Flow Boiling in Open Microchannels with Tapered Manifolds using Ethanol in a Gravity-Driven Flow

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A Thesis Submitted in Partial Fulfillment of the Requirements of the Degree of Master of Science in Mechanical Engineering

Thermal Analysis, Microfluidics, and Fuel Cell Lab
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ABSTRACT

Flow boiling in microchannel heat sinks has been researched extensively for its use in the cooling of high-power electronics. Previous works have proven open microchannels with tapered gap manifolds are effective in delivering enhanced flow boiling performance, with significant reductions in pressure drop. This work explores the feasibility of employing ethanol as a dielectric fluid in an open microchannel geometry with tapered manifolds, under a gravity-driven flow. A heat flux of 217 ± 13 W/cm² was dissipated with a pressure drop of only 8.8 ± 0.5 kPa. Parametric trends are presented regarding flow rate, taper, pressure drop characteristics, and their effect on critical heat flux, providing basic insight into designing high heat flux systems under a given gravitational head requirement. Based on the obtained results, design guidelines are developed for the manifold taper, ethanol flow rate, and imposed heat flux on the heat transfer coefficient and gravity head requirement for electronics cooling. Reducing flow instability and pressure drop, and enhancing heat transfer performance for a dielectric fluid will enable the development of pumpless cooling solutions in a variety of electronics cooling applications.

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# TABLE OF CONTENTS

List of Figures .................................................................................................................. 6
List of Tables .................................................................................................................... 9
Nonmencature .................................................................................................................. 10
1. Introduction .................................................................................................................. 11
2. Literature Review ....................................................................................................... 15
   2.1 Flow Instabilities and Pressure Drop ................................................................. 15
   2.2 Pressure Drop Elements and Artificial Nucleation Sites .................................... 18
   2.3 Microchannels with Increasing Cross-Sectional Area ........................................ 20
3. Experimental Setup .................................................................................................... 32
   3.1 Experimental Procedure ..................................................................................... 34
   3.2 Automated CHF Loop ......................................................................................... 35
   3.3 Data Acquisition System ..................................................................................... 36
   3.4 High-Speed Flow Visualization ........................................................................... 36
   3.5 Liquid Degassing ................................................................................................. 37
   3.6 Test Section .......................................................................................................... 39
      3.6.1 Manifold Blocks ......................................................................................... 40
      3.6.2 Heater Block and Microchannel Fabrication ............................................... 41
4. Heat Flux Calculation ................................................................................................. 42
   4.1 Heat Loss Study .................................................................................................... 43
5. Ethanol: Properties, Flammability & Precautions ..................................................... 44
6. Uncertainty Analysis ................................................................................................... 45
7. Results ......................................................................................................................... 48
   7.1 Plain Chip ............................................................................................................. 49
   7.2 Microchannel Chips ............................................................................................. 51
8. Discussion .................................................................................................................... 54
   8.1 Effect of Flow Rate .............................................................................................. 54
   8.2 Effect of Manifold Taper ...................................................................................... 55
   8.3 Comparison of the Plain and Microchannel Chips .............................................. 55
   8.4 Pressure Drop Performance ............................................................................... 56
   8.5 Flow Visualization ............................................................................................... 58
   8.6 Flow Instability & Pressure Drop Fluctuation ..................................................... 59
8.7 Pressure Drop Modelling ............................................................................................................. 62
  8.7.1 Single-Phase Inlet Region ........................................................................................................ 62
  8.7.2 Single-Phase Channel Region ................................................................................................. 63
  8.7.3 Two-Phase Region .................................................................................................................. 64
  8.7.4 Two-Phase Exit Region .......................................................................................................... 65
8.8 Parametric Study – Heat Transfer Performance as a Function of Available Fluid Head ........ 67
9. Conclusions ..................................................................................................................................... 71
10. Recommendations & Future Work ............................................................................................... 73
    10.1 Manifold Sealing Issues ........................................................................................................ 73
    10.2 Flow Instability ..................................................................................................................... 73
    10.3 Applications of Low Pressure Drop, Gravity-Driven Flow Boiling Systems ..................... 73
11. References ..................................................................................................................................... 75
Appendix ........................................................................................................................................... 77
    Appendix A: Elevated Reservoir Assembly ................................................................................. 77
    Appendix B: Error Propagation – Derivation of Uncertainties ....................................................... 79
      B.1 Heat Flux Uncertainty Derivation ...................................................................................... 79
      B.2 Chip Surface Temperature Uncertainty Derivation .............................................................. 80
      B.3 Heat Transfer Coefficient Uncertainty Derivation .............................................................. 82
      B.4 Exit Vapor Quality Uncertainty Derivation ....................................................................... 83
      B.5 Mass Flow Rate Uncertainty Derivation ............................................................................ 84
    Appendix C: Supplementary Figures ............................................................................................ 86
      C.1 Effect Of Flow Rate ............................................................................................................... 86
LIST OF FIGURES

Figure 1: Pressure drop oscillations with (a) fully open valve and (b) throttled valve [9] ....................... 16

Figure 2: Pressure drop fluctuations - Balasubramanian & Kandlikar [3]............................................ 16

Figure 3: Flow reversals in parallel microchannels. Balasubramanian & Kandlikar [3]......................... 17

Figure 4: Conceptual stepped and diverging microchannels – Mukherjee and Kandlikar [4].............. 18

Figure 5: Inlet restrictors and microchannels with reentrant cavities – Kuo and Peles [11].............. 19

Figure 6: Plots of (a) temperature as a function of heat flux and (b) two-phase heat transfer coefficient as a function of vapor quality - Kuo and Peles [11]................................................................. 20

Figure 7: Schematic of microchannels used by Lu and Pan [15]......................................................... 21

Figure 8: Plot of inlet temperature as a function of heat flux. Lu and Pan [15]................................. 21

Figure 9: (a) Schematic of model control volume; (b) Contour plot of attainable CHF values for different values of channel height and width expansion – Miner et al. [16].................................................. 22

Figure 10: Schematic of tapered microchannel manifold – Kandlikar et al. [5]................................. 23

Figure 11: (a) Heat transfer performance at different flow rates and (b) pressure drop performance of tapered manifold (TM) vs. uniform manifold (UM) – Kandlikar et al. [5]................................. 24

Figure 12: Stepped fin microchannels implemented by Balasubramanian et al. [17].......................... 24

Figure 13: (a) boiling curves, (b) heat transfer coefficient as a function of vapor quality. Balasubramanian et al. [17]........................................................................................................ 25

Figure 14: (a) Schematic of expanding channels, (b) Effect of expansion angle. Miner et al. [18]...... 26

Figure 15: Plots of (a) pressure drop and (b) wall superheat as functions of heat flux – Kalani and Kandlikar [6]........................................................................................................... 26

Figure 16: (a) CHF as a function of mass fraction, and (b) enhancement of heat transfer coefficient by surfactant as a function of heat flux; h’ = HTC with surfactant, h = HTC without surfactant. [21]................. 28

Figure 17: Heat Transfer Coefficient as a function of vapor quality - Diaz & Schmidt [7]............... 29

Figure 18: Heat fluxes of ethanol at different absolute pressures - Kalani & Kandlikar [24]............ 30
Figure 19: Experimental setup .................................................................................................. 33

Figure 20: Volume Fractions of Oxygen in Liquid (Data reproduced from Battino et al. [25]) ........... 38

Figure 21: Schematic of vapor-liquid separator vessel ................................................................. 39

Figure 22: Schematic of Tapered Manifold and Open Microchannels .............................................. 41

Figure 23: Schematic of the heater block and microchannel chip .................................................... 42

Figure 24: (a) Heat loss simulation model, (b) Plot of heat loss as a function of wall superheat ............ 43

Figure 25: Plots showing effect of manifold taper on plain chip at 60 mL/min: (a) boiling curve, (b) heat transfer coefficient as a function of vapor quality, (c) pressure drop as a function of heat flux .................. 50

Figure 26: Plots showing effect of manifold taper for microchannel chips at 60 mL/min: (a) boiling curve, (b) heat transfer coefficient as function of vapor quality, (c) pressure drop as function of heat flux ........................................................................................................... 52

Figure 27: Plots showing effect of flow rate on (a) plain and (b) microchannel chips with 6% manifold taper ........................................................................................................................................ 54

Figure 28: Plots showing effect of chip at 80 mL/min with 6% manifold taper: (a) boiling curve, (b) heat transfer coefficient as a function of vapor quality, (c) pressure drop as a function of heat flux .............. 57

Figure 29: High-speed video frames of growing and coalescing bubbles in chip MC1. Test conditions: 2% manifold taper, 60 mL/min flow rate, heat flux 63.7 W/cm². Frames: (a) 0.00 ms, (b) 3.50 ms, (c) 6.70 ms, (d) 8.00 ms, (e) 10.00 ms, (f) 12.70 ms .......................................................... 58

Figure 30: Pressure drop fluctuation – MC1 with 2% taper at a flow rate of 60 mL/min ..................... 60

Figure 31: ΔP – G curve for a channel with uniform heating .......................................................... 61

Figure 32: (a) Comparison of pressure drop model [27] and experimental data, and (b) Two-phase pressure drop components from model [27], for MC1 with 4% manifold taper at 80 mL/min .............. 66

Figure 33: Required ethanol head for a given heat flux, based on experimental data of all three chips with the 6% manifold taper, measured at (a) 40 mL/min, and (b) 60 mL/min ................................................. 68

Figure 34: Ethanol head as a function of heat flux with the 6% manifold taper, measured at 40 mL/min, 60 mL/min, and 80 mL/min, for the (a) plain chip, (b) MC1, and (c) MC2 ........................................................................... 69

Figure 35: 3D drawings of elevated reservoir assembly ................................................................. 77
Figure 36: Images of (a) elevated reservoir support plate, and (b) detail view of clamps and J-hook ..... 78

Figure 37: Effect of flow rate for plain chip with 2% manifold taper ............................................. 86

Figure 38: Effect of flow rate for plain chip with 4% manifold taper ............................................. 86

Figure 39: Effect of flow rate for plain chip with 6% manifold taper ............................................. 87

Figure 40: Effect of flow rate for chip MC1 with 2% manifold taper ............................................. 87

Figure 41: Effect of flow rate for chip MC1 with 4% manifold taper ............................................. 88

Figure 42: Effect of flow rate for chip MC1 with 6% manifold taper ............................................. 88

Figure 43: Effect of flow rate for chip MC2 with 2% manifold taper ............................................. 89

Figure 44: Effect of flow rate for chip MC2 with 4% manifold taper ............................................. 89

Figure 45: Effect of flow rate for chip MC2 with 6% manifold taper ............................................. 90
LIST OF TABLES

Table 1: CHF data of pure components, Fujita and Bai [20] ................................................................. 27

Table 2: Microchannel chip dimensions .................................................................................................. 40

Table 3: Microchannel chip dimensions .................................................................................................. 42

Table 4: Test Matrix - test cases highlighted in red with an asterisk* were not recorded ......................... 48

Table 5: Plain and microchannel chip performance at CHF and peak heat transfer coefficient values .... 53

Table 6: Trendline equations for ethanol head as a function of heat flux ............................................... 70
**NONMENCLATURE**

$q'' = \text{Heat flux}$

$k_{Cu} = \text{Thermal conductivity of copper}$

$T_s = \text{Chip surface temperature}$

$T_{sat} = \text{Saturation temperature}$

$T_1 = \text{Temperature of top heater block thermocouple}$

$T_2 = \text{Temperature of middle heater block thermocouple}$

$T_3 = \text{Temperature of bottom heater block thermocouple}$

$\Delta T_{sat} = \text{Wall superheat}$

$\Delta d = \text{Heater block thermocouple separation}$

$d_1 = \text{Distance between surface and top heater block thermocouple}$

$h = \text{Heat transfer coefficient}$

$x = \text{Exit vapor quality}$

$\Delta H = \text{Liquid head}$

$q''_{\text{adjusted}} = \text{Heat flux adjusted to account for heat loss across the edges of the chip}$

$q''_{\text{measured}} = \text{Heat flux measured from experimentation}$

$q''_{\text{loss}} = \text{Heat loss across the edges of the chip obtained from simulation}$

$p = \text{Function } p \text{ representing any given property}$

$U_p = \text{Uncertainty in function } p$

$\sigma_i = \text{Variables } \sigma \text{ of function } p$

$U_{\sigma_i} = \text{Uncertainty in variable } \sigma_i$
1. INTRODUCTION

Flow boiling in microchannels has been of great interest to the scientific community in the past decade, primarily due to the high heat fluxes that are made possible via the micro-scale geometries and high heat transfer coefficients associated with these systems. Cooling performance combined with compact size make microchannel flow boiling an excellent candidate for cooling of integrated circuit (IC) microchips, as well as compact heat exchangers, chemical reactors, and military applications. As the heat dissipation of components such as IC chips continues to increase, progressively higher cooling performance will be required to keep these components at their desired operating temperatures. Flow boiling allows for heat fluxes and heat transfer coefficients several orders of magnitude higher than those of single-phase liquid cooling systems. With the correct choice of fluid, with appropriate saturation properties, a chip dissipating heat fluxes on the order of 100 W/cm² could be cooled effectively by a flow boiling system, maintaining the heater surface temperature at a low degree of wall superheat.

While there is enormous potential for high cooling performance in flow boiling systems, there are several issues that have hindered their performance significantly. These include (i) flow instabilities, (ii) flow reversals, (iii) high pressure drop, and (iv) the poor removal of vapor slugs from the surface. These are critical issues that need to be addressed before large-scale implementation is possible. In addition, a major issue for such small-scale systems is the presence of a pump, which adversely affects the cost and reliability of the system.

Flow boiling occurs when a fluid in a forced convection flow is heated to a certain wall superheat, that is, to a given temperature above the fluid’s saturation temperature. The heat flux, degree of liquid subcooling (degrees below saturation), wall superheat, vapor quality, fluid thermal properties, flow rate, and channel geometry all affect the type of boiling flow regime observed and the resulting cooling performance. Depending on flow conditions, it has been found that flows in microchannels may exhibit one of several two-phase flow regimes, with different patterns of vapor bubbles or slugs. Harirchian and Garimella [1] extensively investigated these flow regimes, and determined the following flow regimes to
occur in flow boiling at different heat fluxes: bubbly flow, slug flow, annular flow, churn flow, and wispy-annular flow. For increasing values of heat flux the growth of vapor bubbles or slugs in microchannels can be expected to occur at progressively higher rates for any given flow rate. Given a high enough heat flux, the vapor slugs will grow at a high enough rate to grow not only in the downstream, but also in the upstream direction. Kandlikar [2] suggested that these flow instabilities can be reduced considerably by introducing an upstream pressure drop prior to a microchannel inlet. The flow instabilities and reversals, and high pressure drop were demonstrated visually using high-speed imaging by Balasubramanian and Kandlikar [3]. In their experimental study they found several microchannels in a heat sink exhibited back-flow at moderate heat fluxes and reached an early CHF. In some cases the vapor bubbles traveled upstream and reached the inlet manifold, resulting in severe flow maldistribution, in some cases leading to unexpected channel dry-out and premature onset of CHF. In a numerical study, Mukherjee and Kandlikar [4] proposed the use of channels with increasing cross-sectional area in the direction of flow to promote the unidirectional growth of vapor slugs and thus prevent flow reversals. Their concept, while simple in nature, significantly complicated the machining of the channels, as non-parallel channel walls with variable aspect ratio were needed. This proposed concept was implemented by Kandlikar et al. [5] and Kalani and Kandlikar [6]. They used open microchannels, allowing a volume of fluid to fill the space above the channels, with a tapered gap manifold, in which the cross-sectional area of the manifold increases in the flow-wise direction. This microchannel geometry was found to have a significantly reduced pressure drop in addition to featuring enhanced CHF, in comparison with microchannels with a uniform manifold. The performance of this geometry was claimed to be a result of the favorable path that is provided for the vapor bubbles to expand into, thereby reducing pressure drop and noticeably reducing flow instability. The advantage of the open microchannel geometry with a tapered manifold lies in the fact that machining remains simple, as the channel walls can be parallel, with the manifold taper providing the cross-sectional area increase in the flow-wise direction that was proposed by Mukherjee and Kandlikar [4].

