WAVES, INSTABILITIES, AND RIVULETS IN HIGH QUALITY MICROGAP TWO-PHASE FLOW

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Abstract. Two-phase flow in sub-millimeter microgap channels offers highly potent thermal management capability and is the foundation for the emerging "embedded cooling" paradigm of electronic cooling. While the thermofluid characteristics and operational limits of such microcoolers are intimately tied to distinct forms of vapor-liquid aggregation in the microgap channel, insufficient attention has been paid to the formation of distinct wave patterns and instabilities on the thin liquid films associated with high-quality microgap channel flow. This paper focuses on the results of visualization and heat transfer studies of such two-phase flows, under both adiabatic and diabatic conditions, for FC-72 flowing in a 184 micron microgap channel at a mass flux of 230 kg/m²-s. The study has revealed the existence of a post-annular, high-quality Rivulet flow regime, in which the liquid film breakdown and local wall dryout drives large surface anisothermals and limits the heat transfer rate from the wall.

As predicted by the prevailing flow regime models, annular flow is found to be the dominant flow regime for this microgap configuration. For the adiabatic conditions, flow qualities ranged between 27% and 81%, and widely spaced, 3D waves, with a wavelength that decreases with increasing flow quality, were observed on the liquid-vapor interface. For the diabatic condition, the inlet flow quality was maintained at 36% and the exit flow quality varied between 47% and 97%. For exit qualities greater than 61%, the liquid film would periodically rupture into rivulet of varying width and length. The spacing, length, and width of the rivulets varied considerably, and can easily stretch well into dryout region. The axial variation of the wall heat transfer coefficient was found to reflect and confirm the expected axial propagation of the two-phase flow regimes and the onset of local dryout associated with the newly-defined Rivulet regime.

1. Introduction
The increased integration density of electronic components and subsystems, including the nascent commercialization of 3D chip stack technology, has exacerbated the thermal management challenges facing electronic system developers. The confluence of chip power dissipation above 100 W, localized hot spots with fluxes above 1 kW/cm², and package-level volumetric heat generation that can exceed 1 kW/cm³ has exposed the limitations of the current "remote cooling" paradigm and its inability to facilitate continued enhancements in the performance of advanced silicon and compound semiconductor components. These thermal limitations have compromised the decades-long Moore’s law progression in microprocessor performance and threaten to derail the innovation engine which has been responsible for much of the microelectronic revolution. In conventional cooling architectures for electronics, reliance on thermal conduction and spreading, in the commonly-used chips and substrates and across the multiple material interfaces present in packages and modules, severely constrains

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the ability of remotely located heat rejection surfaces to reduce the temperature rise of critical on-chip hot spots and individual chips in a module or in a stack. Moreover, continued application of this “remote cooling” paradigm, has resulted in electronic systems in which the thermal management hardware accounts for a large fraction of the volume, weight, and cost of advanced electronic systems and undermines efforts to transfer emerging electronic components to portable, as well as other small form-factor, applications. Despite ongoing evolutionary improvements in commercial thermal packaging technology [1], the sequential conductive thermal resistances, inherent to the “remote cooling” paradigm, have resulted in only limited improvements in the overall junction-to-ambient thermal resistance of high-performance electronic systems during the past decade. Consequently, many commercial and military electronic systems are thermally-limited, performing well below the inherent electrical capability of the device technology they exploit. To overcome these limitations and remove a significant barrier to continued Moore’s law progression in electronic components and systems, it is essential to “embed” aggressive thermal management in the chip, substrate, and/or package and directly cool the heat generation sites. The development of such “Gen-3” thermal management technology, following on the Gen-1 air-conditioning approaches of the early years and the decades-long commitment to the Gen-2 “remote cooling” paradigm, has been spearheaded by DARPA, through its Intra/Inter Chip Embedded Cooling (ICECool) thermal packaging program, and embraced by many organizations [2–3]. Detailed consideration of power distribution in the chip and package, and how power delivery needs and constraints are integrated into the design, fabrication, and optimization of advanced electronic and photonic components, can inform and guide the articulation of the Gen-3 embedded cooling paradigm [4]. Thus, it is expected that embedded cooling will involve the creation of a rich micro/nano grid of thermal interconnects, using high thermal conductivity, as well as thermoelectric, materials to link on-chip hot spots to convectively and evaporatively cooled microchannels. Such intra/inter chip enhanced cooling approaches will need to be compatible with the materials, fabrication procedures, and thermal management needs of emerging homogeneous and heterogeneous integration in 3D chip stacks, 2.5D constructs, and planar arrays, as shown conceptually in figure 2. An intrachip approach would involve fabricating micropores and microchannels directly into the chip [5–6], while an interchip approach would involve utilizing the microgap between chips in three-dimensional stacks [7], as the cooling channel. In addition to the inclusion of an appropriate grid of passive and/or active thermal interconnects, it is expected that a combination of intrachip and interchip approaches, linked with thru-silicon and/or “blind” micropores will confer added thermal management functionality. These microchannels and/or micropores will need to be integrated into a fluid distribution network, delivering chilled fluid to the chip or package and extracting a mixture of heated liquid and vapor to be transported to the ambiently cooled radiator. Moreover, on-chip microvalves, thermostatically or digitally controlled, will be needed to regulate the flow of the coolant and assure the most efficient use of its latent and sensible cooling capacity. Successful development and implementation of this Gen-3 thermal packaging paradigm would place thermal management on an equal footing with functional design and power delivery, transforming electronic system architecture and unleashing the power of nanofeatured device technology, while overcoming the SWaP bottleneck encountered by many advanced electronic systems.

