A Thermosiphon Loop for High Heat Flux Removal using Flow Boiling of Ethanol in Open Microchannel Manifold (OMM) with Taper

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By:

Sanskar S. Panse

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
Department of Mechanical Engineering
Kate Gleason College of Engineering

ROCHESTER INSTITUTE OF TECHNOLOGY

Rochester, NY
July 27th, 2016
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ABSTRACT

Open Microchannel Manifold (OMM) with taper has proved instrumental in enhancing heat transfer performance in flow boiling while keep the pressure drop to a minimum. This makes it applicable in developing a low pressure drop system like a thermosiphon loop. To this end, a gravity-driven flow boiling system developed earlier was tested at low flow rates using ethanol. Based on the pressure drop and heat transfer data, a two-phase thermosiphon loop with a small ethanol head below 0.2 m was developed and tested with OMM configuration. A maximum heat flux of 136 W/cm² was recorded at a wall superheat of 42 °C. Pressure drop data showed stable thermosiphon operation with lesser flow and pressure fluctuations over the microchannels with increase in heat flux. Stable operation was complimented with tremendously low pressure drop below 4 kPa near Critical Heat Flux (CHF).

Investigations were also carried out on the effect of flow orientation on flow boiling performance in the gravity-driven flow boiling system by varying the orientations as horizontal flow (0°), vertical upflow (90°) and vertical downflow (-90°) flow. Flow couldn’t be sustained in the vertical upflow orientation, however, the system performed best in the horizontal flow orientation. The heat transfer performance of the thermosiphon loop was independent of the orientation of the test section with the horizontal and the vertical upflow configurations giving similar heat transfer performances.

ACKNOWLEDGEMENTS

I sincerely thank Dr. Satish G. Kandlikar and all the members of the Thermal Analysis, Microfluidics and Fuel Cell Lab for their support and guidance in the completion of this work.
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NOMENCLATURE

$q''$  heat flux, W/cm$^2$

$h$  heat transfer coefficient, W/m$^2$K

$k_{Cu}$  thermal conductivity of copper, W/mK

$\Delta T_s$  degree of wall superheat, °C

$\Delta T$  temperature difference, °C

$dx$  distance between adjacent thermocouples, mm

$\Delta p$  pressure drop, kPa

$H$  gravity head, mm

$\rho$  liquid density, kg/m$^3$

$g$  gravitational acceleration, m/s$^2$

$U_p$  error uncertainty in function $p$

$P$  precision error

$B$  bias error
1. INTRODUCTION

Effective thermal management of high heat flux systems has gained widespread importance in
the scientific community over the last couple of decades. Two-phase heat transfer, which utilizes
latent heat of vaporization of liquid has been studied extensively for its high heat dissipation
performance. Kandlikar and Grande [1] presented a channel classification based on the range of
hydraulic diameters passages employed for fluid flow and termed microchannels as flow passages
with hydraulic diameter between 10 µm to 200 µm, as shown in Table 1. Flow boiling heat transfer
in microchannels has shown great promise in enhancing heat transfer performance due to the
inherent ability of microchannels to provide superior heat transfer coefficient [2]. Studies have
shown heat flux greater than 100 W/cm² being dissipated using flow boiling in microchannels
[3,4]. This makes it a promising candidate for cooling high heat dissipation cooling applications
like cooling of servers, turbine blade, solar arrays, boilers and onboard electronics in aircrafts and
satellites, to name a few. However, flow boiling performance is affected due to inherent issues like
flow instabilities and reversals which lead to surface temperature and pressure fluctuations and
intermittent dry-outs, thereby affecting its performance considerably [1,5]. Moreover, utilization
of a pump to drive the fluid renders the system cost ineffective, energy inefficient and bulky,
wherever weight is a constraint. Hence, it was necessary to address these problems before large
scale implementation.

In flow boiling, a sub-cooled liquid flows under forced convective conditions and undergoes
nucleation when exposed to a certain degree of wall superheat (temperature above the fluid’s
saturation temperature). Flow boiling relies on liquid-vapor phase change process during
nucleation to reject heat away from the surface. Flow boiling performance depends on several
factors including heat flux, exit vapor quality, wall superheat, fluid thermal properties, inlet fluid
temperature (degree of subcooling), flow rate, channel geometry and flow orientation with respect to gravitational force. It is desirable to enhance boiling performance by improving heat flux by maintaining wall superheat to a minimum.

<table>
<thead>
<tr>
<th>Channel Classification</th>
<th>Hydraulic Diameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conventional Channels</td>
<td>&gt;3 mm</td>
</tr>
<tr>
<td>Minichannels</td>
<td>200 µm to 3 mm</td>
</tr>
<tr>
<td>Microchannels</td>
<td>10 µm to 200 µm</td>
</tr>
</tbody>
</table>

*Table 1: Classification of flow channels based on hydraulic diameter ranges, Kandlikar and Grande [1]*

Extensive research has been undertaken in reducing flow fluctuations during flow boiling to enhance performance in microchannels. Kandlikar [6] asserted that the onset of bubble nucleation introduces a pressure spike at the location of nucleation which leads to unstable flow conditions. Use of a pressure drop element at the inlet [7,8] helped overcome this problem by reducing flow and pressure fluctuations considerably. Mukherjee and Kandlikar [9] proposed implementation of diverging and expanding microchannels to overcome the flow instability issues. Few researchers employed diverging microchannels [10] and expanding microchannels [11–13] which encouraged forward liquid motion and allowed easy bubble growth and escape. These configurations proved effective in providing stable flow conditions and enhanced the heat transfer performance. Kandlikar et al [14] pioneered a novel Open Microchannel Manifold (OMM) configuration with an increasing downstream taper which provided an added flow area over the microchannel surface and promoted vapor generation and escape without excess pressure drop penalty. A tremendously
high flow boiling performance was obtained with this configuration with water as working fluid. The OMM geometry also resulted in unprecedented pressure reduction below 10 kPa [15].

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 W/mK. 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. Precautions to take while using ethanol owing to its high flammability is presented in Appendix A.

Buchling and Kandlikar [16] inherited the open microchannel and tapered manifold arrangement from Kandlikar et al [14] and Kalani and Kandlikar [15] to investigate the performance of ethanol. Due to the natural ability of OMM to provide remarkably low pressure drop performance, they explored the possibility of employing gravity-assisted flow system, where ethanol was fed into the system through gravity. Reduced flow instabilities, low pressure drop and high heat transfer performance was achieved in this study thus, paving way for further investigation into pumpless cooling system for high heat dissipation applications.

Due to the inherent ability of microchannels to enhance cooling performance, researchers worldwide are actively exploring the implementation of two-phase thermal management systems for numerous earth-based gravity environments. To understand the effect of gravity on flow boiling, extensive research has been carried out by varying the heat sink orientation in Earth’s gravitational field. The effect of gravity on flow boiling performance is reflected in buoyancy. Buoyancy effects scale up due to large density differences between the liquid and vapor phases, and its strength relative to liquid inertia and surface tension forces influences the vapor to liquid interactions and therefore affects the heat transfer [17].

This work builds up upon the research carried out by Buchling and Kandlikar [16] and applicability of the gravity-assisted system was tested for low flow rates using ethanol. Further
investigations were undertaken to evaluate the effect of vapor buoyancy on flow boiling and pressure drop performance by varying the heat sink orientation to allow fluid flow in horizontal flow (0°), vertically downflow (-90°) and vertically upflow (90°) directions with respect to gravity. High speed imaging was carried out to enable visualization of liquid-vapor interactions at various flow orientations and test conditions. Flow boiling performance of ethanol was evaluated on the basis of heat flux, wall superheat, heat transfer coefficient and pressure drop.

