to extract the average heat transfer coefficient as the only “free” model parameter by iteratively matching the predicted heater temperature with the measured value. A discussion of the heat transfer model used for data reduction, boundary conditions applied and model validation, can be found in sections S-2 through S-5 of the Supplemental Information.

5. Results & Discussion

5.1 Flow Boiling Visualization

Flow visualizations supplemented by illustrated interpretations of the boiling flow regimes for high and low mass fluxes in the bare microgap device are presented in Figs. 4 and 5, respectively. In addition, videos of these flow regimes can be found in section S-6 of the Supplemental Information. The flow visualization images and videos show a top-down view of the microgap with flow going from left to right. The dominant boiling flow regimes from flow

visualization are vapor plume, liquid slug, and liquid film boiling, which are mapped to domains that show change in trends of heat transfer coefficient variation in the next section. The vapor plume flow regime shown in Fig 4-b and c features nucleation points forming elongated plumes or slugs that may either be large enough to span the entire microgap or be shorter in length than the microgap but moving so fast that they appear to span the entire microgap (imaging artifacts due to aliasing). Both possible scenarios are shown in the cross section illustrations, Fig 4-b-2 and 4-b-3. The latter interpretation is proposed because the vapor is moving faster than the microscope frame rate can capture vapor slug boundaries. The hydraulic entrance lengths for laminar flow of R-134a as a saturated liquid at the tested mass fluxes of 1,000, 2,000 and 3,000 kg/m²s are 93, 185 and 278 μ m respectively. Nucleation sites generally appeared in the developing region of the 300 μ m long microgap the mass fluxes tested. At low mass fluxes (<2000 kg/m²s), boiling transitions from the outlet plenum directly to the inlet plenum, with vapor slugs spanning the entire length of the microgap as shown in Fig 4-b. At high mass fluxes ($\geq$ 2000 kg/m²s), nucleation points emerge near the outlet of the microgap, and move upstream as heat flux is increased. Single phase liquid entering the microgap is heated until it reaches saturation temperature, at which point a nucleation front is observed. The extent of single phase flow in the microgap decreases as the heat flux is increased, because the fluid reaches saturation temperature more rapidly; this manifests in gradual movement of the nucleation front upstream with an increase in applied heat flux. As the nucleation sites move toward the microgap inlet, the liquid layer wetting the microgap surface in the region downstream of nucleation is expected to become progressively thinner, as quality and void fraction increase. Improved thermal performance resulting from thinning of the liquid layer wetting the microgap surface is expected to occur, because the vapor core is moving substantially faster than the liquid (due to the density

difference) and accelerates the liquid by tangential stresses at the phase interface. The vapor plumes form a curved nucleation front in bare microgap devices, with more boiling in the center of the microgap, and less near the side walls, due to local variations in wall temperature as the side walls are slightly cooler than the center of the microgap due to heat spreading.

(a-1)

(a-2)

(a-3)

(b-1)

(b-2)

(b-3)

Fig. 4 Boiling flow visualization in the bare microgap device at $3,000 \text{ kg/m}^2\text{s}$ with cross section schematics of flow regime interpretation: (a) Vapor plume flow; (b) Liquid slugs flow; (c) Liquid film breaking up into slugs **(Single-column fitting image)**

Fig. 5 Boiling flow visualization in the bare microgap device at mass fluxes $<2,000 \text{ kg/m}^2\text{s}$, showing (a) vapor plume boiling spans the entire microgap as nucleation points emerge in the inlet plenum; (b) transition of vapor slug boiling directly to liquid slugs with condensation on glass surface is observed without an intermediate thin film regime. **(Single-column fitting image)**