A great deal of research has been performed regarding flow boiling in microchannels, typically involving water as the working fluid. Water has very favorable thermal properties, such as a high thermal
conductivity and latent heat of vaporization, allowing a large amount of heat transfer to take place during boiling. However, it is not an optimal fluid for use in flow boiling systems for the purpose of electronics cooling due to its high saturation temperature. This effectively limits its performance benefits, as extremely high chip surface temperatures (>100°C) are needed before any boiling can take place. Such temperatures are undesirable in processor chips, which typically operate closer to 50-70°C. In addition, the fact that water is not a dielectric fluid complicates its use as a coolant in electrical systems.

Refrigerants, in particular certain fluorocarbons, have gained interest from various researchers in flow boiling applications in recent times. They are dielectric, and their very low saturation temperatures make them good candidates for use in flow boiling systems. However, their extremely poor thermal properties, in addition to their high global warming potential, make them undesirable. For example, FC-72 has a thermal conductivity of 0.057 W/mK at standard atmospheric conditions, which is an entire order of magnitude lower than that of water at 0.67 WmK. Similarly, the latent heat of vaporization of refrigerants is typically lower than 100 kJ/kg, significantly limiting the thermal energy that can be transferred during boiling. The latent heat of water is significantly higher at 2256 kJ/kg.

Ethanol is another attractive alternative to water as a coolant in flow boiling applications. While it has unfavorable chemical properties such as high flammability, it has a fairly low saturation temperature at 78°C. Although this is higher than that of most refrigerants, ethanol features good thermal properties which are intermediate to those of water and refrigerants. Ethanol has a thermal conductivity of 0.179 W/mK and a latent heat of 960 kJ/kg. These properties, in addition to a low toxicity and dielectric properties make ethanol an attractive candidate for use as a coolant in microchannel flow boiling systems for electronic cooling applications.

There are few publications using ethanol in boiling research, in both pool and flow boiling. Some of the published works use binary mixtures of water and ethanol. The only flow boiling studies with ethanol studied either low heat fluxes, binary mixtures, or closed microchannel heat sinks, or as in the case of Diaz and Schmidt [7] and Wang et al. [8], investigated single microchannels. There is a great deal of
investigatory work still to be done regarding the use of ethanol in flow boiling applications in multiple microchannel heat sinks, especially in open microchannels.

The present work focuses on the flow boiling performance of ethanol in open microchannel heat sinks, with tapered gap manifolds. The heat transfer performance of ethanol is evaluated by analyzing the critical heat flux (CHF), wall superheat, heat transfer coefficient, exit vapor quality, and pressure drop. A gravity-driven flow configuration is used, as the low pressure drop of the open microchannels and tapered gap manifolds allows for flow to be supplied purely through gravity, by means of an elevated liquid reservoir, eliminating the need for a pump. Further, the applications of the gravity-driven flow configuration are investigated. If high heat fluxes can be dissipated at low flow rates with a very low pressure drop, a self-sustaining cooling device such as a thermosyphon could be implemented. Design guidelines are developed based for the manifold taper, ethanol flow rate, and the required ethanol head at an imposed heat flux, based on obtained pressure drop data.
2. LITERATURE REVIEW

Despite the potential of flow boiling to dissipate high heat fluxes at low surface temperatures, recent experimental investigations using flow boiling in microchannels as a cooling mechanism have found its performance degraded significantly in comparison to pool boiling. Flow instabilities, oscillations, and even flow reversals, along with fluctuations in pressure drop, have been observed. These instabilities severely affect cooling performance, as heat cannot be effectively removed from the surface. Furthermore, if vapor slugs are allowed to enter the inlet manifold of the channels, channel dry-out and premature CHF can occur. Recent investigations have attempted to remedy these performance issues by implementing a variety of different solutions.

2.1 Flow Instabilities and Pressure Drop

A great deal of research has been published on the topic of flow boiling with water, focusing on the hydrodynamic instabilities and pressure drop fluctuations that are observed in these systems, which result in poor cooling performance, channel dry-out, and in severe cases premature CHF. Kandlikar [2] suggested that such flow instabilities could be reduced considerably by introducing an upstream pressure drop prior to a microchannel inlet. This upstream pressure drop was later implemented in the form of an inlet valve or an inlet restrictor geometry by several authors.

Qu and Mudawar [9] performed an experimental study on the pressure drop fluctuation in a microchannel heat sink with parallel microchannels. Under normal operating conditions, after the onset of nucleate boiling (ONB) was surpassed, they observed a severe pressure drop fluctuation. The boundary between the single phase liquid and two-phase fluid mixture was observed to oscillate back and forth between the inlet and outlet of the microchannels, in some cases so severely that vapor was allowed to enter the inlet manifold. Qu and Mudawar [9] found that they were able to eliminate the pressure drop oscillation almost completely simply by introducing an inlet flow restrictor, in the form of a valve at the inlet of the channels. Their results are shown below in Fig. 1. As can be seen in the unrestricted flow case in Fig. 1(a), there is a severe pressure oscillation in the inlet and outlet pressures, $P_{in}$ and $P_{out}$ respectively. When the
inlet valve was throttled to the desired value the pressure drop increased, as the inlet pressure was raised from an average value of roughly 1.25 bar to 1.30 bar in Fig. 1(b). This, however, resulted in a reduction in pressure drop fluctuation, thus yielding a stable flow.

**Figure 1:** Pressure drop oscillations with (a) fully open valve and (b) throttled valve. *Qu and Mudawar [9]*

Balasubramanian and Kandlikar [3] studied instabilities in microchannel flow boiling systems. They performed several tests with deionized water in parallel microchannels and found severe pressure fluctuations and flow reversals in multiple channels. Their pressure drop results are shown in Fig. 2.

**Figure 2:** Pressure drop fluctuations - Balasubramanian & Kandlikar [3]

Figure 3 shows the flow reversals observed by Balasubramanian and Kandlikar [3], partially responsible for the high pressure drops observed. If the reversed flow continues in the upstream direction,
vapor may reach the inlet manifold and cause unexpected channel dry-out and premature CHF. Balasubramanian and Kandlikar [3] obtained a maximum heat flux of 316 kW/m$^2$ at a wall superheat of 10.2°C in their experiments.

![Image of microchannels]

**Figure 3:** Flow reversals in parallel microchannels. The bulk direction of flow is left to right. Balasubramanian & Kandlikar [3]

Based on the work of Balasubramanian and Kandlikar [3] and the bubble dynamics they observed, Mukherjee and Kandlikar [4] developed numerical models of the bubble growth inside a microchannel. As a nucleating bubble grows inside a microchannel, the bubble will become confined by the walls of the microchannel, and be forced to grow in the stream-wise direction. In some cases the liquid/vapor interface may begin to move backwards, opposing the bulk flow direction. If the flow rate is insufficient for the given heat flux, the overall fluid flow will be in the upstream direction, and a liquid backflow or outflow will occur.

In their numerical study, Mukherjee and Kandlikar [4] concluded that for a spherical bubble the rate of change of bubble diameter is constant, whereas for a cylindrical bubble constricted by channel walls the rate of change of bubble length is proportional to the length of the bubble/slug. Therefore, once the nucleating bubble becomes constricted by the walls of a microchannel, and is forced into a quasi-cylindrical shape, the velocities of its extremities will increase as the bubble continues to grow. This was identified as the cause of the accelerating fluid interfaces observed by Balasubramanian and Kandlikar [3].
In order to ensure unidirectional flow in parallel microchannels and eliminate the flow instabilities and reversals that are often observed, Mukherjee and Kandlikar [4] theorized that vapor plug acceleration must be prevented by relieving the pressure build-up in the microchannels in the flow-wise direction. They proposed a microchannel design with increasing area ratio in the desired direction of fluid flow, which would allow the vapor plugs to grow with constant velocity, with growth being naturally favored in the downstream direction due to the decreasing pressure gradient. A schematic of the two proposed channel designs is shown below in Fig. 4.

![Conceptual stepped and diverging microchannels – Mukherjee and Kandlikar [4]](image)

**Figure 4: Conceptual stepped and diverging microchannels – Mukherjee and Kandlikar [4]**

### 2.2 Pressure Drop Elements and Artificial Nucleation Sites

Several studies have been published in which the use of pressure drop elements, or inlet restrictors, and artificial nucleation sites are investigated, as a means of eliminating flow boiling instability. Kandlikar et al. [10] investigated the implementation of pressure drop elements (PDE) and the incorporation of artificial nucleation sites (ANS). They perform several flow boiling experiments with water in parallel microchannels 1054 x 197 μm with hydraulic diameter 332 μm.

The two pressure drop elements (PDE) were implemented in the form of inlet restrictors, at the inlet of each microchannel, offering 51% and 4% of the cross-sectional area of the microchannel, respectively. The artificial nucleation sites (ANS) were created at constant intervals on the bottom surface of the
microchannels. For the combination of both the 4% PDE and ANS, Kandlikar et al. [10] found extremely stable flow, with no flow reversals, and although the PDE introduced an added pressure drop, the authors found a significant overall reduction in pressure drop oscillation. Kandlikar et al. [10] found that the use of the 4% PDE in combination with the ANS almost completely eliminated all pressure drop oscillations. However, it also generated the highest pressure drop. While this increase in pressure drop is undesirable, the fluctuations in pressure were completely eliminated.

Figure 5: Inlet restrictors and microchannels with reentrant cavities – Kuo and Peles [11]

In a similar study, Kuo and Peles [11] investigated the use of reentrant cavities in the side walls of microchannels in combination with inlet restrictors. Several other works by these authors, such as Kosar et al. [12,13] and Kuo and Peles [14], investigated similar geometries, including interconnected reentrant cavities. The reentrant cavities in Kuo and Peles [11] were intended to function as artificial nucleation sites, reducing the wall superheat required to initiate boiling and possibly delaying the onset of CHF. The reentrant cavities in combination with inlet restrictors, shown in Fig. 5, allowed Kuo and Peles [11] to obtain very promising results, with highly stable flow boiling and excellent cooling performance. They achieved a maximum heat flux of approximately 650 W/cm², albeit at a fairly high wall superheat of 60°C, as shown in Fig. 6(a).
**Figure 6:** Plots of (a) temperature as a function of heat flux and (b) two-phase heat transfer coefficient as a function of vapor quality - Kuo and Peles [11]

### 2.3 Microchannels with Increasing Cross-Sectional Area

Mukherjee and Kandlikar [4] proposed a conceptual channel design with increasing flow area cross-section in the stream-wise direction. Similar geometries, such as diverging channels and channels with expansion angles were investigated experimentally by multiple authors.

Lu and Pan [15] tested the flow boiling heat transfer performance of diverging microchannels with artificial nucleation sites. They tested three different sets of diverging microchannels such as the one shown in Fig. 7, with different numbers of artificial nucleation sites to determine their effect on bubble nucleation, growth, and flow stability. They obtained results for inlet-header fluid temperature as a function of heat flux, shown in Fig. 8.
Figure 7: Schematic of microchannels used by Lu and Pan [15]

Figure 8: Plot of inlet temperature as a function of heat flux. Lu and Pan [15]
Lu and Pan [15] obtained the best stability improvements and oscillation reductions with the Type-2 channels, featuring nucleation sites only on the downstream half of the microchannel bottom surface. This resulted in reduced pressure drop and pressure drop oscillation, and provided the best cooling performance, with the lowest inlet temperature and inlet temperature oscillation. Lu and Pan [15] provide their results with the inlet temperature instead of wall superheat as the preferred measure of performance, as their interest lies in the temperature oscillations and spikes at the inlet, which are indicative of flow oscillations or reversals reaching the inlet header. Fig. 8 shows their best results from the Type-2 channels at a mass flux of 297 kg/m²s, with a heat flux of roughly 375 kW/m² at an inlet temperature increase of only 3°C.

Miner et al. [16] performed a numerical study on an optimized microchannel geometry featuring microchannels with expanding cross-sectional area towards the tops of the channels. A schematic of the control volume for their model is shown below in Fig. 9 (a).

![Figure 9](image)

**Figure 9:** (a) Schematic of model control volume; (b) Contour plot of attainable CHF values for different values of channel height and width expansion – Miner et al. [16]

Miner et al. [16] chose CHF as the most convenient measure of maximum heat transfer performance, and used this to determine the degree to which flow boiling performance could be enhanced with the proposed channel cross-sectional area changes. They found CHF enhancement to be more sensitive to changes in height than width, as shown in Fig. 9(b). If sufficient width and height expansion of the microchannels is provided, the model of Miner et al.[16] predicts a CHF upwards of 600 W/cm² to be attainable with water at standard atmospheric conditions.
A further step towards implementation of variable cross-sectional area microchannels was taken by Kandlikar et al. [5]. They built on the proposed concept of Mukherjee and Kandlikar [4] by simplifying its practical implementation. They used simple parallel microchannels with constant area cross-section, while implementing a manifold gap above the channel fins. This manifold gap was then tapered to provide an increasing channel area in the stream-wise direction, resulting in an increasing volume into which the rapidly-expanding vapor bubbles observed at high heat fluxes may expand into. A schematic of the tapered gap manifold used by Kandlikar et al. [5] is shown in Fig. 10. The test section utilized by Kandlikar et al. [5] in their experimental setup consisted of a tapered manifold and a uniform manifold block, in combination with the open microchannel geometry mentioned.