Further, based on the pressure drop data and heat flux requirements acquired from the gravity-assisted flow system, a self-sustaining two-phase thermosiphon loop was designed and tested in tapered manifold geometry with ethanol. Using the performance characteristics developed in the experimental plan, the application of a gravity-driven flow boiling system using a head below 0.5 m was evaluated.

Thermosiphon cooling is an alternative cooling technology of dissipating high local heat fluxes. Thermosiphons can be designed either as a single tube or as a closed loop. A thermosiphon loop consists of an evaporator where heat is dissipated from the electronic components and a condenser where heat is rejected to the ambient. These two components are connected by a riser and a downcomer. Thermosiphon cooling offers passive circulation and no mechanical driving parts are necessary for the circulation. But designing compact thermosiphon loops which are able to dissipate moderate to high heat fluxes ($q'' > 80$ W/cm$^2$) can help extend their applicability to areas where space is a constraint as an alternative to heat pipes and vapor chambers.
2. LITERATURE REVIEW

Though flow boiling in microchannels has tremendous potential in enhancing heat dissipation rate, its performance is adversely affected due to factors like flow instability, flow reversals, pressure fluctuations and intermittent dry outs. Literature on these root causes and related performance is reviewed in this section. Since the goal of the project is flow boiling in a thermosiphon loop, the systems that have yielded very low pressure drops are reviewed. These systems have potential to be incorporated in a thermosiphon loop as the small liquid head available can sustain only a low pressure drop in the evaporator.

2.1. Instabilities in Microchannels

Flow reversals and pressure drop fluctuations in parallel microchannels of hydraulic diameter 333 µm were observed during flow boiling by Balasubramanian and Kandlikar [18]. High speed videos of flow maldistributions were captured with vapor slugs in some channels extending into the inlet manifold. They found the pressure fluctuations to be chaotic due to channel-to-channel interactions, with slugs from one channel influencing flow in its adjacent channels.

![Figure 1](image-url)

**Figure 1:** (a) Schematic showing flow maldistributions in microchannels; (b) Magnitude of pressure drop fluctuation in microchannels, Balasubramanian and Kandlikar [18]
Fig. 1(a) shows a schematic of two-phase flow reversals in microchannels with the direction of bulk flow from left to right. Fig. 1(b) shows pressure drop fluctuation in microchannels due to periodic flow reversals.

Kandlikar [6] allocated flow reversals to rapid bubble nucleation which leads to localized pressure spike at the point of inception, causing irregular distribution of wall temperature. He proposed the implementation of pressure drop elements (PDEs) at the inlet and artificial nucleation sites (ANS) in microchannels to counter flow reversals and temperature maldistributions respectively. A numerical study by Mukherjee and Kandlikar [9] also confirmed the effectiveness of PDEs in stabilizing the flow.

![Figure 2: Schematic representing pressure variation following nucleation during flow boiling in microchannels, Kandlikar [6]](image)

Fig. 2 shows a schematic representation of pressure variation and spike due to bubble nucleation in microchannels at equilibrium and stable operating conditions.
2.2. Pressure drop Elements (PDEs) and Artificial Nucleation Sites (ANS)

In a following study, Kandlikar et al. [8] employed inlet restrictors of various sizes in conjunction with ANS in 1054×197 µm microchannels. They found inlet restrictors and ANS to have significant effect on stabilizing the flow and reducing the wall temperature. Heat transfer performance was also seen to have improved with this configuration.

**Figure 3:** (a) Successive images of stable flow conditions with PDEs and ANS in a single channel from a set of six parallel channels; (b) Comparison of pressure drop variation across microchannels and PDEs, Kandlikar et al. [8]

Fig. 3(a) illustrates successive high speed images of stable flow conditions due to the implementation of PDEs and ANS in a single microchannel. Fig. 3(b) shows the comparison of pressure drop fluctuation in microchannel with and without PDEs and ANS. Pressure drop fluctuations die down considerably with the best configuration of PDEs and ANS.

Kosar et al. [19] used a dimensionless pressure drop multiplier parameter M to quantify the extent of flow instability suppression due to inlet restrictors. Microchannels 10 mm long and of 227 µm hydraulic diameter were used while inlet orifice length was varied from 0 to 400 µm. They
found the heat flux corresponding to onset of unstable boiling to increase with longer orifice (larger M).

**Figure 4:** (a) Pressure drop fluctuations at Onset of Unstable Boiling; (b) Boiling curve, Kosar et al. [19]

Fig. 4(a) shows severe pressure fluctuations in microchannels with unrestricted inlets at Onset of Unstable Boiling (OUB). These fluctuations are indication of flow reversals and instabilities in unrestricted microchannels. Fig. 4(b) demonstrates effect of inlet restrictors on heat dissipation performance. Increase in heat flux corresponding to OUB was evident with the use of longer inlet restrictors.

Wang et al. [20] investigated the effect of inlet/outlet flow restrictors on flow boiling instabilities in parallel microchannels of hydraulic diameter 186 µm. Stable flow was realized with diminished flow reversals and, temperature and pressure fluctuations with inlet restrictor.
Figure 5: (a) Arrangement of parallel microchannels with restricted inlet; (b) Schematic of microchannels with inlet area restrictors, Wang et al. [20]

Figs. 5(a) and 5(b) how the sketch of inlet restrictor geometry implemented in the study.

2.3. Microchannels with Downstream Area Expansion

Lee et al. [21] asserted that stable flow conditions can be achieved by employing microchannels expanding downstream. They attributed this to the natural tendency of vapor to flow towards low surface tension region due to increase in flow area. Mukherjee and Kandlikar [9] numerically studied the growth of a bubble inside a microchannel and suggested expanding microchannel configuration to promote unidirectional fluid flow and reduced flow instabilities as shown in Fig. 6.

Figure 6: Stepped microchannels and diverging microchannels geometry proposed by Mukherjee and Kandlikar [9]
Lu and Pan [10] employed diverging microchannels in conjunction with 25 equally spaced artificial nucleation sites along the length of microchannels. They found diverging microchannels instrumental in stabilizing the flow, with ANS enhancing heat transfer performance and minimizing wall temperature.

![Figure 7](image)

**Figure 7:** (a) Schematic representation of diverging microchannels employed; (b) Boiling curve comparison for three types of microchannels, Lu and Pan [10]

Fig. 7(a) show the geometry of diverging microchannels used. The performance of diverging microchannels was tested with and without artificial nucleation sites. In Fig. 7(b), diverging microchannels with artificial nucleation sites drilled throughout its length outperformed other configurations at all mass fluxes.

Miner et al. [11] numerically studied the effect of microchannel expansion on flow boiling performance. Their model used CHF as a measure of performance enhancement in microchannels.
They predicted CHF augmentation higher than 600 W/cm² through analysis of expanding microchannels. In a resulting study, Miner et al. [12] studied pressure drop in parallel microchannels with an expanding base at various expansion angle. They found reduction in pressure drop and stable flow conditions with increase in expansion angle.

![Figure 8](image.png)

**Figure 8:** (a) Schematic representation of expanding microchannels; (b) Plot showing pressure drop performance at various expansion angles as a function of mass flux, Miner et al. [12]

Fig. 8(a) is an illustration of the microchannels with expanding base implemented by Miner et al. [12]. The expansion increases microchannel height downstream, thus, providing greater flow area. Fig. 8(b) shows the strong influence of expansion angle on pressure drop performance, with pressure drop decreasing with increase in degree of expansion.