As heat flux is increased, the two-phase flow regime transitions to liquid slug boiling which is similar to the rivulet flow structure described Bar Cohen et al in [25]. At low mass fluxes the liquid layer beneath the vapor slugs gets thinner until the heated gap surface becomes partially dry and liquid slugs are pushed across as shown in Fig 5-b. Since the flow visualization

images show a top-down view, the height of the slugs may be as tall as the gap height or smaller; both possibilities are illustrated in Fig 4-b-2 and 4-b-3. At high mass fluxes, a thin liquid film was observed before it broke up into liquid slugs. Capillary cohesion forces are suspected to play the main role in keeping the film intact. An ultra-thin wavy liquid film was observed at the highest heat flux in a bare microgap device with 5 μ m tall gap height. Details of this flow regime are discussed in section S-1 of the Supplemental Information

The dominant two-phase flow regimes observed in pin fin devices follow those of bare microgap devices and include vapor plume boiling and liquid slug boiling. Representative flow visualization images of these boiling regimes are presented in Fig. 6. The illustrated interpretations are similar to those shown for the same regimes observed in the bare microgap device. No differences in flow regime transition, or behavior between low and high mass fluxes were observed in the pin fin device, likely because of the presence of fins which disrupt the flow patterns and homogenize the flow structure regardless of the magnitude of mass flux. Vapor slugs and liquid slugs appear in a parabolic pattern with more boiling near the side walls and less in the middle of the microgap, differing from bare microgap devices which showed more boiling in the center. Hydrodynamic resistance near the side walls from tight spacing of the pin fins results in locally lower flow rates at the side wall and therefore more boiling. It is difficult to determine whether a liquid film regime exists at high mass fluxes due to the small spacing between the fins, which prevents detailed optical visualization. The bare microgap devices allow us to recognize boiling regimes that were difficult to identify in the pin fin devices and compare heat transfer performance with and without fins. However, the pin fin devices are of more importance in application than bare microgap devices because the pin fin microgap produces

substantially higher two-phase heat transfer coefficients and additional surface area, thus enabling dissipation of substantially higher heat fluxes by the liquid.

Fig. 6 Visualization of pin fin device flow regimes: (a) vapor plume flow, (b) transition to liquid slugs flow, (c) liquid slug flow. **(Single-column fitting image)**

The curved profile of the nucleation front that was observed in bare microgap and pin fin devices has been repeatedly observed in experiments and is not an artifact of boiling hysteresis. A curved nucleation front was first observed as heat flux was increased, and nucleation sites moved towards the microgap inlet. When heat flux was subsequently decreased from its high value (to check for hysteresis) the nucleation sites moved back to their original locations and maintained the same curved nucleation front. In the bare microgap device, there was more nucleation in the center of microgap, and less near the side walls (Fig 4-a), because the side walls are cooler than the bottom surface of the microgap. In the pin fin devices, there were less

nucleation sites in the center of the microgap, and more near the side walls (Fig. 6-a), due to the tight spacing between the fins and side walls, which locally reduced the flow rate near the side walls and resulted in earlier boiling incipience.

5.2 Two-Phase Heat Transfer Coefficient

5.2.1 Heat Flux Dependence

Two-phase heat transfer coefficient as a function of heat flux at mass fluxes up to 3,000 kg/m²s for the bare microgap and pin fin devices are shown in Fig. 7 and 8. Two-phase heat transfer coefficients increased with increases in heat flux in the vapor plume regime, and decreased in the liquid slug regime. As the nucleation front of the vapor plumes moved towards the inlet of the microgap with increasing heat flux, thin film convective boiling with ultra-high heat removal rates dominated the heat transfer mechanism in the microgap causing a gradual increase in heat transfer coefficient. Increases in heat flux increased microgap wall superheat, which caused nucleation to move upstream towards the microgap inlet. When nucleation sites move toward the microgap inlet, the liquid film, wetting the microgap surface, becomes progressively thinner, thereby raising two-phase heat transfer coefficient. Fig. 7 and 8 also show that increasing mass flux increases two-phase heat transfer coefficient, enabling higher heat fluxes to be dissipated before transition to local dryout. Transition to the liquid slug regime occurs when the liquid film wetting the microgap surface breaks apart, resulting in a decrease in two-phase heat transfer coefficient as the dry patches on the microgap surface expand. Specific value of two-phase heat transfer coefficient in the liquid slug regime is not reported for the bare microgap device because boiling in the inlet plenum (which was observed during experiments) was not accounted for in the model, as explained in section S-2 of the Supplemental Information.