![Schematic of tapered microchannel manifold](image)

**Figure 10: Schematic of tapered microchannel manifold – Kandlikar et al. [5]**

Kandlikar et al. [5] determined that the tapered manifold performed better than the uniform manifold, as shown by the boiling curve in Fig. 11(a), and the pressure drop shown in Fig. 11(b). They obtained a maximum heat flux of 500 W/cm² at a wall superheat of only 26.2°C. CHF was not reached in any of the configurations tested. The pressure drop for the tapered manifold remained below 20 kPa for all heat fluxes tested.
Figure 11: (a) Heat transfer performance at different flow rates and (b) pressure drop performance of tapered manifold (TM) vs. uniform manifold (UM) – Kandlikar et al. [5]

A similar approach was taken by Balasubramanian et al. [17] in their experimental study of stepped-fin microchannels. They fabricated microchannel heat sinks with parallel microchannels, the height of which increase with several steps in the flow-wise direction. They tested several stepped fin geometries, with different numbers of channels, heat sink widths, and step positions.

Figure 12: Stepped fin microchannels implemented by Balasubramanian et al. [17]

Balasubramanian et al. [17] obtained extremely high heat transfer performance one of their stepped fin microchannel geometries, the plots of which are shown below in Fig. 13(a) and (b). They obtained a maximum heat flux of 400 W/cm² at a wall superheat of only 15°C, and a heat transfer coefficient of approximately 170,000 W/m²°C.
In a recent experimental study, Miner et al. [18] investigated the conceptual expanding microchannels that had been proposed in the numerical work of Miner et al. In a similar approach to that of Kandlikar et al. [5], Miner et al. [18] fabricated microchannels with constant cross-section, but with an expanding base, defined by an expansion angle. As shown in Fig. 14(a), this results in an expanding channel area, as the base cross-section reduces in height.

In their parametric study, Miner et al. [18] found that pressure drop increases with heat flux, as shown in Fig. 14(b). However, they found that this relation decreases in intensity with increasing degree of channel cross-sectional area expansion. The channel expansion reduces the detrimental effects of heat flux on the channel pressure drop, which the authors expect will delay CHF.

Expanding on the previous work of Kandlikar et al. [5] on open microchannels and tapered manifolds, Kalani and Kandlikar [6] evaluated the pressure drop performance of these same microchannel geometries (Fig. 10) in flow boiling with water, along with a parametric study on the effect of the manifold taper height on flow boiling.
Figure 14: (a) Schematic of expanding channels, (b) Effect of expansion angle. Miner et al. [18]

Kalani and Kandlikar [6] found that the tapered manifolds performed extremely well in reducing pressure drop across the microchannels. Figure 15(a) shows the pressure drop for a tapered manifold with a microchannel chip resulted in a maximum pressure drop of only 3.3 kPa. In their parametric study on manifold taper they concluded that tapers resulted in higher heat fluxes compared to the uniform manifold, and that higher tapers resulted in improved performance.

Figure 15: Plots of (a) pressure drop and (b) wall superheat as functions of heat flux – Kalani and Kandlikar [6]
2.4 Experimental Investigations in Two-Phase Cooling Using Ethanol:

While most experimental investigations into heat transfer enhancement in both pool and flow boiling have used water as the working fluid, implementation of such systems in the cooling of electronic components requires a fluid with a lower boiling point, which preferably has dielectric properties. Ethanol is a suitable alternative to water for such applications, with heat transfer properties far superior to those of refrigerants, and a lower saturation temperature than water at standard atmospheric conditions. There are several publications using ethanol in both pool and flow boiling.

Gogonin and Kutateladze [19] studied the effect of heater size on the critical heat flux (CHF) of ethanol in pool boiling at a range of vacuum pressures. The authors performed experiments on a heater surface insulated on one side, with variable widths between 5 and 50 mm. They varied vacuum pressures (i.e. gage pressures below atmosphere) between 1x10^{-5} and 52x10^{-5} Pa. The best performance of 118 W/cm² was obtained with a heater width of 10 mm, at a vacuum pressure of 24x10^{-5} Pa.

**Table 1: CHF data of pure components, Fujita and Bai [20]**

<table>
<thead>
<tr>
<th>Liquid</th>
<th>q(_{\text{CHF}}) [MWm(^{-2})]</th>
<th>K</th>
<th>R’</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>0.982</td>
<td>0.118</td>
<td>0.0983</td>
</tr>
<tr>
<td>Methanol</td>
<td>0.595</td>
<td>0.145</td>
<td>0.147</td>
</tr>
<tr>
<td>Ethanol</td>
<td>0.466</td>
<td>0.127</td>
<td>0.161</td>
</tr>
<tr>
<td>n-Butanol</td>
<td>0.411</td>
<td>0.133</td>
<td>0.160</td>
</tr>
<tr>
<td>Benzene</td>
<td>0.332</td>
<td>0.140</td>
<td>0.152</td>
</tr>
<tr>
<td>n-Heptane</td>
<td>0.285</td>
<td>0.163</td>
<td>0.172</td>
</tr>
<tr>
<td>Ethylene Glycol</td>
<td>0.999</td>
<td>0.213</td>
<td>0.136</td>
</tr>
<tr>
<td>Ethanol/Water at X1 = 0.87</td>
<td>0.577</td>
<td>0.144</td>
<td>0.162</td>
</tr>
<tr>
<td>Methanol/Water at X1 = 0.60</td>
<td>0.594</td>
<td>0.158</td>
<td>0.150</td>
</tr>
</tbody>
</table>

Fujita and Bai [20] performed an experimental study involving several binary mixtures of alcohols and water at atmospheric pressure, which were tested to obtain CHF data for each mixture in pool boiling conditions in a submerged horizontal platinum wire. They tested a selected set of seven binary mixtures over the full range of concentrations. Fujita and Bai found that aqueous mixtures of ethanol and methanol
showed significant enhancement of CHF compared to that of the pure components. A tabulated summary of their results is shown below in Table 1. The authors obtained a CHF of 46.6 W/cm² for pure ethanol.

Inoue et al. [21] investigated the effect of surface tension on CHF. They tested mixtures of various ethanol fractions in water, added a surface-active agent to the mixtures, and determined the resulting heat transfer coefficient, onset of boiling, and CHF. Inoue et al. [21] carried out their experiments at surfactant concentrations of 0-5000 ppm, and across the range of ethanol fractions. Their results showed that heat transfer coefficients were enhanced for lower ethanol fractions (C ≤ 0.5) and at low heat fluxes, and that low surfactant concentrations reduced surface tension, allowing for enhanced nucleate boiling heat transfer. Further, they found the enhancement effect of the surfactant subsided at concentrations over 1000 ppm, beyond which the surfactant no longer had a depressing effect on surface tension, thus nullifying the heat transfer enhancement. These results are depicted in Fig. 16 below. Inoue et al. obtained a maximum CHF of approximately 1750 kW/m² or 175 W/cm².

**Figure 16:** (a) CHF as a function of mass fraction (○ Cs = 0 ppm, ● Cs = 600 ppm), and (b) enhancement of heat transfer coefficient by surfactant as a function of heat flux; h' = HTC with surfactant, h = HTC without surfactant. Inoue et al. [21]

Hetsroni et al. [22] performed an experimental investigation into the boiling bubble dynamics in multiple-channel, parallel microchannel heat sinks at low vapor qualities. They related the flow instabilities and oscillatory behavior of the flow in the channels to the periodic fluctuations in pressure drop, fluid inlet temperature, and surface temperature. The onset of CHF was also correlated to the reduction of initial film
thickness for a given fluid. Hetsroni et al. [22] obtained results for several heat sinks using both water and ethanol as the working fluids. They obtained a maximum heat flux of 162 kW/m², or 16.2 W/cm² at a mass flux of 124 kg/m²s, using a microchannel heat sink with 13 parallel triangular channels of hydraulic diameter 220μm.

Diaz and Schmidt [7] investigated flow boiling in a single rectangular microchannel using water and ethanol as the working fluids. They used infrared thermography to measure the outer wall temperatures of the channel, allowing for accurate measurement of the temperature distribution, as well as the temperature change across the channel over short time intervals. Figure 17 shows the heat transfer coefficients as functions of vapor quality, obtained with ethanol at a mass flux of 500 kg/m²s. Diaz and Schmidt [7] reported a maximum heat flux of 188.3 kW/m², or 18.83 W/cm².

![Figure 17: Heat Transfer Coefficient as a function of vapor quality - Diaz & Schmidt [7]](image)

Wang et al. [8] also used infrared thermography to obtain temperature distributions in single microchannels using transparent heaters. The authors then used the obtained data to evaluate the accuracy of several existing heat transfer correlations for boiling in microchannels, as well as that of a modified correlation proposed by the authors. Wang et al. [8] used FC-72 and ethanol as the working fluids in their experimental investigation. They obtained a maximum heat flux of 0.88 kW/m² or 0.088 W/cm² with ethanol in a microchannel of hydraulic diameter 1454 μm.
Fu et al. [23] studied the behavior of several water-ethanol mixtures in flow boiling with diverging microchannels and artificial nucleation sites. They found that CHF was enhanced at low mole fractions of ethanol in water, beyond which a decrease of CHF was observed as ethanol mole fraction was increased further. Fu et al. [23] obtained a CHF of nearly 490 kW/m², or 49 W/cm² for pure ethanol.

Kalani and Kandlikar [24] investigated the performance of ethanol in pool boiling at subatmospheric pressures on microchannel chips. They performed tests at several pressures ranging from 101.3 kPa, or atmospheric pressure, down to 16.7 kPa, using several microchannel dimensions. They varied the channel width, depth, and fin width. Kalani and Kandlikar [24] reached a CHF of 113.3 W/cm² at a wall superheat of 20.1°C for a microchannel chip with a channel depth of 470 µm, channel width of 194 µm, and a fin width of 402 µm.

The work of Kalani and Kandlikar [24] showed that high heat flux performance was possible in microchannel chips by boiling ethanol. Further, given the promising performance achieved with open microchannels and tapered manifolds by Kandlikar et al. [5], and the exceptional pressure drop performance recorded by Kalani and Kandlikar [6], the present work seeks to use ethanol as a coolant in a flow boiling system using open microchannels with tapered manifolds. Furthermore, the low pressure drop that is associated with this geometry, as was found by both Kandlikar et al. [5] and Kalani and Kandlikar [6],
enables the use of a gravity-driven flow configuration, eliminating the need for a pump. The feasibility of implementing a system capable of achieving high heat flux dissipation at low flow rates and low pressure drop, in a gravity-driven flow, is investigated in this work.
3. EXPERIMENTAL SETUP

The experimental setup consists of a gravity-driven pipe flow system featuring two fluid reservoirs, one of which is elevated to provide the gravitational fluid head needed to drive the flow at the desired flow rates, and compensate for the expected system pressure drop. A schematic of the experimental setup is shown in Fig. 19. The principle of operation of the system relies on the gravitational pressure gradient acting on reservoir #1, which is elevated to provide a total head of 2.68 m to allow for a pressure drop of 20 kPa at the highest desired flow rate of 80 mL/min. Details on the elevated reservoir assembly are given in Appendix A. During regular testing, fluid flow proceeds vertically downwards from the elevated reservoir through the main loop to the preheater section, which features curved piping to provide a smooth transition from the vertical to the horizontal plane, ensuring steady flow and minimizing pressure drop in this section. The preheater consists of two Omega HTC-060, 125 W rope heaters, which heat the ethanol to the desired inlet temperature of 55°C, corresponding to a degree of subcooling of 23°C below saturation. The use of rope heaters allows for placement of the heaters on non-straight piping sections with optimal contact. After the preheater section, the ethanol is heated past saturation in the test section, in which the desired chip, microchannel, and tapered gap manifold geometries are tested at heat fluxes across the nucleate boiling range and up to CHF. The heater block in the test section is heated via four Watlow FIREROD®, 400 W cartridge heaters, which are inserted into the copper block featuring the plain or microchannel chip. The two-phase mixture of ethanol liquid and vapor is cooled by the condenser back to the liquid phase, before exiting to the bottom reservoir #2. Reservoir #2 is located on the floor to maximize the gravitational pressure gradient. Once testing is complete, the filled reservoir #2 is exchanged with reservoir #1, which is now empty, and a further test can be performed. Reservoir exchange may only be performed at times when there is no concurrent testing in the adjacent flow boiling loop and all nearby PSUs are turned OFF, to avoid any possible sources of ignition.
Two 20 L (5 gal) flexible plastic reservoirs are used. The bags are made of LDPE (Low Density Polyethylene), which is resistant to ethanol corrosion. They feature a handle for ease of transport and handling, and a top lid for filling. The shower heads were removed from the bottom hoses of the bags, which were fitted with ¼” ball valves. Additionally, valves are provided in all key locations where the reservoirs are separated from the system such that minimal leakage occurs during reservoir exchange. Flow rate is measured prior to the preheater section using an OMEGA FLR1008ST air/water flow meter, with a flow range of 20–200 mL/min with ±1% full-scale accuracy. The condenser is a shell and tube heat exchanger with a maximum temperature of 425°C, and maximum fluid temperature difference of 70°C from hot to cold side. The condenser is cooled by a cold water loop with an air-cooled chiller featuring two
fans with a cooling capacity of 2080 kW (7100 Btu/hr) at a water flow rate of 7.6 L/min (2 GPM). Even at CHF and at low flow rate testing, the condenser was capable of cooling the ethanol down to roughly 29°C at the condenser outlet. The condenser is an essential component to prevent damage to the flexible plastic reservoirs caused by high-temperature ethanol, which could produce undesirable pressures and thermal expansion of the reservoirs, eventually causing tearing and leakage.

Temperatures are monitored at the test section inlet and outlet, chip heater block, and condenser outlet using K-type thermocouples, having a rated temperature range of -200-1250°C. The accuracy of the thermocouples was determined via a 3-point NIST calibration procedure. The resulting bias error values were all on the order of ±0.2°C.

3.1 Experimental Procedure:

Testing is initiated by turning on all sensor and solenoid valve PSU’s, turning on the main heater PSU’s, and launching the LabVIEW Virtual Instrument (VI), featuring all the automated loop controls and data acquisition interface. The elevated reservoir must be filled with sufficient fluid volume to conduct a full boiling test, and the pump and fans of the condenser cooling loop must be engaged. First, the valves V-01 and V-03 in Fig. 19 are opened; then valve V-02 is adjusted to set the desired flow rate. The preheater is engaged to the required power input via voltage-control through the LabView VI interface, to precondition the ethanol to the desired inlet temperature of 55°C (subcooling of 23°C). Once the desired inlet temperature is reached, the cartridge heaters of the test section are enabled. Voltages are increased in 5 V increments from 50 V up to the highest possible voltage, where CHF is reached. Data is logged for 10 sec intervals, at a sample time interval of 0.2 sec.

The top reservoir was elevated to provide sufficient fluid head, ensuring steady flow even at higher flow rates. Previous studies by Kandlikar et al. [5] and Kalani and Kandlikar [6] found the open microchannel and tapered manifold configuration to yield a pressure drop below 10 kPa. Assuming a test section pressure drop of 10 kPa, and accounting for additional frictional pressure drop in the tubing of the system, a value of 20 kPa was assumed as the expected pressure drop in the system. At a height of 2.68 m,
an ethanol flow rate of 80 mL/min can be sustained up to a maximum pressure drop of 20.08 kPa. The pressure drop across the test section was monitored using a Honeywell FP2000, 34.47 kPa (5 psi) differential pressure transducer. The maximum pressure drop, which occurs at CHF, did not exceed 9 kPa for any of the tested configurations. Tests were performed at flow rates of 40-80 mL/min. For a reservoir filled with 11 L (3 gal.) of ethanol, fluid can be supplied for 2.4 hours at 80 mL/min before the reservoir is empty, giving more than sufficient time to conduct a flow boiling test up to CHF.