The forced flow of fluid, undergoing vigorous phase change, in a microgap channel is a promising approach for inter-chip embedded cooling of high-powered electronics [8-10], offering high heat transfer coefficients, hot spot mitigation, temperature uniformity, and minimal pumping power consumption. In addition, a microgap cooler can be integrated into electronic substrates, created between chip stacks, or formed directly over the die in a flip-chip package, providing concentrated cooling capability and eliminating the need for an attached microcooler. A microgap cooler can also serve as a fundamental building block for embedded microfluidic cooling designs. Broad application of such microgap coolers requires a fundamental understanding of the underlying hydrodynamic and thermal behavior of two-phase flow and evaporative heat transfer, especially in the thermally-favorable high-quality domain.

2. Two-Phase Flow in Microgap Channels
In two-phase flow, the extent, aggregation, and distribution of each phase is classified into distinct groups called flow regimes. The dominant regimes in two-phase microgap flow are bubbly, intermittent, and annular flow [8-14]. Bubbly flow is observed at the low qualities and consists of small, spherical bubbles, dispersed and transported by the continuous liquid phase. With increasing flow quality, the bubbles grow and coalesce until eventually confined by the lateral bounds of the channel and then continue to elongate axially. This distribution of large vapor “slugs”, separated by liquid plugs or bridges, is known as intermittent flow. As the vapor fraction continues to increase, the larger vapor bubbles coalesce into a continuous annular configuration, consisting of a high-velocity vapor core, and thin, shear-driven, liquid film around the perimeter of the channel. These flow regimes can be clearly seen in the photographic images of figure 1 [12] for the two-phase flow of R134a in a 0.79 mm diameter channel.

![Photograph of flow regimes](image)

**Figure 1.** Prevailing flow regimes in micro-scale channels, R134a, D = 0.79 mm, m = 500 kg/m²-s: (a) bubbly flow at x = 0.03, (b) intermittent flow at x = 0.11, and (c) annular flow at x = 0.73 [12].

Recent work [8-9,11,15-19] has uncovered a strong dependence of the observed M-shape variation in heat transfer coefficient in a microgap channel with L/D ratios between 100 and 500, displayed in figure 2, on the prevailing two-phase flow regimes. The low-quality peak in the M-shape heat transfer coefficient profile corresponds to the incipience of fully developed nucleate boiling or bubbly flow. The bubble nucleation, movement, and acceleration of liquid disrupt and thin the thermal boundary layer, all of which enhance the heat transfer coefficient. With the additional increase in quality, the flow quickly progresses from bubbly to intermittent flow. In intermittent flow, the thermal transport enhancement attributed to bubbly flow is gradually suppressed by the confinement of large vapor bubbles and decreasing liquid plug length, resulting in an overall deterioration in the heat transfer coefficient with increasing flow quality. However, this deterioration is soon overcome by an inflection and monotonic increase in the M-shape heat transfer coefficient profile, corresponding to the transition from intermittent to annular flow and driven by the increasing vapor-liquid velocity difference and thinning of the evaporating liquid film. The heat transfer coefficient continues to increase until reaching a second maximum and then is followed by a deterioration that asymptotically approaches the value for single-phase vapor convection. The mechanisms responsible for this high-quality peak and deterioration in the heat transfer coefficient are poorly understood and have constrained the design, optimization, and implementation of two-phase coolers in the favorable high-quality domain.
Figure 2. Characteristic M-shape variation in the heat transfer coefficient for refrigerant flow in microgap channels. Data from [16-17, 19].