(a)

(b)

Fig. 7 Two-phase heat transfer coefficient vs heat flux in bare microgap device at various mass fluxes: (a) 3,000 kg/m²s (b) 2,000 kg/m²s. Results in the liquid slug regime cannot be presented due to inlet plenum boiling, which is not considered in the model used for data processing.

(a)

(b)

(c)

Fig. 8 Two-phase heat transfer coefficient vs heat flux in pin fin microgap device at various mass fluxes: (a) 3,000 kg/m²s (b) 2,000 kg/m²s (c) 1,000 kg/m²s. The mass flux is defined based on the total cross-sectional area of the test chip, i.e., not accounting for the area occupied by the pin fins. **(Single-column fitting image)**

The dense array of pin fins in the microgap increases heat rejection to the coolant through surface area enhancement, but more importantly increases two-phase heat transfer coefficient of the coolant by constantly breaking the thermal boundary layer growth. The magnitude of two-phase heat transfer coefficient increases between the bare microgap and pin fin microgap devices at the same mass fluxes, by as much as four-fold (after accounting for surface area of the fins) at the highest mass flux of 3,000 kg/m²s, from ~0.4 MW/m²K to ~1.5 MW/m²K. The large improvement in thermal performance is a result of thermal boundary layer interruption by the fins, and further thinning the liquid film covering the heat transfer surface. The thermal entrance lengths for R134a as a saturated liquid at 1000kg/ m²s, 2000 kg/m²s and 3000 kg/m²s are ~ 308 μm, 617μm and ~925 μm respectively, while the spacing between pin fins is 10μm. The tight

spacing of the fins prevents the coolant from thermally developing, and produces a remarkable improvement in thermal performance.

5.2.2 Quality Dependence

Two-phase heat transfer coefficient as a function of exit vapor quality at mass fluxes up to $3,000 \text{ kg/m}^2\text{s}$ for the bare microgap and pin fin devices is shown in Fig. 9. The bare microgap data in Fig 9-a is only presented for the vapor plume regime, for which the model assumption of single phase condition in the inlet plenum was most accurate as observed in experiments. The trend of increasing heat transfer coefficient with increasing quality in the vapor plume regime was expected owing to thinning of the liquid film and reduction of across the film heat transfer resistance. The decline in two-phase heat transfer coefficient in the liquid slug regime shown in Fig 9-b is due to the local dryout of the liquid film wetting the microgap surface.

(a)

(b)

Fig 9. Two-phase heat transfer coefficient vs exit vapor quality in: (a) bare microgap device at mass fluxes and (b) pin fin device. Note: horizontal error bars on quality for 1,000 kg/m²s data points in (a) and (b) are ± 0.1 and were removed for improved data presentation. **(Single-column fitting image)**

Bar Cohen et al. [11] introduced a characteristic M-shaped trend of two-phase heat transfer coefficient as a function of vapor quality for conventional (larger) size microgaps covering the full spectrum of quality. While our results are also shown for a broad range of vapor qualities, the proposed M-shaped trend was not observed for the ultra-small gap dimensions. This is likely due to the absence of the bubbly and intermittent flow regimes, which produce the first peak of the suggested M-shaped trend. The absence of these regimes is expected to be linked to the ultra-small clearance of the geometry investigated, which confines bubble growth to an extent where only one flow regime, vapor plumes, was observed before dryout. Several studies [2,4,6-7,24] have shown a similar phenomenon where the transition to annular convective

flow boiling occurs at lower vapor qualities as the microgap height decreases due to increased bubble confinement.