### 3.2 Automated CHF Loop

An automated CHF loop was implemented, consisting of a plunger and solenoid valves shown in Fig. 19. This loop provided auxiliary cooling at CHF, and was triggered automatically via the LabVIEW VI and solenoid valves. A 7.6 L (2 gal.) plunger was used to manually pressurize the CHF loop. A pressure gauge was used to monitor the pressure of the loop prior to CHF. The loop was pressurized to roughly 100 kPa (15 psi) at the start of each test, and care was taken to ensure the pressure did not drop below 70 kPa (10 psi) at high heat fluxes. An instantaneous readout of the chip surface temperature on the LabView VI was used to determine signs of imminent CHF, at which point the tests were stopped immediately, power supplies were turned off, and the CHF loop was engaged. This was accomplished by setting a limit to the rate of change of temperature (dT/dt), which was programmed to trigger at 0.01°C/Δt. This temperature change rate was computed in real-time as an average over 60 time samples, in order to filter out fast-transient temperature fluctuations which commonly occur during measurement, but do not represent the onset of imminent CHF. A second trigger was implemented by setting a maximum allowable surface temperature of 150°C. If either trigger value was exceeded during testing, the CHF loop would automatically engage. Surface temperatures did not exceed 130°C during any of the conducted tests.

A 19 L (5 gal.) auxiliary reservoir was provided to collect the water-ethanol mixture exiting the system in this configuration. This mixture contained mostly water, and the small volume of ethanol that was contained in the section of piping between the solenoid valves. While this configuration led to losses of small volumes of ethanol every time the CHF loop was engaged, it allowed for testing of CHF conditions
with good repeatability, without damaging any of the test equipment. Furthermore, it was found during testing with ethanol that the temperature overshoot at CHF is quite small compared to that of water. This made it possible in many cases to simply disengage the heater power supply units (PSUs) at CHF and cool the system slowly over time with ethanol in the regular configuration, provided enough fluid volume was still available to do so.

3.3 Data Acquisition System

Data acquisition was performed via a NI CompactDAQ system. All calibrated sensor data was processed and logged via a NI LabVIEW VI. The virtual instrument also controlled the PSUs to the preheater and main heaters, as well as the actuation of the solenoid valves. Data was recorded for a time span of 10 sec at each heat flux tested. A minimum waiting period of 10 min was established between data logging at each heat flux, in order to allow the system to reach steady state. Steady state was defined as the point at which temperature changes fall below \( \pm 0.2^\circ\text{C} \) within a 10 sec time period.

3.4 High-Speed Flow Visualization

Flow visualization data using high-speed video were recorded to observe bubble dynamics and flow patterns for each boiling test. A Photron FASTCAM 1024PCI capable of capturing video at frame rates up to 100,000 fps was used. For most boiling tests video was recorded at 3000 to 6000 fps to allow for a higher resolution, covering a larger area of the surface of the chip. A high intensity light source was placed near the test section manifold to illuminate the chip surface and enable the high-speed camera to provide quality images. The high-speed camera is mounted to a 3-axis motor assembly, featuring 3 electric motors capable of moving the camera in 3 different axes. The high-speed video is recorded from the Photron FASTCAM Viewer software, which allows the user to select frame rate, shutter speed, and resolution settings, as well as the adjustment of brightness, contrast, and gamma level for image optimization.
3.5 Liquid Degassing:

Liquid degassing is an integral part of the system constructed for the present thesis work, and is a concern for all flow boiling applications. Due to natural diffusion phenomena, air will tend to diffuse into a liquid that is exposed to the atmosphere. A large volume of dissolved air can significantly affect the performance of any flow boiling system, due to the typical decrease of air solubility in a fluid with increases in temperature. Thus, as the fluid is heated, its solubility of air by volume decreases. Once a sufficiently high temperature is reached, the air will begin to boil off, leading to an unexpected and premature onset of nucleate boiling (ONB), at a lower than desired temperature. Thus, in order to obtain reliable flow boiling data, the fluid must be degassed prior to testing.

Several methods are available to accomplish degassing, the preferred of which is to heat the liquid in the reservoir to saturation, allowing the dissolved air to escape the liquid. While this method is effective with water, the oxygen solubility of ethanol is quite high, with a weak temperature dependence in comparison to water. Equilibrium values of volume fraction of oxygen in water and ethanol were computed from data given by Battino et al. [25], and shown in Fig. 20. The equilibrium oxygen solubility of water drops from 29.63 mL/L at 20°C to 17.11 mL/L at 100°C, which equates to a 42% decrease in oxygen solubility. Meanwhile, the oxygen solubility of ethanol is much higher with equilibrium values of oxygen solubility in ethanol decreasing from 262.66 mL/L at 20°C to 246.54 mL/L at 70°C, which represents a decrease of only 6% in oxygen solubility. These properties result in a less effective degassing procedure.
Degassing was performed by flowing fluid through the main flow section as during a regular test, and the preheater section heaters are engaged to heat the ethanol to saturation. The cartridge heaters of the test section remained disconnected during the degassing procedure.

The two-phase mixture of ethanol, ethanol vapor, and non-condensable gases, flows through the condenser, where the ethanol is cooled back to single phase. The gas mixture of air and ethanol vapors are extracted through valve V-04 by means of the vacuum pressure provided by the vacuum pump.

A film flow is formed in the section of tubing surrounding valve V-04 thanks to large-diameter tubing, where due to gravity the liquid is confined to the bottom portion of the tubing, while vapors are allowed to accumulate at the top. The vacuum pressure delivered to valve V-04 via the vacuum pump reaches the vapor in this section, allowing it to be removed from the main flow section. The vacuum pressure is set to a very weak vacuum pressure level of approximately 1 kPa below atmosphere, ensuring gases may be removed, but weak enough that liquid cannot enter this region due to its higher density.

As a safety precaution, a vapor-liquid separator vessel was included between valve V-04 and the vacuum pump. This vessel, shown schematically in Fig. 21, allows liquid to fall into the volume of the

Figure 20: Volume Fractions of Oxygen in Liquid (Data reproduced from Battino et al. [25])
vessel, while vapors and gases are allowed to continue flowing to the vacuum pump, and out into the ventilated exhaust hood. This prevents damage to the pump from ethanol corrosion, as any liquid that may accidentally be drawn from the system at valve V-04 becomes trapped within the vessel.

**Figure 21: Schematic of vapor-liquid separator vessel**

To achieve a satisfactory level of degassing of the ethanol, the degassing procedure was repeated several times, exchanging the flexible reservoirs multiple times. After 5 iterations, the ethanol was considered sufficiently degassed. The state of degassing was confirmed by inspection of the flow via high-speed video, ensuring no gases are visible in the flow at the test section inlet below saturation temperatures.

### 3.6 Test Section

The test section consists of the heater block and chip surface, cartridge heaters, the tapered gap manifold, and heater block thermocouples. The microchannel geometry is the same as that of Kandlikar *et al.* [5], and Kalani and Kandlikar [6], featuring open microchannels with a tapered gap manifold. The open microchannel geometry with a tapered manifold, shown in Fig. 22, addresses key issues related to the low CHF observed in conventional microchannel flow boiling systems: (1) the removal of vapor from the chip surface and (2) the supply of liquid to the surface. Both of these considerations contribute to the prevention heater surface dry-out. The arrangement provides an increasing flow area cross-section in the flow-wise direction, into which growing vapor bubbles can expand. This allows for removal of vapor and
replenishment of liquid to the surface with the benefit of a low pressure drop. Temperature in the heater block is monitored at three different heights, with three evenly spaced thermocouples, separated by 3 mm, which allow for the calculation of heat flux and surface temperature.

3.6.1 Manifold Blocks:

The manifold blocks are machined from Polysulfone, which has a glass transition temperature of 140°C, and polished to improve transparency for the purpose of flow visualization. Figure 22 shows the tapered manifold in the test section assembly. Three manifold blocks were tested, having tapers of 2%, 4%, and 6%. These defined tapers are measured by their corresponding taper heights. The taper height is defined by the manifold taper and the total length of the manifold, from microchannel inlet to exit, of 10 mm. The percent values of the taper thus reflect the percent change in height over the length of the microchannels. The three tapers (2%, 4%, and 6%) have tapers heights of 200 µm, 400 µm, and 600 µm, respectively. A silicon gasket defines the inlet gap height in Fig. 22, which is constant across all manifolds, with a height of 127 µm. The nominal exit gap height for the three tapers is thus 327 µm, 527 µm, and 727 µm, respectively. Table 2 summarizes the actual taper dimensions of the manifolds tested in the present work. The deviation from the nominal values described is minimal, with a maximal deviation of 5% from the nominal dimension.

Table 2: Microchannel chip dimensions

<table>
<thead>
<tr>
<th>Taper</th>
<th>Inlet Gap Height</th>
<th>Nominal Taper Height</th>
<th>Measured Taper Height</th>
<th>Measured Outlet Gap Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>2%</td>
<td>127 µm</td>
<td>200 µm</td>
<td>205 µm</td>
<td>332 µm</td>
</tr>
<tr>
<td>4%</td>
<td>127 µm</td>
<td>400 µm</td>
<td>418 µm</td>
<td>545 µm</td>
</tr>
<tr>
<td>6%</td>
<td>127 µm</td>
<td>600 µm</td>
<td>632 µm</td>
<td>759 µm</td>
</tr>
</tbody>
</table>
3.6.2 Heater Block and Microchannel Fabrication:

The heater blocks were machined with a CNC vertical mill from a solid copper block. A schematic of the heater block for the microchannel test section is shown in Fig. 23. All chips have dimensions of 15 mm by 15 mm, the microchannel chips having microchannels located in the central 10 mm by 10 mm area of the chip surface. The microchannel chips will henceforth be referred to as MC1 and MC2. Table 3 shows the dimensions of the microchannels featured on the chip surface. The microchannels were machined using micro-tools on a CNC vertical mill.
Figure 23: Schematic of the heater block and microchannel chip

Table 3: Microchannel chip dimensions

<table>
<thead>
<tr>
<th>Chip</th>
<th>Channel Width</th>
<th>Channel Depth</th>
<th>Fin Width</th>
</tr>
</thead>
<tbody>
<tr>
<td>MC1</td>
<td>299 µm</td>
<td>250 µm</td>
<td>106 µm</td>
</tr>
<tr>
<td>MC2</td>
<td>493 µm</td>
<td>224 µm</td>
<td>104 µm</td>
</tr>
</tbody>
</table>

4. HEAT FLUX CALCULATION

The four WATLOW cartridge heaters of the heater block are voltage-controlled via the LabVIEW VI to provide the desired heating required to test at various heat fluxes, up to CHF. The heat flux into the test chips was calculated as in previous works [5,6] using Fourier’s law for one dimensional conduction in the heater block, defined as shown in Eqn. (1).

\[ q'' = -k_{Cu} \frac{dT}{dx} \] (1)

The temperature gradient was approximated using a three-point backward difference Taylor series approximation, shown in Eqn. (2), using the temperature readings from the three thermocouples mounted to the heater block shown in Fig. 27, and the separation between them.

\[ \frac{dT}{dx} \approx \frac{3T_1 - 4T_2 + T_3}{2\Delta d} \] (2)

The surface temperature was calculated using the top heater temperature and the heat flux from Eqn. (2), accounting for the thermal resistance of the copper and distance between the top thermocouple and the surface, as shown in Eqn. (3).

\[ T_s = T_1 - q'' \left( \frac{d_1}{k_{Cu}} \right) \] (3)

Here, \( d_1 \) is the distance between the top heater thermocouple and the surface, which is 1.5 mm for all chips. The heat transfer coefficient was calculated from the chip surface temperature and the heat flux, based on the projected area of the heater base, and is given by Eqn. (4).
\[
 h = \frac{q''}{\Delta T_{\text{sat}}} = \frac{q''}{(T_s - T_{\text{sat}})}
 \]  

(4)

4.1 Heat Loss Study

A heat loss study was conducted via numerical simulation in ANSYS® Fluent® to establish the amount of heat lost at the edges of the chip surface. This was done by developing a 2-D model of the heater block geometry in ANSYS® GAMBIT, and importing the model into Fluent®. The heater block is modelled with its true dimensions, including the ceramic insulation located on its sides and the silicone gasket on the top surface, as shown in Fig. 24(a). The central portion of the surface, which would normally be cooled by the fluid of choice in the flow boiling system, is perfectly insulated. The input heat from the bottom surface can therefore only flow to the sides of the chip. In this manner, any input heat flux will represent the heat loss for a given surface temperature \( T_s \). There is a contact resistance associated with the ceramic insulation that is not accounted for. This thermal contact resistance arises as a result of the ceramic insulation blocks not being affixed to the surfaces of the copper block, and would result in lower heat losses than the portrayed scenario. Thus, the model as stated represents a worst-case scenario of heat loss at the sides of the chip.

![Figure 24: (a) Heat loss simulation model, (b) Plot of heat loss as a function of wall superheat](image)

Several heat fluxes are simulated in order to cover the entire operating range for the chip surface temperature across the nucleate boiling range, up to CHF. The heat loss and surface temperature are
correlated via a linear trendline equation to establish an estimate of heat loss at every heat flux. A plot of the simulation results is shown in Fig. 24(b) along with the corresponding trendline.

The final heat flux values calculated from the results are thus adjusted taking into account the predicted heat loss for the corresponding wall superheat, as shown in Eqn. (5).

\[
q''_{\text{adjusted}} = q''_{\text{measured}} - q''_{\text{loss}}
\]

\[
= q''_{\text{measured}} - (0.0308\Delta T_{\text{sat}} + 1.675)
\]  

(5)

For the purpose of this analysis it was chosen to neglect the contact resistance between the ceramic blocks and the heater side faces. In reality, this contact resistance will not be negligible due to the lack of adhesion or thermal interface material between the ceramic blocks and the heater side faces. As a result, it is expected that small air pockets will form between the ceramic blocks and the heater side faces, effectively worsening the thermal conductivity in those areas. This estimate of heat loss at the chip edges therefore represents a worst-case study of heat loss.

5. ETHANOL: PROPERTIES, FLAMMABILITY & PRECAUTIONS

Ethanol is an attractive alternative to water for flow boiling systems intended for electronics cooling applications. Its thermal properties are intermediate to those of water and refrigerants, with a boiling point of 78°C, thermal conductivity of 0.179 W/mK, and latent heat of 960 kJ/kg at atmospheric pressure. Furthermore, its dielectric properties make it ideal for electronics cooling. However, as with all flammable or volatile fluids, care must be taken to ensure there are no sources of sparks or open flames to ignite the ethanol.

