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Flow regimes and convective heat transfer of refrigerant flow boiling in ultra-small clearance microgaps

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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 3000 kg/m^2 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 which were 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 MW/m^2 K in the vapor plume regime in the case of the finned microgap.

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 [1–8,13]. 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 (Dh $\geqslant$ 1 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 the liquid film wetting the microgap surface has a low thermal resistance [2,12,17]. As the microgap hydraulic diameter decreases, thin film boiling 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

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stratified flow [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. [17] 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 ~1.6 kW/cm^2, coolant mass fluxes up to 3000 kg/m^2 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 × 200 μm wide × 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. 1a). The RTDs are 80 μm × 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 32 × 20 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].

3. Experimental setup and 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.

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 1000 and 3000 kg/m^2 s 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 and 1090 kPa. Power was applied to the heaters in 0.1–0.25 W 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