The high degree of confinement created in the 10 μm microgap forces vapor bubbles to grow in the span-wise direction upon boiling incipience, forming the observed vapor plumes. The vapor plume regime is similar to the annular regime in the presence of a shear-driven thin film boiling mechanism, defining the heat transfer process and producing high heat removal rates that are inversely proportional to liquid film thickness. With only one convective boiling regime persistent across the entire range of vapor qualities, the ultra-small microgap geometry shows less variation in heat transfer coefficient than larger scale microgaps, for which multiple flow regimes are observed resulting in varying heat removal rates. This is a unique advantage of ultra-small microgaps because the high heat removal rates associated with thin film boiling can be exploited over the full range of quality rather than a partial range.

5.3 Pressure Drop

Pressure drop as a function of heat flux for the bare microgap and fin devices is shown in Fig 10. Pressure drop steadily increases with heat addition in the vapor plume boiling regime due to acceleration between phases and viscous losses. The outlet pressure is always fixed at the system pressure due to the presence of a large buffering reservoir filled with a refrigerant's saturated mixture at a fixed temperature, so the change in inlet pressure directly correlates with the pressure drop.

(a)

(b)

Fig. 10 Pressure drop vs heat flux (a) bare microgap device (b) pin fin device at 820 kPa system pressure (downstream of microgap). Dashed vertical lines indicate the transition to the liquid

slug flow regime and onset of local dryout (not observed at $3,000 \text{ kg/m}^2\text{s}$). ‘VP’ denotes vapor plume domain and ‘LS’ denotes liquid slug domain. **(Single-column fitting image)**

The pin fin device followed the same pressure drop trend as the bare microgap device in the vapor plume flow regime. Note that pressure drop for the $1,000 \text{ kg/m}^2\text{s}$ case exceeds that of $2,000 \text{ kg/m}^2\text{s}$ case in liquid slug boiling because the transition to the liquid slug regime occurred at a lower heat flux for the lower mass flux, and therefore the vapor content is higher at the same heat flux, with corresponding increase in the acceleration and viscous pressure drops, despite the lower mass flux.

While two-phase heat transfer coefficient increased by a factor of ~ 4 in the pin fin microgap over the bare microgap, the improvement in heat removal rate comes at a cost of $\sim 10\times$ increase in pressure drop as shown in Fig 10. The tight spacing of the fin array, which allows for higher heat rejection to the coolant and effectively increases two-phase heat transfer coefficient, also constricts fluid flow in the microgap and increases viscous losses resulting in a large increase in inlet pressure.

5.4 Comparison with Existing Correlations

Two-phase heat transfer coefficients obtained in this study were compared to four published correlations, summarized in Table 1. Correlations 2-4, developed from flow boiling experiments in tubes, were compared to both bare microgap and pin fin devices. Correlation 5, developed for pin fin enhanced microgap boiling, has been only compared to the pin fin device.

Table 1 Two-phase heat transfer correlations

Correlation Number	Reference	Shape /Geometry	Working Fluid(s)	Correlation	Mean Absolute Error (MAE)*
1	Gungor-Winterton [22]	Saturated boiling in vertical and horizontal tubes	Water,R11 R12,R22, R113,R114, ethylene glycol	$h = h_l[1 + 3000Bo^{0.86} + \left(\frac{x}{1-x}\right)^{0.75} \left(\frac{\rho_l}{\rho_v}\right)^{0.41}]$	Bare Microgap: 250% Pin fin Microgap: 189%
2	Kandlikar [21]	Saturated boiling in vertical and horizontal tubes D=0.19-2.92 mm G= 50-570 kg/m2s	R-113, R-141b, HCFC 123	<p>For $100 \leq Re \leq 410$</p> $h_{TP,NBD} = 0.6683Co^{-0.2}(1-x)^{0.8}h_{LO} + 1058Bo^{0.7}(1-x)^{0.8}F_{Fl}h_{LO}$ $h_{TP,CBD} = 1.136Co^{-0.9}(1-x)^{0.8}h_{LO} + 667.2Bo^{0.7}(1-x)^{0.8}F_{Fl}h_{LO}$ $h_{LO} = Nu \left(\frac{k}{D_h}\right)$ <p>h_{TP} is the larger of $h_{TP,NBD}$ and $h_{TP,CBD}$</p>	Bare Microgap: 255% Pin fin Microgap: 358%
3	Shah [26]	Saturated boiling in vertical and horizontal tubes	Water, R11, R12, R113, cyclohexane	$\psi = \frac{h_{TP}}{h_l} = f(Bo, Fr_{le})$ $N_s = Co \text{ if } Fr_{le} \geq 0.04$ $N_s = 0.038Fr_{le}^{-0.3}Co \text{ if } Fr_{le} < 0.04$ $F_{sh} = 14.7 \text{ for } Bo \geq 11 \times 10^{-4}$ $F_{sh} = 15.4 \text{ for } Bo < 11 \times 10^{-4}$ $\psi_{CBD} = 1.8N_s^{-0.8}$ <p>For $N_s > 1$:</p> $\psi_{NBD} = 230Bo^{0.5} \text{ for } Bo > 0.3 \times 10^{-4}$ $\psi_{NBD} = 1 + 46Bo^{0.5} \text{ for } Bo \leq 0.3 \times 10^{-4}$ <p>For $0.1 < N_s \leq 1$:</p> $\psi_{BS} = F_{sh}Bo^{0.5} \exp(2.74N_s^{-0.1})$	Bare Microgap: 91% Pin fin Microgap: 100%