All PSU cables are protected by insulating junction boxes which connect them to the preheater and test section block heaters, leaving no exposed, high-power cables in the vicinity of the experimental setup where leakage could foreseeably occur. The junction boxes, power cables, and fuses, are rated to 20 A, which far exceeds the currents used in testing, even at CHF, minimizing the risk of sparks. There are no other sources
of sparks or flame that could foreseeably occur during regular testing. Furthermore, ethanol vapors from
degassing, and during regular testing, are removed safely with a vacuum pump and directed to a ventilated
exhaust hood from a safe location, removed from the vicinity of the heated test section.

Autoignition of ethanol at high heater temperature may also be concern. Autoignition temperature is
defined as the lowest temperature at which a substance will spontaneously ignite, without any external
source of ignition. The autoignition temperature of ethanol is 365°C, which will not be exceeded under any
foreseeable circumstance. Even at high heat flux testing and near CHF, the heater surface temperature did
not exceed 130°C. Temperatures far beyond this are beyond the CHF limit, and lie within the film boiling
regime, which is not the focus of the present work as the heat transfer coefficients associated with this
boiling regime are very low.

6. UNCERTAINTY ANALYSIS

An in-depth uncertainty analysis was performed on all major computed quantities listed in the results.
The standard expression for error propagation shown in Eqn. (6) was used. The bias uncertainties of the
measured quantities were quantified by the sensor calibration procedures, and the precision errors of the
sampled data were approximated as twice the standard deviation of samples of each data point in order to
account for data variation with a 95% confidence interval. The total uncertainty values in the results are
shown in all figures with appropriate error bars. For errors below 0.5% no error bars are displayed.

\[
U_p = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial p}{\partial \sigma_i} U_{\sigma_i} \right)^2}
\]  
\( 6 \)
The total uncertainty in heat flux was derived by substituting the Eqns. (1) and (2) into Eqn. (6). The resulting expression is shown in Eqn. (7) (see Appendix B for derivation).

\[
\frac{U_q^2}{q^n} = \sqrt{\frac{U_{kCu}^2}{k_{Cu}^2} + \frac{U_d^2}{\Delta d^2} + \frac{9U_{T1}^2}{\alpha^2} + \frac{16U_{T2}^2}{\alpha^2} + \frac{U_{T3}^2}{\alpha^2}}
\]  

The term \( \alpha \) in Eqn. (6) is shown below in Eqn. (8), and is used to simplify the derivation of Eqn. (7). In Eqn. (7), the total uncertainties in the three temperature values were obtained from the combination of bias and precision errors of their respective measurements. The bias error values of the temperatures were determined from the averaged standard deviation of samples obtained during the thermocouple calibration procedure.

\[
\alpha = 3T_1 - 4T_2 + T_3
\]  

The total uncertainty in heat transfer coefficient was derived by substituting Eqn. (4) into Eqn. (6) and is shown below in Eqn. (9) (see Appendix B for derivation).

\[
\frac{U_h}{h} = \sqrt{\frac{U_{q^2}}{q^n^2} + \frac{U_{Tw}^2}{\Delta T_{sat}^2} + \frac{U_{Ts}^2}{\Delta T_{sat}^2}}
\]  

As the pressure drop is a measured quantity in this work, there is no error propagation, and the total uncertainty was calculated from the combination of the bias uncertainty in measurement, obtained from the pressure sensor calibration, and the precision error, calculated from the standard deviation of samples of each data point.

\[
\frac{U_{\Delta P}}{\Delta P} = \sqrt{B_{\Delta P}^2 + p_{\Delta P}^2} = \sqrt{B_{\Delta P}^2 + 4\sigma_{\Delta P}^2}
\]  

The total uncertainty in heat flux was consistently below 7% at high heat fluxes, with values below 13 W/cm². The total uncertainty in heat transfer coefficient remained within 10% on average, with maximum values below 5000 W/m²K. The total uncertainty in pressure drop was within 10%, though this uncertainty was predominantly caused by pressure drop fluctuation, leading to a large precision error. The bias error in
pressure drop was very low thanks to high sensor accuracy. The uncertainties in wall temperature and vapor exit quality were consistently below 0.5%. For the derivation of the above Eqns. (7) and (9), as well as those of surface temperature and exit vapor quality, please refer to Appendix B.
7. RESULTS

Results were obtained for all chips and manifolds, at flow rates between 40 and 80 mL/min. The 2% manifold taper was only tested up to 60 mL/min due to the higher pressure drop associated with this taper. The complete test matrix is shown below in Table 4.

Table 4: Test Matrix - test cases highlighted in red with an asterisk* were not recorded

<table>
<thead>
<tr>
<th>Chip</th>
<th>Channel Width µm</th>
<th>Manifold Taper %</th>
<th>Flow Rate mL/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>2%</td>
<td>40</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2%</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2%*</td>
<td>80*</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4%</td>
<td>40</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4%</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4%</td>
<td>80</td>
<td></td>
</tr>
<tr>
<td></td>
<td>6%</td>
<td>40</td>
<td></td>
</tr>
<tr>
<td></td>
<td>6%</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td></td>
<td>6%</td>
<td>80</td>
<td></td>
</tr>
<tr>
<td>MC1</td>
<td>299</td>
<td>2%</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td>299*</td>
<td>2%*</td>
<td>80*</td>
</tr>
<tr>
<td></td>
<td>299</td>
<td>4%</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td>299</td>
<td>4%</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td>299</td>
<td>4%</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>299</td>
<td>6%</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td>299</td>
<td>6%</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td>299</td>
<td>6%</td>
<td>80</td>
</tr>
<tr>
<td>MC2</td>
<td>493</td>
<td>2%</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td>493*</td>
<td>2%*</td>
<td>80*</td>
</tr>
<tr>
<td></td>
<td>493</td>
<td>4%</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td>493</td>
<td>4%</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td>493</td>
<td>4%</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>493</td>
<td>6%</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td>493</td>
<td>6%</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td>493</td>
<td>6%</td>
<td>80</td>
</tr>
</tbody>
</table>
The results are presented in the boiling curves shown below, plotting heat flux as a function of wall superheat. The wall superheat, or $\Delta T_{\text{sat}}$, is defined as the difference between the heater surface temperature, and the liquid’s saturation temperature at the system pressure. All testing was performed at atmospheric pressure, giving an ethanol saturation temperature of 78.3°C.

### 7.1 Plain Chip

The plain chip was tested in order to establish a performance baseline of ethanol in open microchannels and tapered gap manifolds. The effect of manifold taper on the plain chip is shown in Fig. 25(a), which shows a performance improvement from the 2% to the 6% manifold taper. The same trend is visible in Fig. 25(b), showing an increase in heat transfer coefficient with higher manifold tapers. Figure 25(c) shows pressure drop results as a function of heat flux for the plain chip. The plot indicates that pressure drop increases gradually with heat flux. A noticeable reduction in pressure drop was seen for the 6% taper.

The best performance of the plain chip was obtained at 80 mL/min with the 6% taper, having a maximum heat flux of 194 W/cm² at a wall superheat of 47°C, and a maximum heat transfer coefficient of 43,700 W/m²K at an exit vapor quality of 0.12. The maximum pressure drop measured for these conditions at CHF was 8.0 kPa.
Figure 25: Plots showing effect of manifold taper on plain chip at 60 mL/min: (a) boiling curve, (b) heat transfer coefficient as a function of vapor quality, (c) pressure drop as a function of heat flux.
7.2 Microchannel Chips

Two microchannel chips, referred to as MC1 and MC2, were tested. They feature rectangular microchannels with dimensions shown in Table 3. The microchannel chips showed significant heat transfer enhancement compared to the plain chip, as well as a reduced pressure drop.

The effect of manifold taper for the microchannel chips is shown in Fig. 26(a), which shows the boiling curve for all three tapers at a flow rate of 60 mL/min. Here the enhancement for the 6% taper is more pronounced, as the heat flux is significantly higher throughout than for the other two tapers. This trend is also seen in Fig. 26(b), showing an increase in heat transfer coefficient for increasing manifold taper, with the 6% taper producing significantly higher heat transfer coefficients for comparable vapor qualities.

Pressure drop results for the microchannel chips are shown in Fig. 26(c). Pressure drop increases gradually with heat flux, and increases rapidly as CHF is reached. Again, a noticeable reduction in pressure drop was observed with the 6% taper, compared to the 2% and 4% tapers.

The best performance for the microchannel chips was obtained with the 6% taper at a flow rate of 80 mL/min. This configuration allowed chip MC1 to reach a heat flux of 217 W/cm² at a wall superheat of 34°C, and a maximum heat transfer coefficient of 66,900 W/m²K at an exit vapor quality of 0.18. In the same configuration, MC2 reached a heat flux of 185 W/cm² at a wall superheat of 39°C, and a maximum heat transfer coefficient of 51,500 W/m²K at an exit vapor quality of 0.13. The maximum pressure drops measured for these conditions at CHF were 8.8 kPa and 5.8 kPa for chips MC1 and MC2, respectively. A summary of the performance of the plain and microchannel chips is shown in Table 5.
Figure 26: Plots showing effect of manifold taper for microchannel chips at 60 mL/min: (a) boiling curve, (b) heat transfer coefficient as function of vapor quality, (c) pressure drop as function of heat flux
Table 5: Plain and microchannel chip performance at CHF and peak heat transfer coefficient values

<table>
<thead>
<tr>
<th>Chip</th>
<th>CHF</th>
<th>$\Delta T_{\text{sat,CHF}}$</th>
<th>$h_{\text{max}}$</th>
<th>$\chi_{h_{\text{max}}}$</th>
<th>$\Delta P_{\text{CHF}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>194</td>
<td>47</td>
<td>43,700</td>
<td>0.12</td>
<td>8.0</td>
</tr>
<tr>
<td>MC1</td>
<td>217</td>
<td>34</td>
<td>66,900</td>
<td>0.18</td>
<td>8.8</td>
</tr>
<tr>
<td>MC2</td>
<td>185</td>
<td>39</td>
<td>51,600</td>
<td>0.13</td>
<td>5.8</td>
</tr>
</tbody>
</table>
8. DISCUSSION

A comprehensive discussion of the results in Section 7 is presented and the results of the conducted parametric studies, i.e. effect of flow rate, manifold taper, and channel width are evaluated.

8.1 Effect of Flow Rate

The effect of flow rate on the performance of the plain and microchannel chips is shown in Fig. 27 (a) and (b). For all chips, a performance improvement is observed with increasing flow rates of 40, 60 and 80 mL/min. The results clearly show the expected trend of increasing heat flux with flow rate. These results are consistent across all manifold tapers. Plots for the 6% manifold taper are shown in Fig. 27. Plots showing the effect of flow rate for the remaining tapers are shown in Appendix C. A significant increase in performance was observed between 40 and 60 mL/min, while the increase from 60 to 80 mL/min was comparatively small. The results indicate that both nucleate boiling and convective boiling contributions are present in this configuration. In designing a gravity-driven system such as a thermosyphon loop, low pressure drop is a stringent requirement in order to dissipate high heat fluxes. Although a low flow rate system is desirable in terms of pressure drop, higher flow rates increase the CHF limit.

![Figure 27: Plots showing effect of flow rate on (a) plain and (b) microchannel chips with 6% manifold taper](image-url)
8.2 Effect of Manifold Taper

The results for the plain chip in Figs. 25(a) and 25(b) show a small degree of improvement in heat transfer performance with increasing taper, while the performance improvement is much more pronounced for the microchannel chips. Figures 26(a) and 26(b) show that for both microchannel chips MC1 and MC2 the 6% taper performed significantly better than the 2% and 4% tapers, achieving higher heat fluxes at lower surface temperatures. Figs. 25(b) and 26(b) indicate a proportional relationship between heat transfer coefficient and manifold taper. This suggests that increasing the taper height directly influences the heat transfer performance and delays CHF, thus validating the proposed mechanism of the open microchannels and tapered manifolds: Bubbles are no longer restricted to grow horizontally in closed channels, but instead allowed to grow vertically into the expanding manifold space. The flow area change provided by the manifold results in a pressure drop recovery, significantly reducing the pressure drop across the channels. The result is improved flow stability and enhanced heat transfer performance. Liquid supply to the channels is improved, and the occurrence of upstream growing vapor slugs is eliminated. As a result of the increased pressure drop recovery and improved flow stability, the 6% taper delivered the highest heat transfer performance, as seen in Figs. 25(b), and 26(b), at the lowest pressure drop, shown in Figs. 25(c), 26(c).

8.3 Comparison of the Plain and Microchannel Chips

A direct comparison of the performance of the three chips is shown in Fig. 28. Figure 28(a) shows the microchannel chips perform better than the plain chip, achieving higher heat fluxes at lower wall superheats. Chip MC1 performed particularly well, achieving a much higher degree of heat transfer enhancement. Chip MC2 also showed a significant performance improvement over the plain chip, though it fell short of the performance of chip MC1, which in comparison yielded significantly higher heat transfer coefficients as shown in Fig. 28(b). From data analysis it was observed that chip MC2 often exhibited a higher degree of temperature fluctuation at high heat fluxes. From high-speed video analysis, it was observed that this fluctuation in the surface temperature of MC2 was due to intermittent dry-out at high heat fluxes.
Microchannels are attractive for the following reasons: (i) increased surface area, (ii) reduced flow instability, and (iii) improved CHF and HTC. The effect of microchannel width was studied in chips MC1 and MC2, having channel widths of 200 µm and 400 µm, respectively. As shown in Fig. 28, chip MC1 performed better in terms of heat flux and heat transfer coefficient. The narrower channel width of chip MC1 in comparison to MC2 provides a higher degree of area enhancement, thus delivering improved heat transfer performance. Since the pressure drop is nearly identical in the two cases as seen from Fig. 28 (c), MC1 is preferable with its higher CHF and HTC.

8.4 Pressure Drop Performance

Figure 28(c) provides a direct comparison of the pressure drop data of the plain and microchannel chips. The microchannel chips exhibited reduced pressure drop throughout in comparison to the plain chip. A sharp increase in pressure drop was observed for all chips at CHF, due to intermittent heater dry-out caused by recurrent vapor blanketing. However, the magnitude of the pressure drop was still quite low overall. The highest heat flux was obtained with chip MC1, which reached a maximum pressure drop of 8.8 kPa at CHF.