A similar approach was undertaken by Balasubramanian et al. [13] where they tested the flow boiling performance of deionized water using stepped microchannels. Stepped microchannels were fabricated by reducing the fin height by 400 µm over a certain length, at predetermined locations.
from the inlet. High heat dissipation performance, low pressure fluctuations and stable flow conditions were realized through this configuration.

![Microchannel Heat Sink](image)

**Figure 9:** (a) Stepped microchannel heat sink; (b) Boiling curves at different mass fluxes, Balasubramanian et al. [13]

Fig. 9(b) represents the boiling performance for stepped microchannel geometry. A very high heat flux close to 400 W/cm² was achieved with this configuration.

Kandlikar et al. [14] introduced open microchannel manifold (OMM) geometry by providing a gap over the microchannels for greater fluid flow area. Also, a taper was provided in the manifold with increasing flow area downstream. In addition to a high heat dissipation of 506 W/cm² at 26 °C wall superheat, stable flow conditions with diminished pressure fluctuations were achieved with water as working fluid.
**Figure 10:** *a) Schematic showing open microchannels with tapered manifold; (b) Boiling performance with uniform manifold block, Kandlikar et al. [14]*

Figs. 10(a) and 10(b) show the tapered manifold geometry used in the study and the high boiling performance associated with open microchannels respectively.

In a latter study, Kalani and Kandlikar [15] tested the effect of tapered manifold geometry on pressure drop. The manifold inlet height was maintained at 127 µm while the outlet height was varied as 327 µm, 527 µm and 727 µm. A reduction in pressure drop up to 3.3 kPa was attained with microchannels in a tapered manifold geometry at 281.2 W/cm² and 10.1 °C superheat.
Fig. 11: Pressure drop performance for uniform and tapered manifold blocks with microchannels, Kalani and Kandlikar [15]

Fig. 11 highlights the effectiveness of tapered manifold in keeping the pressure drop to a minimum. Pressure drop reduces with increase in taper angle.

In light of the superior performance achieved using OMM geometry with regards to high heat transfer and remarkably low pressure drop, Buchling and Kandlikar [16] explored the performance of ethanol in open microchannels and tapered manifold. They employed a gravity-assisted flow boiling system, where flow to the test section was provided by gravity through an elevated reservoir. Flow boiling performance of ethanol was tested at flow rates 40, 60 and 80 mL/min with plain and microchannel heat sinks. A heat flux of 217 W/cm² was dissipated at system pressure drop of just 9 kPa. Based on heat transfer and pressure drop performance, they presented opportunity in development of a self-sustaining cooling device like a thermosiphon loop.
Figure 12: (a) Boiling curves at different manifold tapers; (b) Pressure drop performance curves at different manifold tapers, Buchling and Kandlikar [16]

Figs. 12(a) and 12(b) illustrate the effectiveness of open microchannels in tapered manifolds in enhancing the heat dissipation performance and pressure recovery effect respectively.

There is no work directly reported on enhanced flow boiling performance in a gravity-driven pumpless system. Such systems have an advantage over the conventional flow boiling systems in that they do not require a mechanical pump for dissipating high heat fluxes.

The present work further explores the feasibility of gravity-driven system in open microchannel and tapered manifold geometry at low flow rates 10, 20 and 40 mL/min, which gives an inlet Reynold’s number, $Re$ as 35.28, 70.57 and 141.15 respectively at subcooled inlet conditions with ethanol. The overall objective of this research is to develop a self-sustaining flow boiling system like a thermosiphon loop operated under a gravity head of 0.5 m under different flow orientations for Earth-based cooling applications.
2.4. Effect of Gravity on Flow Boiling

Comprehensive studies have been carried out on flow boiling performance of parallel microchannels in horizontal orientation. However, effect of gravity under different heat sink orientations is also an area of interest for earth-based cooling applications. Several researchers carried out investigations on effect of heat sink orientation on thermal performance of both minichannels and microchannels, and studied the various flow patterns associated with them.

Kandlikar [22] reviewed the effect of scaling on flow boiling in microchannels. He identified five major forces influencing micro-scale transport, namely, inertia force, surface tension force, shear force, gravity (buoyancy) force and evaporative momentum force. These forces were normalized with area to investigate their relative effect on each other as,

Inertia force per unit area,

\[
F_i = \frac{G^2}{\rho}
\]

(1)

where, \(G\) – mass flux, \(\rho\) – liquid density

Shear force per unit area,

\[
F_\tau = \frac{\mu G}{\rho D}
\]

(2)

where, \(\mu\) - dynamic viscosity, \(D\) – channel diameter

Buoyancy force per unit area,

\[
F_g = (\rho_l - \rho_g)gD
\]

(3)

where, \(g\) – gravitational acceleration
According to Eqns. (1), (2) and (3), at micro-scale level, the ratio of surface area to volume increases and surface tension and shear forces become more dominant. The effect of gravitational force diminishes with narrower channels.

Kandlikar and Balasubramanian [23] studied the effect of gravitational orientation on flow boiling performance of water in 1054 × 197 µm parallel minichannels. A comprehensive study was undertaken to understand the effect of heat sink orientation on flow patterns in horizontal flow, vertical upflow and vertical downflow orientations. Similar flow patterns with periodic flow reversals were observed in all flow orientations. However, reversals were more prominent in vertical downflow, thereby affecting its performance.

![Figure 13](image)

**Figure 13:** (a) Pressure drop variation in all three flow orientations; (b) Heat transfer performance at all three flow orientations, Kandlikar and Balasubramanian [23]

Fig. 13(a) shows the pressure drop variation in all three flow orientations under same operating conditions. Pressure drop associated with vertical upflow was found to be the lowest. Fig. 13(b) shows lowest heat transfer performance with vertical downflow as compared to other two orientations. Kandlikar and Balasubramanian [23] also asserted that decreasing channel hydraulic diameter serves to weaken the effect of buoyancy on two-phase flow, according to Eqn. (3).
Lee at al. [24] drew a similar conclusion when they tested flow boiling of FC-72 in 80 231 \( \mu \text{m} \times 1000 \mu \text{m} \) at horizontal, vertical upflow and vertical downflow orientations. Their data showed gravity effects are negated in microchannels at velocities far below than those for higher hydraulic diameter channels.

Zhang et al. [25] investigated thermal performance of FC-72 in horizontal, vertical upflow and vertical downflow orientations in 21 parallel microchannels. They found the vertical upflow orientation more effective in maintaining a lower thermal resistance, with better flow stability and lowest pressure drop. However, vertical upflow and downflow orientations were found to perform equally at higher flow rates due to higher mass fluxes, thus, better fluid inertia according to Eqn. (1), which helps to overcome buoyancy forces.

![Figure 14: Heat sink thermal resistance for vertical downflow and vertical upflow orientations, Zhang et al. [25]](image)

From Fig. 14, the thermal resistance of the heat sink was measured to be higher for vertical downflow orientation as compared to vertical upflow, thus, showing a lower thermal performance.
Zhang et al. [26] undertook a detailed analysis of CHF mechanisms of FC-72 associated with a wide range of heat sink orientations. The inlet fluid velocity was varied from 0.1 – 1.5 m/s for each orientation. Wavy Vapor layer regime was observed at higher fluid velocity for all orientations, however, flow regimes in downward flow orientation had a strong dependence on relative magnitudes of liquid inertia and buoyancy forces. Sensitivity of CHF to orientation was strongest for lower fluid velocities and a negligible at higher values.