In an effort to investigate the mechanisms responsible for the thermal deterioration observed in high-quality annular flow, Kabov et al. [20] empirically simulated an adiabatic shear-driven liquid film in a 40 mm x 80 mm x 2 mm high channel by independently injecting FC-72 liquid and nitrogen vapor streams. Deformations and emerging patterns were observed at the liquid-vapor interface and, depending on the flow rate of each phase, were classified into five sub-regimes: (1) cells, (2) structures, (3) 2D waves, (4) 3D waves, and (5) film rupture. The sub-regime map of the interfacial deformations and a photographic example of the 2D and 3D wave patterns are shown in figure 3. At low liquid and vapor velocities, the deformations in the liquid film are weak and classified as cells and structures. As the vapor velocity increases, periodic 2D waves emerge amidst the smooth film; the frequency of the 2D waves increases with vapor velocity until eventually evolving into a 3D wave structure. For low liquid film flow rates and sufficiently large vapor velocities, the liquid film ruptures and is torn from surface. The observed deformations and rupture of the shear-driven liquid film may be responsible for the high-quality thermal deterioration in diabatic microgap channels. The present study provides a detailed exploration of the behavior of the liquid-vapor interface for high-quality, single-component (FC-72) annular flow in a microgap channel, under both adiabatic and diabatic condition. For the adiabatic cases, no additional heat is transferred within the channel itself, and the liquid and vapor is injected into the channel as a single, mixed stream, in which the vapor fraction is controlled with an upstream evaporator. For the diabatic case, the inlet vapor fraction of the injected stream is kept constant and heat is uniformly applied along the entire length of the microgap surface.
Figure 3. Sub-regime map and visualization of interfacial deformations for two-phase flow of FC-72 and nitrogen in a mini-channel, where: (1) cells, (2) structures, (3) 2D waves, (4) 3D waves, and (5) film rupture [20].

3. Experimental Apparatus and Procedure

3.1. Flow-Loop Apparatus

A schematic of the flow loop used for the present study is shown in figure 4. The dielectric coolant used is 3M’s FC-72, with an atmospheric saturation temperature of 56˚C. The flow loop consists of a Fluid-o-Tech magnetically coupled gear pump with a DC motor, Kobold DPM pelton wheel flow sensor, Omega inline resistive coil heater (evaporator), Gems 2200 pressure transducer, Setra 230 differential pressure transducer, Flat Plate liquid-to-liquid brazed plate heat exchanger (condenser), Omega T-type thermocouple probes, and Swagelok stainless steel tubing and fittings. Electrical power is supplied to the evaporator and microgap ceramic resistive heaters with a BK Precision XLN10014 DC power supply and data is acquired using a National Instruments CompactDAQ chassis with a NI 9214 thermocouple, NI 9205 analog voltage, and NI 9203 analog current module. A Nikon D2X SLR camera, configured with a Nikkor PC micro 85mm lens and Speedlight flash was used for the photography.
3.2. Microgap Test Section
A top-down, cross-sectional, and isometric view of the copper microgap test section is shown in figure 5. The microgap channel is 35 mm long, 20 mm wide, and has a gap height of 184 μm. The wall temperature is measured with 14 thermocouples, located 1 mm below the axial centerline of the microgap surface. The holes for the thermocouples are 1 mm in diameter and drilled in a staggered configuration, as shown in figure 5(a). For the diabatic experiments, Joule heating is applied to the bottom of the microgap channel with two ceramic resistors that are directly soldered to the copper surface. The upper portion of the channel is confined with a polycarbonate cover and is hermetically sealed using an o-ring. The polycarbonate cover is optically transparent, providing a top-down view of the microgap flow.

The surface roughness of the milled & polished copper microgap surface was measured using white light interferometry with a Veeco Wyko NT1100 optical profiling system. A representative example of the microgap’s surface topography is shown in figure 6. The RMS surface roughness varies from 200 to 400 nm and the copper surface is coated with a natural layer of oxide.

Figure 5. Microgap test section: (a) top-down, (b) cross sectional, (c) and isometric view.
3.3. Experimental Procedure
The flow rate for the adiabatic and diabatic micro-gap cases is maintained at 0.5 mL/s, corresponding to a mass flux of 230 kg/m²-s. For the adiabatic cases, the flow quality injected into the microgap is finely controlled with the inline heater (evaporator) and ranges between 27 and 81%. For the diabatic cases, the microgap inlet flow quality is maintained at 36% and the exit quality varies between 36 and 99%, from the addition of heat supplied by the ceramic resistive heaters. Data and visualization is acquired once a quasi-steady state condition has been reached for a given set of conditions. It is to be noted that the interfacial wave structures were found to display considerable spatial and temporal variation. Consequently, the images shown should be understood to represent an average or typical conditions.

3.4 Data Reduction
The flow quality for the adiabatic cases, and inlet flow quality for the diabatic cases, is calculated using the following energy balance:

\[ x = \frac{\eta q - M_j c_{p,j} \left[ T_{in}(P) - T_{res,p} \right]}{M_j h_{lg}} \]  \hspace{1cm} (1)

The inline heater adds the heat, \( q \), where the conversion efficiency, \( \eta \), is nearly unity (97.4%) given that the heater is extensively insulated. Before the single-phase FC-72 liquid enters the inline heater, the initial temperature, \( T_{res,p} \), and mass flow rate, \( M_j \), are measured. The FC-72 fluid properties were obtained from 3M.