Figure 26(c) shows the pressure drop performance of the microchannel chips MC1 and MC2. It is evident that there is a significant dependence of pressure drop on the manifold taper. The pressure drop for the 6% taper was significantly lower than for the remaining tapers, showing the effect of pressure drop recovery caused by the expanding manifold cross-sectional flow area. The pressure recovery provided by the taper allows for high heat transfer at a very low pressure drop. In the current configuration of the experimental setup, a gravitational head of 2.68 m was provided, allowing the system to compensate for a pressure drop of up to 20 kPa at a flow rate of 80 mL/min. The actual pressure drop, even at CHF, for all tested configurations remained at much less than 50% of this value. At a heat flux of 100 W/cm², an ethanol head of only 0.38 m is sufficient to drive the system at a flow rate of 80 mL/min. At lower flow rates, even less ethanol head would be required.
Figure 28: Plots showing effect of chip at 80 mL/min with 6% manifold taper: (a) boiling curve, (b) heat transfer coefficient as a function of vapor quality, (c) pressure drop as a function of heat flux.
8.5 Flow Visualization

Figure 29 shows frames of high-speed video taken with a Photron FASTCAM 1024PCI at 6000 fps. The frames show growing and coalescing bubbles on the microchannel chip MC1, tested at a flow rate of 60 mL/min. The frames were taken at a heat flux of 63.7 W/cm² and a wall superheat of 23.3°C. The frames show a bubble, circled with a solid line, traveling along the path provided by one of the microchannels. In frame (c), the bubble approaches a group of neighboring bubbles travelling along the paths provided by neighboring microchannels, and in frame (d) these bubbles begin to coalesce as they continue to travel downstream. Two larger bubbles are formed, and are visible in the solid circle in frame (e). The top bubble marked by the solid circle in frame (e) has travelled across a channel fin and onto the neighboring channel to its right.

**Figure 29:** High-speed video frames of growing and coalescing bubbles in chip MC1. Test conditions: 2% manifold taper, 60 mL/min flow rate, heat flux 63.7 W/cm². Frames: (a) 0.00 ms, (b) 3.50 ms, (c) 6.70 ms, (d) 8.00 ms, (e) 10.00 ms, (f) 12.70 ms.
The frames in Fig. 29 demonstrate the mechanism of heat transfer enhancement in open microchannels with a tapered gap manifold. Once bubble nucleation occurs, the bubbles grow upward into the space provided by the manifold taper and away from the surface. This is evidenced by the rolling of the bubbles in the solid circle shown in frames (d) – (f), and by the growth of the bubbles beyond the width of the channel walls. The bubbles marked in the dotted and dashed circles in frames (d) – (f) adhere to the channel walls as they flow downstream, providing a structured flow path, and expand beyond the channel walls as they continue to grow. This shows the bubbles are effectively sitting atop the channels in the space provided by the tapered gap manifold. The upward bubble growth into the manifold space allows for improved liquid supply, as liquid is allowed to fill the bottom of the channels as evaporation takes places and the bubbles continue to grow. The expansion of the cross-sectional flow are virtually eliminates downstream flow restriction along the channels, resulting in pressure recovery across the channel length, and improved vapor removal from the surface.

Another major advantage of this geometry is that large vapor slugs occurring at higher heat fluxes do not hinder heat transfer, as the slugs will sit atop the channel fins, and bubble nucleation is allowed to continue within the active cavities at the bottoms of the channels.

8.6 Flow Instability & Pressure Drop Fluctuation

Analysis of high-speed video throughout testing revealed that there was some flow instability at the microchannel test section. While the bulk flow of fluid was observed to flow in the downstream direction, sporadic reductions in velocity and in some cases even flow reversals were observed for short time intervals, before the flow once again flowed downstream, converging asymptotically to the initial flow rate value. These flow instabilities manifested as fluctuations in flow rate and pressure drop. Fig. 30 below shows the time-variation of pressure drop across the microchannels for chip MC1 with the 2% taper manifold. The fluctuations shown were recorded at a flow rate of 60 mL/min and a heat flux of approximately 48 W/cm². The fluctuations at this heat flux in this particular configuration were quite significant, and some slight flow
reversals were observed. The reversals produced spikes of negative pressure drop, which can be seen in Fig. 30.

![Figure 30: Pressure drop fluctuation – MC1 with 2% taper at a flow rate of 60 mL/min](image)

These fluctuations are caused by Ledinegg instability, as well as upstream compressible volume instability. The upstream compressible volume instability refers to the instability and/or flow reversal caused by upstream gas bubbles entering the microchannels. These gas bubbles are caused by non-condensable gases contained in the liquid. As mentioned in the degassing section, the oxygen solubility of ethanol is quite high, with a weak temperature dependence. In the case of the data collected in Fig. 30, it is clear that the liquid was in an insufficiently degassed state.

Ledinegg instability, or flow excursion instability, often manifests in systems where the design constraints are such that, in a plot of pressure drop, ΔP as a function of mass flux, G, shown in Fig. 31, within the two-phase flow region, the gradient of the pump supply (curve A in Fig. 31) is greater (has a smaller negative value) than that of the system demand curve [26], as shown in Eqn. (11).

\[
\left. \frac{\partial (\Delta P)}{\partial G} \right|_D \leq \left. \frac{\partial (\Delta P)}{\partial G} \right|_S
\]  

(11)
In these conditions, the system is unable to respond effectively to even minor changes in mass flux from the equilibrium condition, at point a in Fig. 31, causing the flow to spontaneously shift to a stable flow condition, such as points b and c in Fig. 31, where the gradient discrepancy in Eqn. (10) is reversed, i.e. the pressure-mass flux gradient of the pump supply curve is less than that of the system demand curve.

Figure 31: $\Delta P - G$ curve for a channel with uniform heating

Ledinegg instability is characteristic of systems where flow is supplied between two constant pressure reservoirs, as is the case of the present work. In this case both reservoirs are at atmospheric pressure, while the gravitational head from the elevated reservoir induces a higher pressure at the test section inlet. This characteristic of constant-pressure reservoir systems was discovered in early flow boiling studies on microchannels, and was mitigated in later works thanks to the implementation of positive displacement pumps to drive the flow [26]. The $\Delta P$–$G$ slope of positive displacement pumps is inherently quite steep, with minimal changes in mass flux, regardless of pressure drop. This would result in a supply curve slope (line A in Fig. 31) much steeper than that of the demand curve at point a in Fig. 31, giving rise to a stable system.
Due to the nature of the gravity-driven flow configuration used in this work, which requires two reservoirs by design, Ledinegg instability cannot be avoided. It can however be mitigated by introducing a high pressure drop upstream of the channel inlet. This is readily achieved with an inlet restrictor, as has been discussed previously. In the present work, an inlet restrictor was built into the manifold block design. As can be seen in Fig. 30, instability was still present in the system, causing the chip surface to reach higher wall superheats, and leading to lower CHF values. Nevertheless, exceptional performance with ethanol was achieved, as shown in Figs. 25 – 28.

8.7 Pressure Drop Modelling

A two-phase pressure drop model was developed by Kalani and Kandlikar [27] for the open microchannel geometry with tapered gap manifolds. They utilized the homogeneous flow model as the basis of their model. The homogeneous flow model assumes a homogeneous mixture of liquid and vapor phase within the channel region, incorporating two-phase frictional and fluid acceleration effects. This model constitutes a valid approach for the open microchannel geometry, given that it typically exhibits a high degree of flow mixing of the liquid and vapor phases, due to the upward bubble growth that is allowed by the expanding manifold gap. Aside from incorporating a single phase pressure drop region into their model, accounting for the subcooled inlet region, Kalani and Kandlikar [27] accounted for the effect of the manifold taper by introducing an area change term.

The model considers various components contributing to the total two-phase pressure drop. First, the single phase components are described.

8.7.1 Single-phase inlet region:

The inlet region of the microchannels, where the circular channel inlet transitions to a rectangular duct, is responsible for a frictional pressure drop component. The frictional pressure drop for both the inlet pipe and duct were calculated using Eqn. (12).
\[ \Delta P_{\text{inlet}} = \frac{2f \rho u_m^2 L}{D} \]  

(12)

where \( u_m \) in Eqn. (12) is the mean fluid velocity, which is calculated with Eqn. (13), and \( f \) is the fanning friction factor, shown in Eqn. (14) which is determined with the Poiseuille number and the Reynolds number.

\[ u_m = \frac{\dot{m}}{\rho A_c} = \frac{\dot{V}}{\rho A_c} \]  

(13)

\[ Po = f Re \]  

(14)

For the circular pipe inlet section, the Poiseuille number is taken as 16, whereas for the rectangular duct section, the Poiseuille is defined via the channel aspect ratio \( \alpha_c \), as shown in Eqn. (15).

\[ Po = 24(1 - 1.3553\alpha_c + 1.9467\alpha_c^2 - 1.7012\alpha_c^3 + 0.9564\alpha_c^4 - 0.2537\alpha_c^5) \]  

(15)

8.7.2 Single-phase channel region:

The channel region refers to the flow region between the inlet and outlet ducts, delimited by the plain chip surface or the microchannels, and the tapered manifold. Again, a frictional pressure drop component was considered, now also taking into account the entrance losses due to the area change at the inlet, as described in Eqn. (16).

\[ \Delta P_{sp} = \frac{\rho u_m^2}{2} \left( K_i + \frac{4f_{app}L_{sp}}{D_h} \right) \]  

(16)

In Eqn. (16), \( f_{app} \) is the apparent fanning friction factor, which accounts for the pressure drop due to friction, as well as developing flow region effects due to the channel area change. The entrance loss coefficient \( K_i \) was determined from graphs by Kays and London [28]. \( L_{sp} \) is the single phase channel length due to the degree of liquid subcooling, past which the fluid becomes saturated and two-phase effects occur. The single phase channel length was calculated using Eqn. (17).
\[ L_{sp} = \frac{m c_p \Delta T_{sub}}{q'' W} \]  

(17)

where \( W \) is the chip width of 10 mm, and \( \Delta T_{sub} \) is the degree of liquid subcooling.

The Poiseuille number, using a non-dimensional length \( x^+ \), defined in Eqn. (18), and together with the aspect ratio, are used to determine the apparent fanning friction factor \( f_{app} \) in Eqn. (16) via a look-up table derived by Phillips [29].

\[ x^+ = \frac{L_{sp} / D_h}{Re} \]  

(18)

Now, the two-phase components are considered.

### 8.7.3 Two-phase region:

The two-phase pressure drop consists of three key components: (i) frictional, (ii) accelerational, and (iii) gravitational pressure drop components. In the present work, the gravitational term was zero due to the horizontal flow configuration. Kalani and Kandlikar [27] used the homogeneous model [30] for the two-phase pressure drop calculations.

The frictional and accelerational terms are grouped as follows in Eqn. (19).

\[-\left(\frac{dP}{dz}\right) = \frac{2 f_{tp} G^2 v_f}{D_h} \left[ 1 + x \left(\frac{v_g}{v_f}\right) \right] + G^2 v_{fg} \frac{dx}{dz} \left(1 + G^2 x \left(\frac{d v_g}{dp}\right)\right)\]  

(19)

where \( x \) is the exit vapor quality, \( v_f \) is the specific volume of the liquid, \( v_{fg} \) is the difference in specific volume of the saturated liquid and vapor, \( G \) is the mass flux, and \( f_{tp} \) is the two-phase friction factor.

The exit quality was calculated by taking into account the degree of liquid subcooling at the inlet, as shown in Eqn. (20).

\[ x = \frac{1}{h_{fg}} \left[ \left(\frac{q'' A}{\dot{m}}\right) - c_p \Delta T_{sub} \right] \]  

(20)
where \( h_{fg} \) is the latent heat of vaporization, \( A \) is the projected area of the heater base, and \( c_p \) is the specific heat capacity.

Kalani and Kandlikar [27] added one further term affecting the two-phase pressure drop, which is due to the area change arising from the manifold taper. The area pressure gradient term was defined as follows:

\[
-\left( \frac{dP}{dz_{taper,area}} \right) = \frac{-2G^2\nu_f}{A_c} \left[ 1 + x \left( \frac{\nu_g}{\nu_f} \right) \right] \frac{dA}{dz} + \frac{G^2}{1 + G^2} x \left( \frac{dv_g}{dp} \right)
\]

(21)

where \( A_c \) is the cross-sectional area and \( \frac{dA}{dz} \) represents the change in cross-sectional area along the channel length of the two-phase region. The low quality change over the length of the channels observed by Kalani and Kandlikar [27], as well as in the present work, allowed for the use of an average quality and average mass flux at the center of the two-phase flow region. Kalani and Kandlikar [27] used several viscosity averaging models to determine the two-phase friction factor. The present work used Owens [31] viscosity model, in which the two-phase viscosity is simply approximated by the liquid viscosity value.

8.7.4 Two-phase exit region:

The model by Kalani and Kandlikar [27] also considers the exit region losses, accounting for both the frictional pressure drop and the exit region effects, as shown in Eqn. (22).

\[
\Delta P_{exit} = \frac{\rho u_m^2}{2} \left( K_e + \frac{4f_{tp}L}{D_h} \right)
\]

(22)

Kalani and Kandlikar [27] describe in detail the analysis of the three two-phase pressure drop components in a horizontal flow boiling system with a tapered manifold, which are (i) friction, (ii) acceleration, and (iii) area change due to taper. The area change term arising from the manifold taper is crucial in regards to the pressure drop reduction seen in tapered manifolds, as it results in a pressure drop recovery.
Figure 32(a) shows a comparison between the experimental pressure drop data, obtained in the present work, and the prediction of the model by Kalani and Kandlikar [27]. The data in the figure was taken at a flow rate of 80 mL/min, with the 4% manifold taper. The error is higher at lower heat fluxes due to the low value of pressure drop in itself, and the channeling effect caused by isolated nucleation that introduce deviations from the homogeneous flow model.

![Figure 32: (a) Comparison of pressure drop model [27] and experimental data, and (b) Two-phase pressure drop components from model [27], for MC1 with 4% manifold taper at 80 mL/min](image)

Figure 32(b) shows the three two-phase pressure drop components for the same flow conditions shown in Fig. 32(a). Note that in addition to these three components, the total pressure takes into account single-phase frictional pressure drop and inlet and exit losses, as described previously. These terms are constants for any given flow with respect to heat flux, and are not of critical importance in this analysis. The pressure drop contributions from the friction and acceleration terms increase with increasing heat flux, resulting in higher pressure drop at higher heat fluxes. Meanwhile, the area change term results in a pressure drop recovery, as evidenced by the negative pressure drop values it displays in Fig. 32(b). The pressure drop recovery further increases at high heat fluxes.
The model predicted the pressure drop to within 30% MAE for the 4% manifold taper, and showed fairly consistent agreement for the remaining tapers and flow rates, with an MAE of less than 50% on average. A rapid spike in pressure drop is seen in the experimental data at CHF, due to intermittent vapor blanketing. Pressure drop fluctuation due to Ledinegg instability accounts for the magnitude of the error bars displayed in Fig. 32(a).

Kalani and Kandlikar [27] noted that an increase in manifold taper provides further increases in pressure drop recovery at higher heat fluxes. The same findings were obtained in this work, as per the results discussed in Section 8.4. This is especially advantageous in the case of gravity-driven systems as the reduction in pressure drop provided by the manifold taper at higher heat fluxes allows for an increase in flow rate through the system. This in turn allows for improved heat transfer performance, as seen from Fig. 27. It is therefore observed that the OMM configuration with ethanol is well-suited for gravity-driven systems, through this self-modulating behavior. The pressure recovery provided by the taper allows for high heat transfer at a remarkably low pressure drop. Given the pressure drop results as a function of heat flux, it is seen that at a heat flux of 100 W/cm², an ethanol head of only 0.33 m is sufficient to drive the system at a flow rate of 80 mL/min. At lower flow rates, even less ethanol head would be required.