				<p>For $N_s \leq 0.1$:</p> $\psi_{BS} = F_{sh} Bo^{0.5} \exp(2.74 N_s^{-0.15})$ <p>$\psi = \text{greater of } \psi_{CBD} \text{ and } \psi_{NBD} \text{ or } \psi_{BS}$</p>	
4	Chen [20, 17] **	Saturated two-phase convective annular flow in vertical tubes	Water, methanol, cyclohexane, pentane, heptane, benzene	$h = h_{mic} + h_{mac}$ $h_{mac} = h_l F(X) Pr_l^{0.296}$ $h_l = 0.023 \left(\frac{k_l}{D} \right) Re_l^{0.8} Pr_l^{0.4}$ $Re_l = \frac{G(1-x)D}{\mu_l}$ $h_{mic} = 0.0122 \left[\frac{k_l^{0.79} c_{p,l}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29} h_{lv}^{0.24} \rho_v^{0.24}} \right]$ $\times [T_w - T_{sat}(P_l)]^{0.24} [P_{sat} - P_l]^{0.75} S$ $Re_{TP} = Re_l [F(X)]^{1.25}$ $S = (1.253 \times 10^{-6} Re_{TP}^{1.17})^{-1}$ $F(X) = 1 \quad \text{for } X^{-1} \leq 0.1$ $F(X) = 2.35 \left(0.213 + \frac{1}{X} \right)^{0.736} \quad \text{for } X^{-1} > 0.1$ $X = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_G}{p_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_G} \right)^{0.1}$	<p>Bare Microgap: 61%</p> <p>Pin Fin Microgap: 77%</p>
5	Krishnamurthy and Peles [16]	Rectangular microgap (1800 μm wide x 1cm long x 250 μm deep) with circular, staggered pin fins G=346-794 kg/m ² s 0<x _e <0.17	Water	$h_{TP} = \zeta (\phi_l^2)^{0.2475} Pr^{0.333} h_{sp}; \zeta = 1$ $h_{sp} = Nu \left(\frac{k}{d} \right)$ $Nu = 0.76 \left(\frac{S_t}{d} \right)^{0.16} \left(\frac{S_l}{d} \right)^{0.2} \left(\frac{L}{d} \right)^{-0.11} Re^{0.33}$ $\phi_l^2 = 1 + \frac{0.24}{X_{vv}} + \frac{1}{X_{vv}^2}$ $X_{vv} = [(\Delta P_f / \Delta Z)_f / (\Delta P_f / \Delta Z)_v]$ $(\Delta P_f)_f = \frac{f N (G(1-x))^2}{2 \rho_f}$ $(\Delta P_f)_v = \frac{f N (Gx)^2}{2 \rho_v}$ $f = 63.246 (Re_d)^{-0.7797}$	<p>Bare Microgap: N/A</p> <p>Pin fin Microgap: 69%</p>

* $MAE = \frac{1}{n} \sum_{i=1}^n \frac{|h_{experimental} - h_{predicted}|}{h_{experimental}} \times 100$; where 'h' refers to the two-phase heat transfer coefficient and 'n' refers to the number of data points (at one mass flux) compared to a correlation.