Similarly, the exit flow quality for the diabatic cases is calculated from the following energy balance:

\[ x = \frac{\eta q}{M_j h_{lg}} \]  \hspace{1cm} (2)

The heat loss, \( q_{loss} \), and conversion efficiency, \( \eta \), is calculated from equations (3) and (4), respectively. The heat loss was found experimentally and varies linearly with the average temperature difference of the channel.

\[ q_{loss} = 0.052 \left( T_{wall,avg} - T_e \right) \]  \hspace{1cm} (3)

\[ \eta = \frac{q - q_{loss}}{q} \]  \hspace{1cm} (4)
The superficial liquid and vapor velocity is calculated from equations (5) and (6), respectively:

$$u_{sl} = \frac{(1-x)\dot{m}}{\rho_l}$$  \hspace{1cm} (5)

$$u_{sg} = \frac{x\dot{m}}{\rho_g}$$  \hspace{1cm} (6)

The spatially- and temporally-averaged heat transfer coefficient for the heated wall of the microgap channel is calculated as follows:

$$h_{avg} = \frac{\eta q}{A(T_w - T_{sat}(P_{avg}))}$$  \hspace{1cm} (7)

where $\eta$ is the conversion efficiency (fraction of the resistor power that enters the channel as heat), $q$ the power input, $A$ the surface area of the microgap’s surface, $T_w$ the average wall temperature (as measured from the thermocouples), and $T_{sat}$ the average saturation temperature of the FC-72 for the average pressure, $P_{avg}$, of the microgap. The FC-72 fluid properties were obtained from 3M.

A summary of the RSS uncertainty in the experimental parameters is given in Table 1.

| Experimental Parameter          | Uncertainty | Uncertainty (%) |
|---------------------------------|-------------|-----------------|
| Temperature, $T$ (K)            | ±0.60       | 1.0–0.81        |
| Heat Input, $q$ (W)             | ±0.060–0.18 | 0.30            |
| Heat Flux, $q’$ (W/cm²)         | ±0.0075–0.023 | 1.8–0.13        |
| Flow Rate, $\dot{V}$ (m³/s)     | ±0.030 x 10⁴ | 6.0             |
| Channel Inlet Pressure, $p$ (kPa)| ±0.50       | 3.9–6.5         |
| Superficial Liquid Velocity, $u_{sl}$ (m/s) | ±0.0062–0.0074 | 7.9–25        |
| Superficial Vapor Velocity, $u_{sg}$ (m/s) | ±0.45–1.2 | 9.0–9.9         |
| Mass Flux, $\dot{m}$ (kg/m²-s) | ±15         | 6.5             |
| Heat Transfer Coefficient, $h$ (W/m²-K) | ±100–500 | 2.0–11         |

4. Visualization and Discussion

4.1. Adiabatic Visualization

The adiabatic photographic visualization for the 184 μm microgap channel, with a mass flux of 230 kg/m²-s, is shown in figure 7, for a flow quality of 27, 41, 54, 66, and 78%, respectively, and described in detail in [18, 20]. At the lowest quality of 27%, figure 7(a), the flow is in the intermittent regime. The flow consists of randomly dispersed, large two-dimensional vapor bubbles with smooth, continuous liquid-vapor interfaces. With increasing streamwise distance, the vapor bubbles—on average—grow in size, from both evaporation and coalescence. With an increase in quality from 27 to 41%, depicted in figure 7(b), the vapor bubbles are noticeably larger; as shown in the downstream portion of figure 7(b), and 3D waves emerge at the liquid-vapor interface as the flow locally transitions into the annular regime. At a quality of 54%, as seen in figure 7(c), the entire channel has transitioned into the annular regime; the flow pattern now consisting of a thin liquid film, fully wetting the rectangular perimeter, and a high velocity vapor core. The magnified portion of figure 7(c) depicts the observed 3D waves at the liquid-vapor interface, with a streamwise wavelength that varies spatially. As the quality increases to 66%, shown in figure 7(d), the amplitude of the 3D waves decreases on the now
thinner liquid film. In figure 7(e), showing the highest observed quality of 78%, the liquid-vapor interface consists of wide, continuous waves, with very small amplitudes; similar in appearance to Tollmien-Schlichting waves.