8.8 Parametric Study – Heat Transfer Performance as a Function of Available Fluid Head

Thanks to the remarkably low pressure drop of the open microchannel and tapered manifold configuration, a gravity-driven flow system was successfully implemented for the completion of this work. In order to determine whether this configuration could be implemented in a small-scale cooling system for high-heat flux applications where pumping power cost is a concern, a parametric study was conducted, expressing heat transfer performance as a function of ethanol head. This study was conducted with the 6% taper, as this taper resulted in the lowest pressure drop. Figures 33(a) and 33(b) plot ethanol head as a function of heat flux, for the 6% taper at flow rates of 40 mL/min and 60 mL/min, respectively.
Figure 33: Required ethanol head for a given heat flux, based on experimental data of all three chips with the 6% manifold taper, measured at (a) 40 mL/min, and (b) 60 mL/min

Figure 33 expresses the ethanol head required in order to achieve the desired heat flux at the given flow rate. The measured pressure drop across the chip at a given heat flux was related to ethanol head via Eqn. (23), the expression for hydrostatic pressure for a column of liquid. The equation is rearranged to solve for the height as shown in Eqn. (24) below.

\[ \Delta P = \rho g \Delta H \]  

\[ \Delta H = \frac{\Delta P}{\rho g} \]  

The effect of the microchannels is shown in Fig. 33, with chip MC1 once again performing better than the plain chip or chip MC2. Figure 33 shows a critical component of the design considerations for a small-scale gravity-driven flow-boiling cooling application, which is the channel geometry. The narrower channel width of MC1 provides higher area enhancement, resulting in higher heat removal. This allows the chip to achieve higher heat fluxes, at comparable, or lower wall superheats, ultimately resulting in significant increases in heat transfer coefficient.
Figure 34: Ethanol head as a function of heat flux with the 6% manifold taper, measured at 40 mL/min, 60 mL/min, and 80 mL/min, for the (a) plain chip, (b) MC1, and (c) MC2.
The effect of flow rate was studied directly in Fig. 3 to further determine optimal design parameter ranges for the geometry of the present work. For all three plots of Fig. 3, it is clear that there is an increase in heat flux with flow rate, which is accompanied by a further increase in ethanol head. As the vital concern of the application, being driven only by gravity, is the pressure drop, it is conceivable that for some applications a compromise in cooling performance must be made to reduce the pressure drop. A determination can be made as to whether it is more favorable to run the system at a lower flow rate, or if the increased cooling performance is a priority.

The trendlines shown in Figs. 33 and 34 serve as a means of estimating performance at any desired operating point. Table 6 shows the trendline equations of the curves in Figs. 33 and 34. The equations serve as guidelines for the design of a small-scale cooling device, capable of dissipating high heat fluxes at low pressure drop without the need for a pump, in a flow driven only by gravity.

Table 6: Trendline equations for ethanol head as a function of heat flux

<table>
<thead>
<tr>
<th>Flow Rate</th>
<th>Plain Chip</th>
<th>Chip MC1</th>
<th>Chip MC2</th>
</tr>
</thead>
<tbody>
<tr>
<td>40 mL/min</td>
<td>$\Delta H = 4.40E-7q^2$</td>
<td>$\Delta H = 3.65E-6q^2$</td>
<td>$\Delta H = 8.29E-6q^2$</td>
</tr>
<tr>
<td></td>
<td>+ 0.00380057q''</td>
<td>+ 0.00217893q''</td>
<td>+ 0.00216134q''</td>
</tr>
<tr>
<td></td>
<td>+ 0.07101813 (25)</td>
<td>+ 0.03467295 (26)</td>
<td>+ 0.04739236 (27)</td>
</tr>
<tr>
<td>60 mL/min</td>
<td>$\Delta H = 5.66E-6q^2$</td>
<td>$\Delta H = 2.294E-5q^2$</td>
<td>$\Delta H = 1.945E-5q^2$</td>
</tr>
<tr>
<td></td>
<td>+ 0.00253946q''</td>
<td>- 0.00183754q''</td>
<td>- 0.00029567q''</td>
</tr>
<tr>
<td></td>
<td>+ 0.17211963 (28)</td>
<td>0.20000000 (29)</td>
<td>+ 0.26174756 (30)</td>
</tr>
<tr>
<td>80 mL/min</td>
<td>$\Delta H = 1.890E-6q^2$</td>
<td>$\Delta H = 1.632E-5q^2$</td>
<td>$\Delta H = 1.094E-5q^2$</td>
</tr>
<tr>
<td></td>
<td>+ 0.00435q''</td>
<td>+ 0.000115q''</td>
<td>+ 0.0021476q''</td>
</tr>
<tr>
<td></td>
<td>+ 0.1477 (31)</td>
<td>0.211346 (32)</td>
<td>+ 0.15532811 (33)</td>
</tr>
</tbody>
</table>

Given an application that dissipates a heat flux of 100 W/cm², that were to use flow boiling in open microchannels, with a tapered manifold, and microchannel chip MC1 is chosen to cool the device, then with an ethanol flow rate of 60 mL/min and according to Eqn. (29), which closely matches with the
experimental data in Fig. 32(b), an ethanol head of only 0.33 m would be sufficient to drive the flow of the open microchannel heat sink.

9. CONCLUSIONS

An experimental flow boiling study was conducted to determine the performance of ethanol in open microchannels with tapered manifolds. The study aimed to determine design guidelines to develop a cooling apparatus using gravity to drive the fluid flow. The experimental results led to the following conclusions:

(a) The open microchannel chips delivered excellent performance using ethanol, reaching a maximum heat flux of 217 W/cm², while keeping the chip surface at a wall superheat of 34°C. This represents a performance enhancement of over 340% compared to previous results in flow boiling with ethanol, and a 92% improvement over previous works in pool boiling with ethanol. The results confirm the potential of the open microchannel and tapered manifold geometry of dissipating extremely high heat fluxes, as was found by Kandlikar et al [5], and it can be expected that similar levels of enhancement may be obtained with other fluids. Dielectric fluids such as FC-72 and HFE-7000 may be used if flammability is a concern.

(b) Area enhancement and improved liquid supply paths provided by the microchannels resulted in improved heat transfer due to nucleate boiling, allowing for higher heat fluxes to be reached before the onset of CHF. Higher area enhancement of the narrower channel width of MC1 resulted in improved performance over MC2, yielding higher heat transfer coefficients, thus allowing for higher heat fluxes at lower wall superheats. A maximum heat transfer coefficient of 66,900 W/m²K was obtained with chip MC1.

(c) Higher flow rates resulted in higher heat fluxes and heat transfer coefficients, indicating that both nucleate boiling and convective boiling effects are present in this configuration. Even at low flow rates of 40 mL/min, heat fluxes of 160 W/cm² could be achieved with chip MC1. Low flow rates are an essential consideration for compact gravity-driven flow systems, as low pressure drop is a stringent requirement for such systems.
The effect of manifold taper was studied. Higher tapers provided higher pressure recovery, resulting in reduced pressure drop. The 6% manifold taper delivered the lowest pressure drop. A maximum pressure drop of 8.8 kPa was recorded at CHF. The remarkably low pressure drop of these channels enabled the use of a gravity-driven flow configuration, eliminating the need for a pump. The use of even higher tapers is a promising pathway for further heat transfer enhancement in future works.

Pressure drop data for ethanol was compared to the prediction of the two-phase pressure drop model developed by Kalani and Kandlikar [27]. The model accurately predicted the pressure drop behavior for some manifold tapers. Future work may further investigate the approach of the model to enhance its applicability to different manifold tapers.

The pressure recovery due to the tapered manifold provides a self-modulating effect in a gravity-driven cooling system. The flow rate can increase at higher heat fluxes due to this pressure recovery, and the heat transfer performance would be improved accordingly. The pressure recovery provided by the taper allows for high heat transfer possible at a very low pressure drop. At a heat flux of 100 W/cm², an ethanol head of only 0.33 m is sufficient to drive the system at a flow rate of 60 mL/min.

The OMM configuration provides design opportunities to dissipate a range of desired heat fluxes in a relatively simple, compact configuration without the need for an external pump.
10. RECOMMENDATIONS & FUTURE WORK

10.1 Manifold Sealing Issues

During assembly of the test section, a significant sealing issue became apparent. In the current design, there is a single 127 µm compressible silicone gasket, which seals the entire chip surface from the surrounding test fixture. The system was deemed satisfactorily sealed if a significant pressure (> 5 psi, or 34.4 kPa) could be maintained, with minimal loss of pressure over a time period of 1 min. Repeated attempts to seal the system eventually resulted in a satisfactory level of sealing in most cases. In some cases, the gaskets had to be replaced.

In future works, sealing could be facilitated by using compressible gaskets of a higher thickness, allowing for a higher level of interference when the manifold is tightened down onto the chip surface.

10.2 Flow Instability

Significant flow instability was observed during testing. This form of Ledinegg instability is unavoidable in the current configuration due to the use of two constant (atmospheric) pressure reservoirs. It is well known that the use of inlet restriction will result in improved flow stability. An inlet restriction valve was in place in the experimental apparatus of this work, although the valve was likely placed too far upstream from the test section inlet to further reduce the oscillations.

In future works, an inlet restriction valve could be used in immediate vicinity of the test section inlet, although this will result in a higher pressure drop. In a gravity-driven flow configuration, the relative benefit of increased flow stability versus lower pressure drop must be evaluated in order to optimize system performance.

10.3 Applications of Low Pressure Drop, Gravity-Driven Flow Boiling Systems

The present work explored the feasibility of a flow boiling system with a gravity-driven flow configuration and ethanol as the working fluid, with a low-pressure-drop microchannel geometry. The results of the present work demonstrate the effectiveness of the system, which was capable of reaching high
heat fluxes at low flow rates, with remarkably low pressure drop. Additionally, a parametric study was presented relating the desired heat transfer performance to the required ethanol head, which serves as a basic design guideline relating system performance to the scale of the system. It was proven that for heat fluxes on the order of 100 W/cm² an ethanol head of only 0.33 m would be sufficient to drive the system at a flow rate of 60 mL/min. Even lower heights are feasible at lower flow rates.

Future work could explore the design of a small-scale system, similar to a thermosyphon, in which flow is driven entirely by gravity, and low-flow rate forced-convection boiling is used to cool the heated surface in an open microchannel configuration with a tapered manifold.

Regarding the microchannel geometry specifically, different channel geometries can be explored. Higher area enhancement provided in MC1 in the results of the present work showed a significant enhancement of heat transfer performance compared to the plain chip, and it is thus expected that different channel geometries may result in similar, and significant changes in performance.

The effect of higher tapers could also be investigated, as the system performance also showed significant sensitivity to increases in manifold taper. It is likely that there will be a limiting condition to the enhancement provided by the taper, past which flow separation and other adverse effects may occur.


11. REFERENCES


APPENDIX

Appendix A: Elevated Reservoir Assembly

In order to elevate reservoir #1 to the required height, a structure was built consisting of T-slotted aluminum framing. This structure was attached to a frame assembly around the vibration isolation table used for testing. The elevated reservoir assembly is made of the same T-slotted framing as the existing aluminum framing structure.

A 6ft long, four-slot, hollow, metric, T-slotted aluminum extrusion was used, shown in Fig. 36. This extrusion was rigidly fixed to the side of one of the upright t-slotted beams of the frame structure via three aluminum plates, held together via several standard fasteners.

In order to support the reservoir, an aluminum plate was vertically fastened to the L-shaped aluminum brackets shown in Fig. 36(b). The aluminum plate was used as a mounting surface for two clamps and one J-hook, which are used to support the elevated reservoir by its handle.

Figure 35: 3D drawings of elevated reservoir assembly
Figure 36: Images of (a) elevated reservoir support plate, and (b) detail view of clamps and J-hook
Appendix B: Error Propagation – Derivation of Uncertainties

The derivation of the uncertainty equations shown in Section 6, of all key calculated parameters is shown below.

B.1 Heat Flux Uncertainty Derivation

The relation for heat flux, measured from the three heater block thermocouples, is computed from Eqns. (1) and (2) in Section 4, and shown below in Eqn. (34)

\[ q'' = -k_{Cu} \left( \frac{3T_1 - 4T_2 + T_3}{2\Delta d} \right) \]  

(34)

The variables in Eqn. (34) are substituted into the error propagation equation (Eqn. 6), as shown in Eqn. (35) and Eqn. (36) below.

\[ \frac{U_{q''}}{q''} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial q''}{\partial \sigma_i} U_{\sigma_i} \right)^2} \]  

(35)

\[ \therefore \frac{U_{q''}}{q''} = \sqrt{\left( \frac{\partial q''}{\partial k_{Cu}} U_{k_{Cu}} \right)^2 + \left( \frac{\partial q''}{\partial \Delta d} U_{\Delta d} \right)^2 + \left( \frac{\partial q''}{\partial T_1} U_{T_1} \right)^2 + \left( \frac{\partial q''}{\partial T_2} U_{T_2} \right)^2 + \left( \frac{\partial q''}{\partial T_3} U_{T_3} \right)^2} \]  

(36)

It will be useful for the derivation to define a variable \( \alpha \) defined as shown in Eqn. (37).

\[ \alpha = 3T_1 - 4T_2 + T_3 \]  

(37)

This will simplify the derivations that follow. The sensitivity coefficients for each variable, which are expressed as partial derivatives of the equation (function \( p \) in Eqn. 6) with respect to the individual variables of interest (variables \( \sigma_i \) in Eqn. 6), are evaluated individually. The expressions are then rewritten in terms of the function of interest, \( q'' \), as shown in Eqns. (38-42).

\[ \frac{\partial q''}{\partial k_{Cu}} = -\frac{\alpha}{2\Delta d} \frac{q''}{k_{Cu}} \]  

(38)
\[
\frac{\partial q''}{\partial \Delta x} = -k_{cu} \frac{\alpha}{2\Delta d^2} = -\frac{q''}{\Delta d}
\] (39)

\[
\frac{\partial q''}{\partial T_1} = -k_{cu} \frac{3}{2\Delta d} = \frac{3q''}{\alpha}
\] (40)

\[
\frac{\partial q''}{\partial T_2} = -k_{cu} \frac{-4}{2\Delta d} = -\frac{4q''}{\alpha}
\] (41)

\[
\frac{\partial q''}{\partial T_3} = -\frac{k_{cu}}{2\Delta d} = \frac{q''}{\alpha}
\] (42)

The terms in Eqns. (38-42) are substituted back into Eqn. (36) to obtain the following.