**Figure 15:** (a) Flow orientation guide showing flow direction, channel orientation and heater position; (b) CHF variation with flow orientation and flow velocity, Zhang et al. [26]

Fig. 15(a) illustrates the various flow orientation implemented in the study. Fig. 15(b) shows the variation in CHF with orientation and flow velocity. Effect of orientation on CHF was the lowest for two highest velocities, 1 m/s and 1.5 m/s, with buoyancy effect more dominant at lower velocities, affecting the performance in downward facing heater orientations (135° - 270°). A CHF maximum at θ = 45° was explained by the buoyancy force both aiding vapor removal away from the heated wall as well as along the channel.
Effect of flow orientation on flow boiling performance of HFE-7100 was studied by Wang et al. [27] in multiport microchannels of hydraulic diameter 825 µm. They varied the orientation from -90° to 90° with respect to the horizontal, with a step increment of 45°. The performance with 45° upward flow exceeded other orientations at the lowest flow rate tested, with downward flow orientation being the least performing. Higher slug velocity assisted with buoyancy force was found to be reason behind superior performance with 45° upward flow.

*Figure 16: (a) Schematic of test orientations; (b) Influence of flow orientation on heat transfer performance, Wang et al. [27]*

Fig. 16(b) shows significant influence of orientation on heat transfer performance with performance improves as orientation was varied from 0° to 45°. Vertical downflow (90°) gives the lowest heat transfer performance.

Leao et al. [28] investigated the effect of heat sink orientation on flow boiling performance of R245fa in 50 parallel microchannels. Tests were carried out up to a heat flux of 300 kW/m² (30
W/cm²) and for mass velocities 300 – 1000 kg/m²s. They found increasing effect of heat sink orientation on heat transfer coefficient at higher mass fluxes.

**Figure 17:** Boiling curves at 500 and 800 kg/m²s at all three orientations, Leao et al. [28]

Fig. 17 shows dependence of boiling performance on orientation and mass flux with effect of orientation magnifying at higher mass fluxes.

Similar trends were observed by Gersey and Mudawar [29,30] when heater orientation effect on CHF diminished at increasing flow velocities and increasing subcooling for FC-72.
3. EXPERIMENTAL SETUP FOR LOW FLOW RATE PERFORMANCE TESTING OF OMM WITH TAPER

Flow boiling relies on forced convection effects of the working fluid for heat transfer. Thereby, it can be tempting to increase the fluid flow rate to enhance performance. However, increasing flow rate increases the system pressure drop and thus, calls for higher pumping power, rendering the system complex and inefficient. Also, low pressure drop is a stringent requirement for developing self-sustaining cooling systems like thermosiphon loops, where fluid flow is governed by gravity. Kandlikar et al. [14] and Kalani and Kandlikar [15] proved the superiority of open microchannels with tapered manifolds (OMM) in providing high heat transfer performance and simultaneously maintaining a minimum pressure drop. This makes OMM configuration a great candidate for application in developing a self-sustaining flow boiling system.

A setup was developed to investigate flow boiling in open microchannels and tapered manifolds as shown in Fig. 18. A pump was absent in this system as fluid flow was achieved solely with the help of gravity between the two reservoirs. A vertical separation of 2.86 m was maintained between the reservoirs, which provided a gravity head of 20 kPa with ethanol.

A flow boiling test was initiated by feeding the ethanol from the elevated reservoir into the preheater section, where it was heated to a subcooled temperature of 55°C, degree of subcooling of 23°C below saturation temperature, with the help of two Omega HTC-060, 125 W rope heaters. When the desired subcooled temperature was reached, the main heaters to the test section were turned ON where subcooled ethanol was heated to saturation inside the test section. The test section consisted of plain or microchannel heat sink on a copper heater block and a polished uniform or tapered manifold for fluid flow. Flow boiling tests were run over a ranges of heat fluxes up to
CHF. The liquid ethanol and vapor mixture from the test section flowed into the condenser where it was cooled down to room temperature before flowing downstream into the lower reservoir. The lower reservoir was placed on the ground to ensure maximum gravitational head for ethanol flow. On attainment of CHF, the power to the preheater and test section were turned OFF and system was allowed to cool down to room temperature with the unused fluid from the elevated reservoir or with a fan. The lower reservoir, which was now filled was swapped with the empty elevated reservoir and the setup was ready for the next test.

Figure 18: Schematic representation of the test setup
The reservoirs used on the setup were flexible jerry cans having a storage capacity of 10 L (2.6 gal), enough to run a complete flow boiling test. The bags were hermitically sealed to prevent air leakage and were resistant to ethanol. The flow through the system was measured by an OMEGA 1007 air/water turbine flow meter which had an operating range from 10 – 100 mL/min measured with an accuracy of +/-1%. The flow rate was regulated by adjusting valves V-01 and V-02. The pressure drop across the test section was measured by Honeywell FP2000, 5 psid (34.47 kPa) differential pressure transducer. The condenser was a shell and tube heat exchanger cooled by a cold water loop with an air-cooled chiller loop featuring two fans with a cooling capacity of 2082 kW at a flow rate of 7.6 L/min (2 gal/min). The condenser ensured cooling down of ethanol liquid and vapor mixture to room temperature before flowing downstream into the lower reservoir. This was essential since high ethanol temperature may cause high pressure buildup inside the bags which may eventually lead to bursting. K-type thermocouples with an operating range of -200-1250°C were used to monitor temperature at the test section inlet, test section outlet, heater block and condenser outlet.

Fig. 19 shows the various heat sink orientations tested and the flow direction associated with them. The heat sink was oriented horizontally and vertically to allow fluid flow in horizontal flow (0°), vertical upflow (90°) and vertical downflow (-90°) directions. All tests were run at atmospheric conditions and changing heat sink orientation did not alter the overall layout of the system.
3.1. Experimental Procedure

The test was commenced by powering ON the preheater and main heater power supply units, solenoid power supply and data acquisition system. The LabVIEW VI was initiated and the cold water loop to the condenser and the chiller assembly was plugged in. The output from both the heater power supply units was controlled with the LabVIEW VI. Ethanol flow rate was regulated using valves V-01 and V-02 shown in Fig. 18. The rope heaters were then engaged to precondition ethanol to predetermined subcooled temperature of 55°C, 23°C degree of subcooling below saturation. Once the required subcooled temperature was reached, the main heater power supply unit was enabled to heat ethanol in the test section up to saturation. The main heater unit was operated at a step increment of 5V from 40 V up to the maximum voltage possible before CHF. Once steady temperature conditions were realized, data was logged for a period of 10 sec at a step size of 0.2 sec. CHF was identified as the point wherein there was a precipitous rise in wall superheat. All tests were terminated when wall temperature reached 125°C, at which point, the
power supplies to the main heater and the preheater heater were switched OFF. This precautionary measure was taken to protect the test section and especially, the manifold from damage due to overheating.

3.2. Degassing Procedure

Liquid degassing is an important step involved in performing a flow boiling test. Degassing procedure involves eliminating the presence of dissolved air in the working fluid by heating it to saturation temperature. The oxygen solubility limit of a liquid is inversely related to its operating temperature. Thus, a liquid at saturation temperature displays least oxygen solubility, at which point the undissolved air can be extracted out. An improperly degassed liquid when engaged in a flow boiling test leads to formation of gas bubbles, leading to premature onset of nucleate boiling at wall temperatures lower than desired and eventually, early CHF.