** The method described in reference [17] was used to compare our results to the Chen correlation.

Fig. 11 compares two-phase heat transfer coefficient as a function of quality at mass fluxes of 2,000 and 3,000 kg/m²s in the bare microgap device to correlations 1-3. It is clear that the Shah Correlation best predicted experimental two-phase heat transfer coefficients at the mass fluxes tested and is most accurate at high vapor quality. Although the authors of the Kandlikar correlation suggest using the larger value of two-phase heat transfer coefficient between a correlation for nucleate boiling heat transfer and that for convective boiling heat transfer, we only report values of the convective boiling heat transfer coefficient in this comparison. The prediction of two-phase heat transfer coefficient based on convective boiling yielded better approximations to experimental values and is more consistent with boiling observations than the nucleate boiling heat transfer coefficient, which was several times larger than the convective boiling value with physically plausible reasons.

(a)

(b)

Fig. 11 Comparison of two-phase heat transfer coefficient vs exit vapor quality to correlations 1-3 in Table 1 at mass fluxes (a) 2,000 kg/m²s and (b) 3,000 kg/m²s. **(Single-column fitting image)**

In general, correlations found in literature (with the exception of Chen [20]) largely overpredict two-phase heat transfer coefficient of the bare microgap device because the gap height tested (10 μm) is at least an order of magnitude smaller than the diameter for which most correlations were developed. This indicates that the rate at which two-phase heat transfer coefficient increases with decreasing gap height in the microscale generally does not conform to trends in larger scale channels due differences in heat transfer mechanism and boiling flow regimes. Furthermore, some correlations investigated tend to neglect the role of convective boiling dominant heat transfer in the microscale, with the Kandlikar correlation being a good example of such inconsistency. The Shah correlation, which performed best at high vapor quality, predicted that the boiling mechanism corresponded to a bubble suppressed regime

containing contributions from both nucleate and convective boiling. This emphasizes the importance of using the two-phase flow regime and the dominant boiling mechanism as the key proxies for selecting a proper empirical correlation. Fig. 11 shows that the experimental results tend to be lower bounded by the Chen Correlation and increase towards the Shah correlation as vapor quality increases. The Chen correlation, which has been commonly used in predicting two-phase heat transfer coefficient in annular flow driven regimes, significantly underpredict experimental results at high qualities, as shown in Fig.11, although it is surprisingly accurate at low qualities. The Chen correlation predicts two-phase heat transfer coefficient by summing contributions from a microscopic and macroscopic correlation for heat transfer coefficient [20]. In using this correlation to compare with our experimental results, it was found that the macroscopic heat transfer coefficient term contributes >75% of the predicted two-phase heat transfer coefficient value for nearly all data points. Since this correlation showed considerable success in predicting heat transfer coefficient for larger clearance microgaps [17, 27], the underprediction of experimental data shown in Fig. 11 suggests that the microscopic contribution to heat transfer coefficient term may be dominant for ultra-small clearance microgaps and underestimated in the Chen correlation. Rahim et al [27] similarly observed a large reduction in predictive accuracy of the Chen correlation in two-phase microgaps as gap height decreased from 500 μm (24% discrepancy) to 200 μm (34% discrepancy) to 100 μm (65% discrepancy).