The superficial velocities from the adiabatic cases are plotted on a Taitel & Dukler flow regime map in figure 8, with a modified Ullman-Brauner intermittent-to-annular and Rahim et al [12] bubbly-to-intermittent flow regime boundary, as given by equations (8) and (9), respectively:

\[ u_{sl} = \frac{1 - \varepsilon_{crit}}{\varepsilon_{crit}} u_{sg} \], where \( \varepsilon_{crit} = 0.67 \) \hspace{1cm} (8)

\[ u_{sg} = \frac{6.2 \left[ \sigma g (\rho_g - \rho_l) \right]^{1/4}}{\rho_l^{1/2}} \] or \[ \frac{We}{Bo^{1/4}} = 6.2 \] \hspace{1cm} (9)

The agreement between the modified Taitel & Dukler flow regime map and the visualization presented in figure 7 is remarkably good. At the lowest quality of 27%, corresponding to a superficial vapor and liquid velocity of 4.4 and 0.10 m/s, the flow regime is both classified and visually confirmed to be in the intermittent regime. As the quality increases to 41%, corresponding to a superficial vapor and liquid velocity of 6.6 and 0.082 m/s, it straddles the intermittent-to-annular boundary, supported by the visualization in figure 7(b)–a upstream and downstream distribution of intermittent and annular flow, respectively. At a flow quality of 54%, corresponding to a superficial vapor and liquid velocity of 8.6 and 0.058 m/s, both the map and visualization confirm the onset and, for a flow quality of 66 and 78%, continuation of annular flow. It is, nevertheless, significant that rather than the classically depicted smooth liquid film, several sub-regimes of interfacial wave structures are observed to characterize the adiabatic annular film interface, as suggested in [16].

(a) \( x = 27\% \), \( u_{sg} = 4.4 \) m/s, \( u_{sl} = 0.10 \) m/s

(b) \( x = 41\% \), \( u_{sg} = 6.6 \) m/s, \( u_{sl} = 0.082 \) m/s

(c) \( x = 54\% \), \( u_{sg} = 8.6 \) m/s, \( u_{sl} = 0.058 \) m/s
Figure 7. Experimental adiabatic microgap visualization for FC-72, $\dot{m} = 230$ kg/m$^2$-s, and $H = 184$ μm: (a) intermittent flow, $x = 27\%$, (b) intermittent flow, $x = 41\%$, (c) annular flow with 3D waves, $x = 54\%$, (d) annular flow with 3D waves, $x = 66\%$, (e) annular flow with Tollmien-Schlichting wave patterns, $x = 78\%$. Flow is from left to right. Adapted from [21-22].
Figure 8. Taitel-Dukler flow regime map with a modified Ullman-Brauner bubbly-to-intermittent and intermittent-to-annular boundary for a microgap channel, $H = 184 \, \mu m$, and FC-72. Experimental data from Fig. 7, $\dot{m} = 230 \, kg/m^2\cdot s$, and $x = 27, 41, 57, 69, \text{ and } 81\%$.

A linear stability model, described in full detail in [21], was used to numerically predict the interfacial wavelengths and compare them to the measured wavelengths for a mass flux of $230 \, kg/m^2\cdot s$. Despite the complex nature of the flow, and 3D features not fully accounted for in the 2D stability model, as well as the considerable uncertainty in the void fraction predictions, figure 9 shows that the distribution of wavelengths predicted by the linear stability analysis appear to agree reasonably well with the range of wavelengths observed in the experiments. The most unstable wavelengths predicted by the linear stability analysis has an overall error of only 3.9% for the three cases shown here.
Figure 9. Comparison of experimental (black) and the numerically predicted (gray) probability distribution of wavelengths for a mass flux of 220 kg/m²-s: (a) $x = 54\%$, (b) $x = 66\%$, (c) $x = 78\%$.

4.2. Diabatic Annular Microgap Flow – Wave Patterns and Rivulets

The diabatic photographic visualization for the 184 μm microgap channel, with a mass flux of 230 kg/m²-s, and inlet quality of 36%, taken from [16], for an exit quality of 47, 61, 76, 83, 90, and 97%, respectively, and reproduced in figure 10 of this paper. For an inlet flow quality of 36% and exit quality of 47%, figure 10(a), the flow regime distribution is representative of what was earlier observed in figure 7(c). The channel mostly consists of large amplitude 3D waves with a wide range of interfacial wavelengths; both the physical appearance and frequency of 3D interfacial waves are consistent with those observed for the adiabatic visualization. The uniform distribution of 3D waves persists for an exit quality up to 61% but with a notable increase in wave frequency, particularly near the channel exit where the vapor velocity is highest. However, for an exit quality of 76%, the downstream portion of the lower liquid film ruptures, creating liquid fingers or rivulets and adjacent dryout regions; the dryout boundary (lower liquid film front) is marked with a dotted line in figure 10(c). Dryout of the lower wall continues to migrate upstream, with increasing wall heat flux and the associated exit quality, as shown in figure 10(e)–(f), for an exit quality of 90 and 97% respectively.

It may, thus, be argued that a previously unrecognized, post-annular Rivulet flow regime is characteristic of high-quality two-phase flow in microgap channels and that these liquid rivulets are encountered along the dryout boundary, separating the wetted and dry regions of the heated wall, as depicted in figure 10(c), 10(d), 10(e) and
10(f). The spacing, length, and width of the rivulets vary considerably, and they can easily extend well into the dryout region.