The degassing process was carried out by simultaneous boiling and condensing ethanol inside the test system. Ethanol was allowed to flow from the elevated reservoir into the preheater section where it was heated to saturation. The mixture of ethanol vapor and other dissolved gases were then allowed to flow through the condenser, where ethanol vapors condensed back to liquid. The area near valve V-03 was intentional of a larger diameter as compared to the rest of the setup which allowed for a film of vapor to form over the liquid surface due to density difference. The 2-phase mixture of undissolved gases and uncondensed ethanol vapor were siphoned out the system through valve V-03, while condensed ethanol continued downstream into the reservoir. A schematic representation of this phenomenon was shown in Fig. 20. A vacuum pump, set to a low pressure drop of 1.3 kPa, was engaged to draw out the two-phase mixture. A liquid-vapor separator chamber was installed between the vacuum pump and the test setup, which segregated the liquid-vapor mixture by drawing liquid ethanol into the chamber, while the vapors continued towards the
vacuum pump. The liquid-vapor separator chamber was essential to the current setup since it prevented damage to the pump by corrosive action of ethanol. The gases expelled from the vacuum pump were then siphoned out of the laboratory environment with an exhaust fume hood.

![Diagram](image_url)

**Figure 20:** Separation of liquid ethanol and two-phase mixture near valve V-03

### 3.3. High Speed Visualization

High speed visualization was carried out using Photron FASTCAM 1024PCI to study bubble growth and flow patterns associated with ethanol. The camera has an ability to capture high-speed videos at 100,000 fps, however, visualizations was carried out at frames rates 3,000 – 6,000 fps for most of the tests. The high-speed videos were viewed through Photron FASTCAM Viewer computer application which allowed the user to control frame rate, resolution, shutter speed, as well as, control brightness, contrast and gamma level for optimal video quality. The camera was mounted on a four-axis stand whose motion was controlled by servo motors mounted on each axis. The motors were user-operated through a software called VELMEX COSMOS which communicated with VELMEX motor controller units for proper placement of the camera over the
desired visualization area. A high intensity light source was used to illuminate the test section for greater quality images.

3.4. Data Acquisition System

Data acquisition was carried out with NI CompactDAQ system which communicated with NI LabVIEW VI. The data acquisition system consisted of modules for calibrated thermocouples, flow sensor and pressure sensor. The LabVIEW VI was capable of controlling PSUs to the preheater and main heaters independently. Data was logged when steady state conditions were realized for a period of 10 sec at a step size of 0.2 sec. Thus, 48 samples were recorded per data point. A standard waiting period of 10 mins was established between two log cycles. Steady state was defined as the point when temperature variation was within +/- 0.2°C over a time period of 10 mins.

3.5. Data Reduction

Heat flux through the heater block was calculated using the Fourier’s law equation for 1-D conduction,

\[ q'' = -k_{cu} \frac{dT}{dx} \] (4)

The temperature gradient \( \frac{dT}{dx} \) in Eqn. (1) was approximated using Taylor’s backward difference,

\[ \frac{dT}{dx} \approx \frac{3T_1-4T_2+T_3}{2\Delta x} \] (5)

where \( T_1 \), \( T_2 \) and \( T_3 \) are the temperature reading from the top, middle and bottom thermocouples respectively and \( \Delta x \) was the separation between adjacent thermocouples, which was 3 mm.
The surface temperature was extrapolated using top thermocouple and the heat flux, accounting for thermal resistance of copper and the separation between the top thermocouple and heat sink surface $x_I$ as,

$$T_s = q'' \left( \frac{x_I}{k_{Cu}} \right)$$  \hspace{1cm} (6)

Heat transfer coefficient was calculated taking into account the heat sink surface temperature, heat flux and saturation temperature of the fluid as,

$$h = \frac{q''}{(T_s - T_{sat})}$$  \hspace{1cm} (7)

The exit vapor quality is calculated taking into consideration the inlet subcooling as,

$$x = \frac{1}{h_{fg}} \left[ \left( \frac{q'' A}{\dot{m}} \right) - c_p \Delta T \right]$$  \hspace{1cm} (8)

where, $h_{fg}$ – latent heat of vaporization, $q''$ – heat flux, $A$ – projected area, $\dot{m}$ - mass flow rate, $c_p$ – fluid specific heat, $\Delta T$ – degree of subcooling.

3.6. Uncertainty Analysis

A comprehensive uncertainty analysis was undertaken to compute uncertainties associated with each measured and computed quantity. Standard expression for error propagation for any property $p$ as a function of an independent variable $\sigma$, over $n$ samples was given by,

$$U_p = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial p}{\partial \sigma_i} U_{\sigma_i} \right)^2}$$  \hspace{1cm} (9)

Precision error, $P$ was calculated as the standard deviation over a set of $n$ samples for a given data point. Bias error, $B$ was the uncertainty arising from the accuracy of the instrument measurement after sensor calibration has been performed. All uncertainty measurements fell
within the 95% confidence interval as specified by JHT. The total uncertainty values in the results are shown as error bars in all the figures.

The total uncertainty associated with heat flux, $q''$ was calculated by substituting Eqns. (1), (2) in Eqn. (5) as,

$$\frac{u_{q''}}{q''} = \sqrt{\frac{u_{kCu}^2}{k_{Cu}^2} + \frac{u_{q}'^2}{\Delta d^2} + \frac{9u_{T1}^2}{\alpha^2} + \frac{16u_{T2}^2}{\alpha^2} + \frac{u_{T3}^2}{\alpha^2}}$$ (10)

where, $\alpha = 3T_1 - 4T_2 + T_3$, helps to simplify the above equation.

The uncertainties $U$ related to temperature readings $T_1$, $T_2$ and $T_3$ was a combination of bias and precision errors of their respective measurements, where the bias error associated with temperature was computed from the 3 point NIST thermocouple calibration process.

The total uncertainty in measurement of heat transfer coefficient $h$ was computed by substituting Eqns. (4) in Eqn. (5) as,

$$\frac{u_{h}}{h} = \sqrt{\frac{u_{q}^2}{q'^2} + \frac{u_{T1}^2}{\Delta T_{sat}^2} + \frac{u_{T2}^2}{\Delta T_{sat}^2}}$$ (11)

Since, pressure drop was a measured quantity in this work, there was no error propagation associated with it. The total uncertainty was calculated as a combination of bias errors and precision errors as,

$$\frac{u_{\Delta p}}{\Delta p} = \sqrt{B_{\Delta p}^2 + P_{\Delta p}^2}$$ (12)

The maximum uncertainty associated with heat flux was below 10%, not exceeding 14 W/cm² at higher heat fluxes. Uncertainty in heat transfer coefficient measurement was up to 10%, below 7000 W/m²K. The bias error associated with pressure drop was minimum, thanks to high accuracy.
of pressure transducer. However, large precision error were due to periodic flow instabilities due to rapid bubble nucleation.

3.7. Test Section

The test section shown in Fig. 21 consists of a tapered manifold, heater block embedded with plain microchannel heat sinks and heater block thermocouples. The microchannel heat sink featured in this study was the best performing microchannel geometry, referred to as MC1 in Buchling and Kandlikar [16]. The manifold blocks incorporated in the study were the same as in Kandlikar et al. [14] and Kalani and Kandlikar [15]. The tapered manifold blocks were provided with a recess which gradually increases the flow area downstream. A recess was absent in case of uniform manifold, wherein manifold height remains constant over the heat sink surface. This configuration addresses the issues of flow instability and low CHF incurred in flow boiling in microchannels by allowing easy bubble growth and escape and replenishment of liquid to prevent channel dry-out.