Fig. 12 compares two-phase heat transfer coefficient as a function of quality at mass fluxes up to 3,000 $\text{kg/m}^2\text{s}$ in the pin fin microgap device to correlations 1-5. The Krishnamurthy and Peles correlation, developed for flow boiling of water in a staggered micro pin fin microgap, best captured the experimental results and overall trends obtained in this study. The introduction of pin fins into the microgap impacts the boiling process by interrupting thermal boundary layer

growth and homogenizing coolant flow through recirculation zones, resulting in a monotonically increasing heat transfer coefficient as a function of vapor quality, which differs from the concave function predicted by bare microgap correlations. A limitation of the Krishnamurthy and Peles correlation is that the prediction of heat transfer coefficient unphysically grows to singularity as quality approaches unity and therefore does not account for dryout. Since the Krishnamurthy and Peles correlation was developed for vapor qualities up to 0.17 and mass fluxes up to 794 kg/m²s, the correlation is not expected to capture the decrease in two-phase heat transfer coefficient at high vapor qualities in the liquid slug regime that was observed experimentally. Furthermore, our experimental data shows a stronger dependence on mass flux than was predicted by the Krishnamurthy and Peles correlation, which overpredicted heat transfer coefficient at 1,000 kg/m²s and under predicted results at 3,000 kg/m²s. Reeser et al [15] enhanced the Krishnamurthy and Peles correlation by modifying the ‘ ζ ’ constant into an expression with mass flux and exit quality dependence. Their adjusted correlation, fitted with empirical constants, showed remarkably low error (<15% MAE) for qualities approaching unity for inline and staggered finned microgaps with water and HFE-7200 as coolants. This suggests that the Krishnamurthy and Peles correlation has a potential to accurately predict two-phase heat transfer coefficient for a broad range of experimental conditions in finned microgaps and could be used as a credible starting point for further extension to other geometries and flow regimes in finned microgaps. The results presented here were not compared to the Reeser et al modification of the Krishnamurthy and Peles correlation due to their use of empirical constants, which have a strong dependence on the coolant used.

(a)

(b)

(c)

Fig. 12 Comparison of two-phase heat transfer coefficient vs quality to correlations 1-4 in Table 1 at (a) 1,000 kg/m²s (b) 2,000 kg/m²s (c) 3,000 kg/m²s. **(Single-column fitting image)**

6. Conclusions

In this study convective flow boiling of R134a was investigated in two microgap configurations: a bare microgap and pin fin populated microgap, both with 10μm gap height. Flow visualizations were obtained for heat fluxes up to ~1.6 kW/cm² at three mass fluxes: 1,000 kg/m²s, 2,000 kg/m²s and 3,000 kg/m²s, and inlet pressures up to 1.5 MPa. The main conclusions from this study are:

- Under high geometric confinement, the dominant flow boiling regimes observed are vapor plumes and liquid slugs. In the vapor plume regime, a liquid layer wetting the microgap surface becomes thinner with increases in heat flux and exit vapor quality, with an associated increase in the two-phase heat transfer coefficient. Local dryout occurs in

the liquid slug regime as the liquid film wetting the microgap surface breaks up and results in degradation of thermal performance.

- Two-phase heat transfer coefficient increased with increases in heat flux and quality in the vapor plume regime, but decreased in the liquid slug regime. The pin fin microgap showed up to ~4x increase in two-phase heat transfer coefficient (after accounting for additional surface area of the fins) over the bare microgap device at a mass flux of 3,000 kg/m²s with a maximum value of 1.5 MW/m²K based on the total (including pin fins) heat transfer area.
- Correlations found in literature over predicted two-phase heat transfer coefficient as a function of quality in the bare microgap device, but captured the general concave trend of the data and transition to dryout. The Shah correlation best predicted two-phase heat transfer coefficient with a high level of accuracy at high vapor quality by considering contributions from both nucleate and convective boiling terms.
- Correlations developed for flow boiling in tubes do not capture the trend of two-phase heat transfer coefficient as a function of quality in the pin fin microgap. The Krishnamurthy and Peles correlation, which was developed for flow boiling of water in a pin fin enhanced microgap, best predicted two phase heat transfer coefficient and captured the trend of experimental data. This correlation deviated from experimental results at the transition to the liquid slug regime and should be extended to higher exit vapor qualities and mass fluxes to account for dryout.

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