Regrettably, the traditional two-phase flow regime maps, including the Taitel and Dukler map discussed in section 4.1, do not include this post-annular Rivulet regime. However, Lee and Lee [23] did propose a criteria for stable, shear-driven Rivulet, or ribbon, flow in a circular tube. The boundary was empirically formulated using flow pattern data for air-water and air-methanol flow in Teflon, polyurethane, and polyethylene tubes. The proposed annular-to-rivulet boundary is given by equation (10) below and occurs at a constant superficial liquid velocity:

\[ U_{sl - Rivulets} = (a + b\theta) \]  

(10)

where \( \theta \) is the contact angle and the coefficients \( a \) and \( b \) are provided for three different cases:

- High wetting case (\( \theta < 50^\circ \)): \( a = 0.00185, \ b = 0.0309 \)  
  (11)
- Marginal wetting case (\( 50^\circ < \theta < 90^\circ \)): \( a = -0.254, \ b = 0.324 \)  
  (12)
- Poor wetting case (\( \theta > 90^\circ \)): \( a = 0.123, \ b = 0.0838 \)  
  (13)

The locus of the Rivulet flow regime boundary, as proposed by Lee and Lee [23], has been added to the modified Taitel and Dukler flow regime map shown in figure 12 for an assumed contact angle of 20 degrees. It may be seen that all of the diabatic cases are in the annular regime and are accurately classified by the flow regime map; most interestingly, the rivulet boundary is in surprisingly good agreement with the data points associated with the appearance of rivulets.

5. Heat Transfer Coefficient and Rivulet Regime

A plot of the average heat transfer coefficient versus exit flow quality is provided in figure 11 and replotted in terms of superficial vapor velocity above the flow regime map in figure 12. As shown, the heat transfer coefficient rises with increasing flow quality within annular flow, from a value of 3500 to 4900 W/m²-K for an exit flow quality of 47 to 83%, respectively. In annular flow, as the evaporating liquid-vapor interface is brought closer to the heated surface (i.e., the thinning of the liquid film with increasing flow quality and void fraction) the thermal resistance to heat transfer through the FC-72 liquid film is reduced and, accordingly, the heat transfer coefficient increases. At an exit flow quality of 76%, the evaporating liquid film begins to breakdown into rivulet structures, but the heat transfer coefficient continues to rise and peak at a value of 5200 W/m²-K for an exit flow quality of 76 and 83%, despite the intermittent breakdown in the liquid film. It appears that the enhancement from very thin film evaporation initially overcomes the deterioration caused by the formation of rivulets on the heated surface for exit flow qualities from 76 to 90%. It’s not until an exit flow quality of 97% that the heat transfer coefficient begins to fall, to a value of 4400 W/m²-K, and the breakdown in the liquid film becomes more widespread both spatially and temporally.

Plotted alongside the experimental data in figure 11 is the venerable Chen correlation. The venerable Chen correlation [24] asserts that the saturated two-phase heat transfer coefficient is a sum of microscopic (bubble nucleation) and macroscopic (bulk convection) mechanisms and makes no formal allowance for the onset of local dryout on the heated surface. As described in [24], the macroscopic contribution is predicted using the turbulent Dittus-Boelter convective heat transfer coefficient equation and the microscopic contribution is predicted using the Forster-Zuber pool boiling correlation. The microscopic contribution is multiplied by a “suppression factor”—a function of the two-phase Reynolds number and Martinelli parameter—to account for the suppression of nucleate boiling at higher flow qualities where wall superheats decrease and convective mechanisms dominate. The Chen correlation was specifically derived for macro-sized tubes with upward flow of water, methanol, cyclohexane, n-pentane, n-heptane, and benzene, for qualities ranging from 0 to 71%, heat fluxes up to 240 W/cm², and system pressures from 0.55 to 35 bar, yielding an average accuracy of +/-12% for these original datasets [24] and providing agreement to within typically +/-40% for a wide range of data [25].
The Chen correlation has been found in prior studies to provide strong agreement with the heat transfer coefficients for microgap channels operating in annular flow [8–9,11,13]. As may be seen in figure 11, the agreement between the measured and Chen-predicted average heat transfer coefficients in the two-phase microgap channel studied herein is excellent, with an overall percent error of only 3%. The dominance of annular flow in the present microgap channel may help explain this excellent agreement.