![Figure 21: Schematic representation of Tapered Manifold and Open Microchannels](image-url)
The manifold was fabricated from lexan polysulphone, which has a glass transition temperature of 140 °C and was polished to provide clear visualization with a high speed camera. Four types of manifold tapers are employed in the study, one uniform and three manifolds with gradually increasing tapers of varying degrees. The manifold tapers were varied as 2%, 4% and 6%, where the percent taper corresponds to the change in taper height along the heat sink length of 10 mm. Thus, 2%, 4% and 6% tapers corresponded to change in taper height of 200 µm, 400 µm and 600 µm respectively. A compressible silicon gasket was placed between the manifold and the heater block. When all the screws were tightened, the gasket served the purpose of sealing the test section and also compresses to the required manifold inlet gap height of 127 µm. While the inlet gap height was maintained at 127 µm for all the tests, the exit gap height was varied as 127 µm, 327 µm, 527 µm and 727 µm for the uniform, 2%, 4% and 6% taper manifolds respectively. Table 2 includes tabulation of manifold types used in the study with their inlet and outlet configurations.

<table>
<thead>
<tr>
<th>Manifolds</th>
<th>Inlet height</th>
<th>Outlet height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td>127 µm</td>
<td>127 µm</td>
</tr>
<tr>
<td>2% taper</td>
<td>127 µm</td>
<td>327 µm</td>
</tr>
<tr>
<td>4% taper</td>
<td>127 µm</td>
<td>527 µm</td>
</tr>
<tr>
<td>6% taper</td>
<td>127 µm</td>
<td>727 µm</td>
</tr>
</tbody>
</table>

*Table 2: Manifold geometries tested in the study*

The heat sink and heater block assembly was shown in Fig. 22. The heater block was machined from a solid copper block using a computer numerical control (CNC). The heat sink surface was
embedded over the copper block. The heat sink had a dimension of 15 mm × 15 mm on which microchannels were machined over a central 10 mm × 10 mm area using CNC. Microchannels were absent on the plain chip and had a flat polished surface. The contact of test heat sink and the fluid was restricted to 10 mm × 10 mm area with the help of high temperature silicon gasket. The microchannel heat sink presented in this study consists of 25 parallel channels each having a channel depth, channel width and fin width of 250 µm, 299 µm and 106 µm respectively. The plain heat sink was also tested having the same dimension to establish baseline performance.

Figure 22: Schematic representation of the heat sink and heater block assembly

Three holes were drilled below the heat sink surface to accommodate the thermocouple probes. The holes had a separation of 3 mm from each other, with the top hole drilled 1.5 mm from the heat sink surface. Four ports were provided near the base of the heater block to accommodate four WATLOW 400 W cartridge heaters which are voltage-controlled using NI LabVIEW VI. The heat
loss from the heater block was minimized by surrounding it with a high temperature ceramic sleeve, ceramic blocks and base ceramic plate.

A heat loss study of the test section conducted in [31] using ANSYS Fluent® is presented in Appendix B.
4. RESULTS

The flow boiling tests were performed with a plain and a microchannel heat sink at flow rates 10, 20 and 40 mL/min. Three flow orientations, horizontal flow (0°), vertical downflow (-90°) and vertical upflow (90°) were tested in four manifold geometries, uniform, 2% taper, 4% taper and 6% taper at atmospheric conditions with ethanol. Flow boiling performance was evaluated as a function of heat flux, wall superheat, heat transfer coefficient and pressure drop. All tests were performed for a range of heat flux up to CHF. CHF was identified as the point wherein there was a sudden rise in wall superheat, after which, the test was terminated.

The manifold with 6% taper was not tested at vertical downflow orientation due to tremendously poor flow performance achieved with 4% taper accompanied by intense flow reversals and pressure fluctuations. Due to large instabilities and frequent reversals, flow could not be sustained, especially at lower flow rates, leading to longer dry-out periods.

Also, flow boiling data for vertical upflow orientation is not presented in this study due to inability of gravity-driven flow to provide a continuous liquid supply over the heat sink surface at this orientation. This led to periodic reversals of flow from the outlet manifold back over the heat sink surface, obstructing the flow of incoming liquid. This phenomenon was observed at moderate heat fluxes, thus rendering it difficult to run entire flow boiling tests. The inlet/outlet manifold configuration was found to be reason behind the issue. The modified manifold design that can be implemented to overcome this problem is presented in the Section 8.

4.1. Horizontal Orientation

The heat sink was oriented at 0° with reference to horizontal, as shown in Fig. 19. The flow boiling performance results at this orientation are presented here.
4.1.1. **Plain Heat sink**

The plain heat sink was tested to establish baseline performance in uniform and tapered manifold geometries. Figs. 23(a) and 24(a) show the boiling curves for all manifolds at flow rate of 10 mL/min and 20 mL/min respectively. Superior boiling performance is achieved when heat dissipation is greater with a lower wall superheat. Heat dissipation was maximum with 4% taper manifold geometry as compared to all other tapers at both flow rates. A heat flux of 64 W/cm$^2$ at 45.73°C wall superheat was obtained at 10 mL/min, while 90.46 W/cm$^2$ at 43.56°C was achieved at 20 mL/min. Figs. 23(b) and 24(b) show the variation in system pressure drop as a function of heat flux up to CHF at 10 mL/min and 20 mL/min respectively. Pressure drop increases with increase in heat flux, however, higher manifold tapers prove effective in maintaining it to a minimum. Lowest pressure drop was recorded in the 6% taper manifold, while it was highest in uniform manifold. A pressure drop, as low as, 1.28 kPa and 1.61 kPa respectively was recorded at highest heat flux point at those flow rates. Figs. 23(c) and 24(c) show variation in heat transfer coefficient as a function of exit vapor quality at 10 mL/min and 20 mL/min respectively. The manifold with 4% taper provides the highest heat transfer performance as compared to other manifolds, with heat transfer coefficient of 15,500 W/m$^2$K and at 29,900 W/m$^2$K at 10 mL/min and 20 mL/min respectively. The exit vapor quality in these cases was 0.28 and 0.22.
**Figure 23:** Effect of manifold taper for plain heat sink at 10 mL/min in horizontal orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality.
Figure 24: Effect of manifold taper for plain heat sink at 20 mL/min in horizontal orientation, a) Boiling curve; b) Pressure Drop Performance; c) heat transfer coefficient with exit vapor quality.
4.1. *Microchannel Heat sink*

The microchannel geometry proved instrumental in enhancing the heat transfer performance and lowering the pressure drop over the plain heat sink. Figs. 25(a) and 26(a) show the boiling curves for all manifold tapers at 10 mL/min and 20 mL/min. Heat dissipation from microchannel surface was the highest with the 4% manifold taper at these flow rates. A heat flux of 90.63 W/cm$^2$ at 41.53°C wall superheat was obtained at 10 mL/min, while 115.53 W/cm$^2$ at 48.36°C was achieved at 20 mL/min. Figs. 25(b) and 26(b) show the variation in system pressure drop as a function of heat flux up to CHF at 10 mL/min and 20 mL/min respectively. Pressure drop increases with increase in heat flux, however, higher manifold tapers proved effective in maintaining it to a minimum. Lowest pressure drop was recorded in the 6% taper manifold, while it was highest in uniform manifold. A pressure drop, as low as, 0.92 kPa and 1.39 kPa respectively was recorded at highest heat flux point at those flow rates. Figs. 25(c) and 26(c) show variation in heat transfer coefficient as a function of exit vapor quality at 10 mL/min and 20 mL/min respectively. The manifold with 4% taper provided the highest heat transfer performance as compared to other manifolds, with heat transfer coefficient of 28,900 W/m$^2$K and 34,643 W/m$^2$K obtained at 10 mL/min and 20 mL/min respectively. The exit vapor quality in these cases was 0.39 and 0.37.