\[
U_{q''} = \frac{q''}{q''} = \sqrt{\left(\frac{q''}{k_{cu} U_{k_{cu}}} \right)^2 + \left( -\frac{q''}{\Delta d} U_{\Delta d} \right)^2 + \left( \frac{3q''}{\alpha} U_{T_1} \right)^2 + \left( -\frac{4q''}{\alpha} U_{T_2} \right)^2 + \left( \frac{q''}{\alpha} U_{T_3} \right)^2}
\] (43)

The squared terms in each of the parentheses in the numerator of Eqn. (43) are expanded in Eqn. (44).

\[
U_{q''} = \sqrt{\frac{q''^2}{k_{cu}^2 U_{k_{cu}}} \Delta x^2 + \frac{q''^2}{\alpha^2} U_{\Delta d}^2 + \frac{9q''^2}{\alpha^2} U_{T_1}^2 + \frac{16q''^2}{\alpha^2} U_{T_2}^2 + \frac{q''^2}{\alpha^2} U_{T_3}^2}
\] (44)

This allows all \(q''^2\) terms in the numerator to cancel with the denominator, thus yielding the final expression in Eqn. (7), which is reproduced below in Eqn. (45).

\[
U_{q''} = \frac{U_{q''}}{q''} = \sqrt{\frac{U_{k_{cu}}^2}{k_{cu}^2} \Delta x^2 + \frac{U_{\Delta d}^2}{\Delta d^2} + \frac{9U_{T_1}^2}{\alpha^2} + \frac{16U_{T_2}^2}{\alpha^2} + \frac{U_{T_3}^2}{\alpha^2}}
\] (45)

**B.2 Chip Surface Temperature Uncertainty Derivation**

The relation for surface temperature is calculated from the heat flux over the projected area, taking into account the thermal resistance of copper, and the distance between the top thermocouple and the surface, as is shown in Eqn. (3) in Section 4. While the uncertainty in surface temperature is expected to be quite low, mathematical rigor dictates it is required in order to compute the uncertainty in heat transfer coefficient.
The variables in Eqn. (3) are substituted into the error propagation equation (Eqn. 6), as shown in Eqn. (46) and Eqn. (47) below.

\[
\frac{U_{T_s}}{T_s} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial T_s}{\partial \sigma_i} U_{\sigma_i} \right)^2}
\]  \hspace{1cm} (46)

\[
\therefore \frac{U_{T_s}}{T_s} = \sqrt{\left( \frac{\partial T_s}{\partial T_1} U_{T_1} \right)^2 + \left( \frac{\partial T_s}{\partial q''} U_{q''} \right)^2 + \left( \frac{\partial T_s}{\partial d_1} U_{d_1} \right)^2 + \left( \frac{\partial T_s}{\partial k_{Cu}} U_{k_{Cu}} \right)^2}
\]  \hspace{1cm} (47)

The sensitivity coefficients for each variable are evaluated individually below. The expressions are then rewritten in terms of the function of interest, \( T_s \), as shown in Eqns. (48-51).

\[
\frac{\partial T_s}{\partial T_1} = 1 - 0 = 1 = \frac{T_s}{T_s}
\]  \hspace{1cm} (48)

\[
\frac{\partial T_s}{\partial q''} = 0 - 1 \left( \frac{d_1}{k_{Cu}} \right) = - \left( \frac{d_1}{k_{Cu}} \right) \frac{T_s}{T_s}
\]  \hspace{1cm} (49)

\[
\frac{\partial T_s}{\partial d_1} = - \frac{q''}{k_{Cu}} = - \frac{q''}{k_{Cu}} \frac{T_s}{T_s}
\]  \hspace{1cm} (50)

\[
\frac{\partial T_s}{\partial k_{Cu}} = -q''d_1 = -q''d_1 \frac{T_s}{T_s}
\]  \hspace{1cm} (51)

The terms in Eqns. (48-51) are substituted back into Eqn. (47) to obtain the following expression.

\[
\frac{U_{T_s}}{T_s} = \sqrt{\left( \frac{T_s}{T_s} \cdot U_{T_1} \right)^2 + \left( - \frac{d_1}{k_{Cu}} \frac{T_s}{T_s} \cdot U_{q''} \right)^2 + \left( - \frac{q''}{k_{Cu}} \frac{T_s}{T_s} \cdot U_{d_1} \right)^2 + \left( -q''d_1 \frac{T_s}{T_s} \cdot U_{k_{Cu}} \right)^2}
\]  \hspace{1cm} (52)

The squared terms in each of the parentheses in the numerator of Eqn. (52) are expanded in Eqn. (53).
This allows all \( q''^2 \) terms in the numerator to cancel with the denominator, thus yielding the final expression shown in Eqn. (54).

\[
\frac{U_T}{T_s} = \sqrt{\frac{T_s^2 U_T^2}{T_s^2} + \frac{d_1^2 T_s^2 U_T^2}{k_{Cu}^2 T_s^2} + \frac{q''^2 T_s^2 U_T^2}{k_{Cu}^2 T_s^2} + \frac{q''^2 d_1^2 T_s^2 U_T^2}{k_{Cu}^2 T_s^2}}
\]  (53)

\[
\frac{U_T}{T_s} = \sqrt{\frac{U_T^2}{T_s^2} + \frac{U_q^2 d_1^2}{k_{Cu}^2 T_s^2} + \frac{U_{d_1}^2 q''^2}{k_{Cu}^2 T_s^2} + \frac{U_{k_{Cu}}^2 q''^2 d_1^2}{T_s^2}}
\]  (54)

**B.3 Heat Transfer Coefficient Uncertainty Derivation**

The relation for heat transfer coefficient is shown in Eqn. (4) in Section 4. The variables in Eqn. (4) are substituted into the error propagation equation (Eqn. (6) in Section 4), as shown in Eqn. (55) and (56) below.

\[
\frac{U_h}{h} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial h}{\partial \sigma_i} U_{\sigma_i} \right)^2}
\]  (55)

\[
\frac{U_h}{h} = \sqrt{\left( \frac{\partial h}{\partial q''} U_{q''} \right)^2 + \left( \frac{\partial h}{\partial T_s} U_{T_s} \right)^2 + \left( \frac{\partial h}{\partial T_{sat}} U_{T_{sat}} \right)^2}
\]  (56)

The sensitivity coefficients for each variable are evaluated individually. The expressions are then rewritten in terms of the function of interest, \( h \), as shown in Eqns. (57-59).

\[
\frac{\partial h}{\partial q''} = \frac{1}{T_s - T_{sat}} = \frac{h}{q''}
\]  (57)

\[
\frac{\partial h}{\partial T_s} = -\frac{q}{(T_s - T_{sat})^2} = -\frac{h}{(T_s - T_{sat})}
\]  (58)

\[
\frac{\partial h}{\partial T_{sat}} = \frac{q}{(T_s - T_{sat})^2} = \frac{h}{(T_s - T_{sat})}
\]  (59)

The terms in Eqns. (57-59) are substituted back into Eqn. (56) to obtain the following.
\[ U_h = \sqrt{\left( \frac{h}{q^n U_q^n} \right)^2 + \left( -\frac{h}{(T_s - T_{sat}) U_{rs}} \right)^2 + \left( \frac{h}{(T_s - T_{sat}) U_{Ts}} \right)^2} \]  

(60)

The squared terms in each of the parentheses in the numerator of Eqn. (60) are expanded in Eqn. (61).

\[ \frac{U_h}{h} = \sqrt{\frac{h^2 U_q^2}{q^n^2} + \frac{h^2 U_{rs}^2}{(T_s - T_{sat})^2} + \frac{h^2 U_{Ts}^2}{(T_s - T_{sat})^2}} \]  

(61)

This allows all \( h^2 \) terms in the numerator to cancel with the denominator, thus yielding the final expression in Eqn. (9), which is reproduced below in Eqn. (62).

\[ \frac{U_h}{h} = \sqrt{\frac{U_q^2}{q^n^2} + \frac{U_{rs}^2}{\Delta T_{rs}^2} + \frac{U_{Ts}^2}{\Delta T_{Ts}^2}} \]  

(62)

### B.4 Exit Vapor Quality Uncertainty Derivation

The relation for exit vapor quality is shown in Eqn. (63) below. The variables in Eqn. (63) are substituted into the error propagation equation (Eqn. (6) in Section 4), as shown in Eqn. (64) and (65) below.

\[ x = \frac{q''A - \dot{m} c_p \Delta T_{sub}}{\dot{m} h_{fg}} = \frac{qA - \dot{m} c_p (T_{sat} - T_{in})}{\dot{m} h_{fg}} \]  

(63)

\[ \frac{U_x}{x} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial x}{\partial \sigma_i} U_{\sigma_i} \right)^2} \]  

(64)

\[ \frac{U_x}{x} = \sqrt{\left( \frac{\partial x}{\partial q^n} U_q \right)^2 + \left( \frac{\partial x}{\partial \dot{m}} U_{in} \right)^2 + \left( \frac{\partial x}{\partial c_p} U_{cp} \right)^2 + \left( \frac{\partial x}{\partial T_{sat}} U_{Ts} \right)^2 + \left( \frac{\partial x}{\partial T_{in}} U_{Tin} \right)^2 + \left( \frac{\partial x}{\partial h_{fg}} U_{h_{fg}} \right)^2} \]  

(65)

The sensitivity coefficients for each variable are evaluated individually. The expressions are then rewritten in terms of the function of interest, \( h \), as shown in Eqns. (66-71).
\[ \frac{\partial x}{\partial q^n} = \frac{\partial}{\partial q^n} \left( \frac{q^n A}{\dot{m} h_{fg}} - \dot{m} c_p (T_{sat} - T_{in}) \right) = \frac{1}{\dot{m} h_{fg}} - 0 = q^n - \dot{m} c_p (T_{sat} - T_{in}) \] (66)

\[ \frac{\partial x}{\partial \dot{m}} = -\frac{q^n A}{\dot{m} h_{fg}} = -\frac{q^n A}{\dot{m} h_{fg}} x \] (67)

\[ \frac{\partial x}{\partial c_p} = 0 - \frac{(T_{sat} - T_{in})}{h_{fg}} = -\frac{(T_{sat} - T_{in})}{h_{fg}} x \] (68)

\[ \frac{\partial x}{\partial T_{sat}} = -\frac{c_p}{h_{fg}} = -\frac{c_p}{h_{fg}} x \] (69)

\[ \frac{\partial x}{\partial T_{in}} = \frac{c_p}{h_{fg}} = \frac{c_p}{h_{fg}} x \] (70)

\[ \frac{\partial x}{\partial h_{fg}} = -\frac{x}{h_{fg}} \] (71)

The terms in Eqns. (66-71) are substituted back into Eqn. (65) to obtain the following.

\[ \frac{U_x}{x} = \sqrt{\left( \frac{x}{q^n A - \dot{m} c_p \Delta T_{sub}} \right)^2 + \left( -\frac{q^n A}{\dot{m} h_{fg}} x U_m \right)^2 + \left( -\frac{\Delta T_{sub} x}{h_{fg}} U_p \right)^2 + \left( -\frac{c_p x}{h_{fg}} x U_{sat} \right)^2 + \left( \frac{c_p x}{h_{fg}} x U_{in} \right)^2 + \left( \frac{c_p x}{h_{fg}} x U_{fs} \right)^2} \] (72)

The squared terms in each of the parentheses in the numerator of Eqn. (72) are expanded, allowing all \( x^2 \) terms in the numerator to cancel with the denominator, yielding the final expression in Eqn. (73).

\[ \frac{U_x}{x} = \sqrt{\left( \frac{U_{q^n}^2}{q^n A - \dot{m} c_p \Delta T_{sub}} \right)^2 + \frac{q^n x^2 U_m^2}{\dot{m} h_{fg}^2 x^2} + \frac{\Delta T_{sub}^2 x^2 U_p^2}{h_{fg}^2 x^2} + \frac{c_p x^2 U_{sat}^2}{h_{fg}^2 x^2} + \frac{c_p x^2 U_{in}^2}{h_{fg}^2 x^2} + \frac{U_{fs}^2}{h_{fg}^2}} \] (73)

**B.5. Mass Flow Rate Uncertainty Derivation:**

In order to determine the total uncertainty of the exit vapor quality in the above equation, the uncertainty in mass flow rate must first be determined. The equation for mass flux is expressed as follows in Eqn. (74), as a function of the volumetric flow rate, which is the measured quantity obtained from testing.
\[ \dot{m} = \rho \dot{V} \]  

(74)

The variables in Eqn. (74) are substituted into the error propagation equation (Eqn. (6) in Section 4), as shown in Eqn. (75) and (76) below.

\[
\frac{U_{\dot{m}}}{\dot{m}} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial \dot{m}}{\partial \sigma_i} U_{\sigma_i} \right)^2} \tag{75}
\]

\[
\therefore \frac{U_{\dot{m}}}{\dot{m}} = \sqrt{\left( \frac{\partial \dot{m}}{\partial \rho} U_{\rho} \right)^2 + \left( \frac{\partial \dot{m}}{\partial \dot{V}} U_{\dot{V}} \right)^2} \tag{76}
\]

The sensitivity coefficients for the density and volumetric flow rate are individually evaluated. The expressions are then rewritten in terms of the function of interest, \( \dot{m} \), as shown in Eqns. (77) and (78).

\[
\frac{\partial \dot{m}}{\partial \rho} = \dot{V} = \frac{\dot{m}}{\rho} \tag{77}
\]

\[
\frac{\partial \dot{m}}{\partial \dot{V}} = \rho = \frac{\dot{m}}{\dot{V}} \tag{78}
\]

The terms in Eqns. (77) and (78) are substituted back into (76), to obtain the final expression for mass flow rate uncertainty, shown below in Eqn. (80).

\[
\frac{U_{\dot{m}}}{\dot{m}} = \sqrt{\left( \frac{\dot{m}}{\rho} U_{\rho} \right)^2 + \left( \frac{\dot{m}}{\dot{V}} U_{\dot{V}} \right)^2} = \sqrt{\frac{\dot{m}^2 U_{\rho}^2}{\rho^2} + \frac{\dot{m}^2 U_{\dot{V}}^2}{\dot{V}^2}} \tag{79}
\]

\[
\frac{U_{\dot{m}}}{\dot{m}} = \sqrt{\frac{U_{\rho}^2}{\rho^2} + \frac{U_{\dot{V}}^2}{\dot{V}^2}} \tag{80}
\]
Appendix C: Supplementary Figures

C.1 Effect of Flow Rate:

Figure 37: Effect of flow rate for plain chip with 2% manifold taper

Figure 38: Effect of flow rate for plain chip with 4% manifold taper
Figure 39: Effect of flow rate for plain chip with 6% manifold taper

Figure 40: Effect of flow rate for chip MC1 with 2% manifold taper
Figure 41: Effect of flow rate for chip MC1 with 4% manifold taper

Figure 42: Effect of flow rate for chip MC1 with 6% manifold taper
Figure 43: Effect of flow rate for chip MC2 with 2% manifold taper

Figure 44: Effect of flow rate for chip MC2 with 4% manifold taper
Figure 45: Effect of flow rate for chip MC2 with 6% manifold taper