(a) \( x_{\text{in}} = 36\%, \quad x_{\text{exit}} = 47\%, \quad q'' = 1.25 \text{ W/cm}^2, \quad G = 220 \text{ kg/m}^2\cdot\text{s}, \quad T_{\text{sat, in}} = 59^\circ\text{C}, \quad T_{\text{wall, avg}} = 61^\circ\text{C}, \quad h_{\text{avg}} = 3500 \text{ W/m}^2\cdot\text{K} \)

(b) \( x_{\text{in}} = 36\%, \quad x_{\text{exit}} = 61\%, \quad q'' = 2.50 \text{ W/cm}^2, \quad G = 220 \text{ kg/m}^2\cdot\text{s}, \quad T_{\text{sat, in}} = 59^\circ\text{C}, \quad T_{\text{wall, avg}} = 63^\circ\text{C}, \quad h_{\text{avg}} = 4500 \text{ W/m}^2\cdot\text{K} \)

(c) \( x_{\text{in}} = 36\%, \quad x_{\text{exit}} = 76\%, \quad q'' = 3.50 \text{ W/cm}^2, \quad G = 220 \text{ kg/m}^2\cdot\text{s}, \quad T_{\text{sat, in}} = 59^\circ\text{C}, \quad T_{\text{wall, avg}} = 65^\circ\text{C}, \quad h_{\text{avg}} = 4900 \text{ W/m}^2\cdot\text{K} \)

(d) \( x_{\text{in}} = 36\%, \quad x_{\text{exit}} = 83\%, \quad q'' = 4.10 \text{ W/cm}^2, \quad G = 220 \text{ kg/m}^2\cdot\text{s}, \quad T_{\text{sat, in}} = 59^\circ\text{C}, \quad T_{\text{wall, avg}} = 66^\circ\text{C}, \quad h_{\text{avg}} = 5200 \text{ W/m}^2\cdot\text{K} \)

(e) \( x_{\text{in}} = 36\%, \quad x_{\text{exit}} = 90\%, \quad q'' = 5.00 \text{ W/cm}^2, \quad G = 220 \text{ kg/m}^2\cdot\text{s}, \quad T_{\text{sat, in}} = 59^\circ\text{C}, \quad T_{\text{wall, avg}} = 67^\circ\text{C}, \quad h_{\text{avg}} = 5200 \text{ W/m}^2\cdot\text{K} \)
Figure 10. Experimental diabatic microgap visualization for FC-72, $\dot{m} = 230$ kg/m²-s, $T_{sat,in} = 59^\circ$C, $x_{inlet} = 36\%$, and $H = 184$ μm: (a) annular flow, $x_{exit} = 47\%$, (b) annular flow, $x_{exit} = 61\%$, (c) annular flow with downstream dryout, $x_{exit} = 76\%$, (d) annular-dryout with magnified liquid film remnants, $x_{exit} = 83\%$, (e) annular-dryout with magnified liquid film rivulets, $x_{exit} = 90\%$, (f) annular-dryout with single long rivulet, $x_{exit} = 97\%$. Flow is from left to right. Adapted from [22].

Figure 11. Average heat transfer coefficient vs. exit flow quality profile for a microgap channel, $H = 184$ μm and FC-72. Experimental data from figure 10, $\dot{m} = 230$ kg/m²-s, $q^\prime\prime = 1.25$–5.63 W/cm², $x_{exit} = 47$–97%, and $x_{in} = 36\%$. 
4. Conclusions
Two-phase flow in microgap channels offers highly potent thermal management capability and is a promising solution for the emerging "embedded cooling" paradigm of electronic cooling. While the thermofluid characteristics and operational limits of such microcoolers are intimately tied to distinct forms of vapor-liquid aggregation in the microgap channel, insufficient attention has been paid to the formation of distinct wave patterns and instabilities on the thin liquid films associated with high-quality microgap channel flow. The purpose of the present study was to experimentally visualize the annular flow of FC-72, at a mass flux of 230 kg/m²-s, in a 184 μm microgap channel for an adiabatic and diabatic condition. The adiabatic visualization and modified Taitel & Dukler flow regime map confirmed the expected existence of intermittent flow at a flow quality of 27% and the onset and continued dominance of annular flow for qualities above 41%. In annular flow, periodic 3D wave patterns appear along the shear-driven liquid film interface, with a wavelength and amplitude that decreases with increasing flow quality. A linear stability model from a previous work is used to numerically predict the interfacial wavelengths with an average error of only 3.9%.

The study has also revealed the existence of a post-annular, high-quality Rivulet flow regime. For exit qualities greater than 61%, the liquid film would periodically rupture into rivulets of varying width and length, yet the average heat transfer coefficient continued to rise. However, rivulet flow became more frequent and widespread as the exit quality and heat flux increased, causing an eventual downward deterioration in the heat transfer coefficient at exit qualities above 90%. The spacing, length, and width of the rivulets varied considerably, and can easily stretch well into dryout region. The axial variation of the wall heat transfer coefficient was found to
reflect and confirm the expected axial propagation of the two-phase flow regimes and the onset of local dryout associated with the newly-defined rivulet regime. The heat transfer coefficient profile was well predicted by the Chen correlation, with a percent error of only 3%, and the transition of annular to rivulet flow was well predicted by the rivulet flow regime boundary proposed by Lee and Lee.