The performance diminished with increase in the taper gradient to 6% for both plain and microchannels chips. This is attributed to the higher manifold taper height which leads to vapor accumulation to due low mass fluxes and low fluid inertia associated with low the flow rates of 10 and 20 mL/min.
Figure 25: Effect of manifold taper for microchannel heat sink at 10 mL/min in horizontal orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality
Figure 26: Effect of manifold taper for microchannel heat sink at 20 mL/min in horizontal orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality.
4.2. Vertically Down Orientation

In this case, the heat sink was orientated vertically at -90° with respect to horizontal, as shown in Fig. 19, with ethanol flowing vertically downward over the heat sink surface. 6% taper was not tested at this orientation due to tremendously poor flow performance achieved with 4% taper accompanied by intense flow reversals and pressure fluctuations. Due to large instabilities and frequent reversals, flow could not be sustained, especially at lower flow rates, leading to longer dry-out periods.

4.2.1. Plain Heat sink

Plain heat sink was tested to establish baseline performance in vertical downflow orientation. Figs. 27 and 28 show the boiling curve, pressure drop performance and heat transfer performance at 10 mL/min and 20 mL/min respectively. Figs. 27(a) and 28(a) show slight improvement in heat dissipation performance with increase in manifold taper from uniform to 2%, however, performance was affected as manifold taper was increased further to 4%. Pressure drop performance at flow rates, 10 mL/min and 20 mL/min for all manifold taper was compared in Figs. 27(b) and 28(b). The plots show a weak dependence of pressure drop on the heat flux, however, higher manifold tapers proved effective in maintaining low pressure drops, similar to in case of horizontal flow orientation. Pressure drop in 4% taper manifold was maintained at 1.21 kPa and 1.44 kPa at the two flow rates respectively. Figs. 27(c) and 28(c) show the variation in heat transfer coefficient as a function of exit vapor quality for all manifold tapers at 10 mL/min and 20 mL/min respectively. A negative effect of taper was prominently seen on the heat transfer performance with uniform manifold showing performance superior to tapered manifolds. Highest heat transfer coefficient with uniform manifold was recorded at 14,492 W/m²K as against 9872 W/m²K with 4% taper manifold at 10 mL/min and, 17,762 W/m²K with uniform manifold as against 15,241
W/m²K with 4% taper manifold at 20 mL/min. This is a drastic variation from the trends observed in horizontal flow orientation, thus, pointing towards pronounced effect of buoyancy forces on liquid-vapor interactions and thus, the flow boiling performance at this orientation, which will be discussed later.
Figure 27: Effect of manifold taper for Plain heat sink at 10 mL/min in vertical downflow orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality.
Figure 28: Effect of manifold taper for Plain heat sink at 20 mL/min in vertical downflow orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality
4.2.2. Microchannel Heat sink

Figs. 29 and 30 show the plots for boiling curve, pressure drop performance and heat transfer performance at 10 mL/min and 20 mL/min respectively for the microchannel heat sink in all manifolds. Figs. 29(a) and 30(a) show a weak dependence of heat dissipation performance on manifold taper geometry with 4% taper providing better performance than other manifolds. Pressure drop performance for all manifold taper was compared in Figs. 29(b) and 30(b). The plots show a weak dependence of pressure drop on the heat flux, however, higher manifold tapers proved effective in maintaining low pressure drops, similar to in case of horizontal flow orientation. Pressure drop in 4% taper manifold was maintained at 1.22 kPa and 1.55 kPa at the two flow rates respectively at highest heat fluxes. Figs. 29(c) and 30(c) show the variation in heat transfer coefficient as a function of the vapor quality for all manifold tapers. Similar to the case of plain heat sink, heat transfer performance with uniform manifold was superior to tapered manifold geometries. Highest heat transfer coefficient with uniform manifold was recorded at 27,290 W/m²K as against 23,413 W/m²K with 4% taper manifold at 10 mL/min and, 32,040 W/m²K with uniform manifold as against 28,500 W/m²K with 4% taper manifold at 20 mL/min. Even though conclusions cannot be drawn on the effect of manifold taper on heat transfer coefficient from Figs. 29(c) and 30(c), it won’t be surprising if the curves follow the trends as in case of plain heat sink in Figs. 27(c) and 28(c).
Figure 29: Effect of manifold taper for Microchannel heat sink at 10 mL/min in vertical downflow orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality
Figure 30: Effect of manifold taper for Microchannel heat sink at 20 mL/min in vertical downflow orientation, a) Boiling curve; b) variation in pressure drop with heat flux; c) variation in heat transfer coefficient with exit vapor quality
5. DISCUSSIONS

5.1. Effect of Flow Orientation

In this section, effect of flow orientation (horizontal flow (0°) and vertical downflow (-90°)) on heat transfer and pressure drop performance are analyzed for plain and microchannel heat sink in uniform and tapered manifold geometries.

5.1.1. Effect of Flow Orientation on Heat Transfer Performance

Effect of horizontal flow (0°) and vertical downflow (-90°) flow orientations on the performance of plain and microchannel heat sinks is presented in Figs. 31(a) and 31(b) in uniform and 4% taper manifold geometries. It is seen that both plain and microchannel heat sinks show superior heat dissipation performance in horizontal orientation as compared to vertical downflow orientation in both manifolds. This difference in performance is attributed to the action of gravitational force which is reflected in the form of buoyancy on the liquid-vapor interface. In downward moving fluid, buoyancy effect forces the vapor to rise in the direction opposite to that of incoming fluid. The effect of buoyancy on flow boiling performance depends on its relative strength to liquid inertia forces. Thus, at low mass fluxes buoyancy effects scale up due to low liquid inertia, leading to intensified flow reversals and pressure fluctuations, based on Eqns. (1) – (3). Similar trends are observed in Figs. 32(a) and 32(b) where, heat transfer coefficient is higher when fluid flows horizontally over the heat sink surface.