5. References
[1] Bar-Cohen A and Bloschok K 2012, Advanced Thermal Management Technologies for Defense Electronics Proceedings of Security and Sensing Conference, Baltimore, MD.
[2] Bar-Cohen A 2013 Thermal Packaging: From Problem Solver to Performance Multiplier Electronic Cooling Magazine pp 8–11
[3] Bar-Cohen A 2013 Gen 3 Thermal Management Technology: Role of Microchannels and Nanostructures in an Embedded Cooling Paradigm ASME Journal of Nanotechnology in Engineering and Medicine vol 4 020907-1 to 020907-3
[4] Bar-Cohen A and Geisler K 2011 Cooling the Electronic Brain Mech. Eng. vol 133(4) pp 38–41
[5] Tuckerman D and Pease R 1981 High-Performance Heat Sinking for VLSI IEEE Electron Device Lett. vol 2(5) pp 126–129
[6] Yarin L, Mosyak A, and Hetsroni G. 2009 Fluid Flow, Heat Transfer, and Boiling in Micro-Channels Springer, Berlin
[7] Bar-Cohen A, Sheehan J, and Rahim E 2012 Two-Phase Thermal Transport in Microgap Channels-Theory Experimental Results, and Predictive Relations Microgravity Sci. Technol. vol 24 pp 1–15
[8] Bar-Cohen A, Rahim E, and Ali I 2012 Two-Phase Microgap Cooling of a Thermally-Simulated Microprocessor Chip Proc. of 13th IThERM Conf. pp. 1090-1105
[9] Kim D, Rahim E, and Bar-Cohen A 2010 Direct Submount Cooling of High-Power LEDs IEEE Trans. Comp. vol 33(4) pp. 698-712
[10] Alam T, Lee P, Yap C, and Jin L 2012 Experimental investigation of local flow boiling heat transfer and pressure drop characteristics in microchannel Int. J. of Multiphase Flow vol 42, pp 164-174
[11] Bar-Cohen A and Rahim E, 2009 Modeling and prediction of two-phase microgap channel heat transfer characteristics Heat Transf. Eng. vol 30(8) pp 601-625
[12] Rahim E, Revellin R, Thome J, and Bar-Cohen A 2011 Characterization and prediction of two-phase flow regimes in miniature tubes Int. J. of Multiphase Flow vol 37(1) pp 12-23
[13] Kim D, Rahim E, Bar-Cohen A, and Han B 2008 Thermofluid Characteristics of Two-Phase Flow in Micro-Gap Channels Proc. of 11th IThERM Conf. pp 979-992
[14] Thome J, Bar-Cohen A, Revellin R, and Zun I 2013 Unified Mechanistic Multiscale Mapping of Two-Phase Flow Patterns in Microchannels Exp. Thermal and Fluid Science vol 44 pp 1-22
[15] Bar-Cohen A and Geisler K 2011 Cooling the Electronics Brain Mechanical Engineering Magazine https://www.asme.org/about-asme/news-media/newsletters
[16] Yen T, Kasagi N, and Suzuki Y 2003 Forced convective boiling heat transfer in microtubes at low mass and heat fluxes Int. J. of Multiphase Flow vol 29(12) pp 1771-1792
[17] Mastrullo R, Mauro A, Thome J, and Vanoli G 2012 CO2 and R410A: Two-phase flow visualization and flow boiling measurements at medium (0.50) reduced pressure Applied Thermal Eng. vol 49 pp 2-8
[18] Cortina-Diaz M and Schmidt J 2006 Flow boiling heat transfer of n-hexane and n-octane in a minichannel Proc. of 13th Int. Heat Trans. Conf.
[19] Yang Y and Fujita Y 2004 Flow boiling heat transfer and flow pattern in rectangular channel of mini-gap Proc. of 2nd Int. Conf. on Microchannels and Minichannels ICMM2004-2383
[20] Kabov O, Zaitsev D, Cheverda V, and Bar-Cohen A. 2011 Evaporation and flow dynamics of thin, shear-driven liquid film in microgap channels Exp. Thermal and Fluid Sciences vol 35(5) pp 825-831
[21] Bar-Cohen A, Holloway C, Kaffel K, and Riaz A 2016 Waves and instabilities in high quality adiabatic flow in microgap channels Int. J. of Multiphase Flow vol 83 pp 62-76
[22] Holloway C and Bar-Cohen A 2014 Liquid Film Wave Patterns and Dryout in Microgap Channel Annular Flow Proc. of 15th Int. Heat Trans. Conf.
[23] Lee C and Lee S 2008 Influence of surface wettability on transition of two-phase flow pattern in round
mini-channels *Int. J. of Multiphase Flow* pp 1474-1479

[24] Chen J 1966 Correlation for Boling Heat Transfer to Saturated Fluids in Convective Flow *I&EC Process Design and Development* vol 5(3) pp 322-329

[25] Kandlikar S 1990 A General Correlation for Saturated Two-Phase Flow Boiling Heat Transfer Inside Horizontal and Vertical Tubes *J. of Heat Transfer* vol 112 pp 219-228