It is also worth noting that effect of flow orientation was more prominently seen on the performance of 4% taper manifold than on the uniform manifold. Figs. 31 and 32 show a notable reduction in heat flux and heat transfer coefficient with 4% tapered manifold as the orientation was varied from horizontal flow (0°) to vertical downflow (-90°). This is attributed to the presence of
manifold taper which increases the flow area downstream. Buoyancy which was a function of the channel hydraulic diameter increases linearly with increasing taper. Since buoyancy counter acts the inertia effects of the incoming fluid, it is seen a major cause of vapor accumulation over the channel surface and intense flow reversals in tapered manifolds in vertical downflow orientation especially at low mass fluxes.
5.1. Effect of Flow Orientation on Pressure drop

Fig. 33 illustrates the effect of flow orientation on pressure drop in uniform and 4% tapered manifold at 10 mL/min. It is seen that flow orientation has an insignificant effect on pressure drop over the entire range of heat flux with similar pressure drop values recorded at both orientations at comparable heat fluxes. Consistent trends are observed even at higher flow rates.
**Figure 33:** Effect of flow orientation on pressure drop performance of microchannel in uniform and 4% tapered manifolds at 10 mL/min

**Figure 34:** Time variation of pressure drop fluctuations in microchannel heat sink with uniform manifold at 40 mL/min

Fig. 34 compares the fluctuation in pressure drop over a period of 10 sec in microchannel heat sink with uniform manifold at both flow orientations at 40 mL/min and heat flux around 100 W/cm². Pressure drop fluctuations with periodic flow reversals were seen at both the orientations,
however, flow reversals were more intense and frequent in vertically downward flow, as seen in the figure. The negative spike in pressure drop specific to vertically down (-90°) flow in Fig. 34 was the leading cause of periodic flow reversals which caused variation in wall temperature and inferior heat transfer performance. These flow reversals lead to accumulation of vapor inside the inlet manifold, obstructing the flow of incoming liquid. High-speed videos have shown liquid entering the channels only through a partial section of the inlet manifold since most of it was filled with vapor.

5.2. Effect of Manifold Taper

Figs. 23(a), 23(c), 24(a), 24(c), 25(a), 25(c) and 26(a), 26(c) show improvement in heat transfer performance with increase in manifold taper in horizontal flow orientation for both plain and microchannel heat sink. However, performance was affected as taper was increased to 6%. A weaker dependence of heat transfer performance on manifold taper was seen in vertically downward flow orientation. However, increase in manifold taper shows deterioration in performance as shown Figs. 27(a), (b), 28(a), (b), 29(a), (b) and 30(a), (b). These trends are the effects of low mass fluxes employed in this study. Open microchannels with tapered manifolds that encourage bubble growth lead to accumulation of vapor on the heat sink surface triggering early CHF and poor heat transfer due to low inertia of the incoming liquid and its inability to wash away excess vapor.

Manifold taper proved instrumental in minimizing the pressure drop in both flow orientations. Lowest pressure drop of 0.92 kPa was recorded with 6% manifold taper in horizontal flow orientation in microchannel heat sink at 10 mL/min. At a heat flux of 50 W/cm², an ethanol head of 0.12 m was enough to drive the system in this configuration.
5.3. Effect of Flow Rate

Fig. 35 depicts the effect of flow rate on heat transfer performance of plain and microchannel heat sink in 2% tapered manifold at horizontal flow (0°) and vertical downflow (-90°) orientations. An increasing trend in performance was observed with increase in flow rate as expected. A prominent performance improvement was seen as the flow rate was increased from 20 mL/min to 40 mL/min, with improvement being relatively slight as flow rate was increased from 10 mL/min to 20 mL/min. This proves the scope for enhancement in performance with further increase in flow rate. Buchling and Kandlikar [16] achieved a heat flux close to 200 W/cm² with this configuration at 80 mL/min. However, high flow rates are accompanied by high pressure drops, thus, a higher gravity head to drive the system, compromising the compactness of the system. Hence, a judicious choice of flow rate for a predetermined heat dissipation requirement can help keep the gravity head to a minimum towards designing an optimal, self-sustaining flow boiling system.

The pressure drop with this configuration was recorded between 1.7 – 5.5 kPa for the range of flow rates tested, making it possible to design a gravity-assisted flow boiling system under 0.5 m head with ethanol with a potential to dissipate appreciably high heat flux around 100 W/cm².
5.4. Comparison of Plain and Microchannel Heat sinks

Fig. 36 compares the flow boiling performance of plain and microchannel heat sink at 40 mL/min with 2% taper manifold at horizontal flow (0°) and vertical downflow (-90°) orientations. Microchannel heat sink shows superior performance to the plain heat sink at comparable values of wall superheat, from Fig. 36(a). Fig. 36(b) shows tremendous improvement in heat transfer coefficient with microchannel heat sink over plain heat sink. These trends were consistent even at lower flow rates, thus, underlining the superiority of microchannels in enhancing the flow boiling performance. Higher surface area to volume ratio provided by microchannels in addition to higher heat transfer performance make microchannels an attractive candidate in high heat dissipation applications.

Figure 36: Effect of heat sink geometry on, a) boiling curve, b) heat transfer coefficient at 40 mL/min with 4% manifold taper
5.5. Ethanol Head as a Function of Heat Flux

Based on the pressure drop data obtained through experiments, the gravity head required to drive ethanol through the system without requiring a pump was computed. Fig. shows the required head at flow rates of 10, 20 and 40 mL/min and the corresponding heat fluxes. It acts as a predictor into what performance to expect if a certain ethanol head was applied. This information was useful in developing self-regulating systems like thermosyphons, vapor chambers etc.

An ethanol head around 0.5 m was enough to dissipate heat flux around 150 W/cm² in open microchannel and tapered manifold geometry at flow rate of 40 mL/min. The required head increases with heat flux. For a lower flow rate system, lower heads are enough to obtain a moderately high heat transfer performance. From the figures, a head of 0.2 m was capable of achieving a heat flux over 80 W/cm² without any external pumping power in this configuration.

The driving head was calculated using the relation for static liquid head,

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

(13)

Figure 37: Plots showing ethanol head as a function of heat flux in a) 4%, b) 6% taper manifold
5.6. DEVELOPMENT OF A THERMSYPHON LOOP

A moderately high heat transfer performance (80-140 W/cm²) is achieved with a gravity-driven flow system at low flow rates while maintaining a considerably low pressure drop. This configuration can be replicated at low flow rates into a pumpless configuration. A thermosiphon loop is a device where heat is transmitted continuously from the evaporator to the condenser. The high heat removal capacity and compactness makes thermosiphon an attractive device for high-powered cooling applications. Thus, based on the performance of the gravity-driven system at low flow rates, a self-sustaining two-phase thermosiphon loop is developed and tested with a tapered microchannel manifold. While the fluid flow through the gravity-driven system solely relies on the available gravity head, thermosiphon loop accomplishes fluid flow through liquid-vapor density difference and vapor buoyancy in addition to gravity.

6.1. Test Setup

The thermosiphon loop is shown in Fig. 38. It consists of an evaporator section featuring the open microchannel heat sink with tapered manifold, copper heater block, and the condenser section featuring fluid reservoir and condensing copper coils. The evaporator and the condenser are vertically separated by 0.15 m (150 mm) to provide the required driving head for ethanol flow. The required driving head is computed based on the pressure drop and heat transfer results acquired with gravity-driven setup at low flow rates.
The setup consists of vertical tubes of identical cross-section namely, downcomer and riser at the evaporator inlet and outlet respectively for fluid transport from the condenser to the evaporator and back. The riser has a higher height (by 50 mm) than the downcomer and extends further into the condensing chamber, above the liquid level in the chamber, giving it a total height of 200 mm (0.2 m). This ensures one-way flow from the condenser to the evaporator through the downcomer.

**Figure 38: Two-phase Thermosiphon Loop**
The condensing chamber which serves as liquid ethanol reservoir consists of copper condensing coils at the top. The liquid-vapor mixture from the riser condenses back into liquid and settle back into the reservoir, thus maintaining the desired overall fluid volume in the system. A K-type thermocouple records the liquid ethanol temperature inside the condenser, while a pressure gauge was installed to measure the gauge pressure within the chamber. Atmospheric pressure was maintained inside the chamber over the entire range of the tests. Also, a vacuum pump attached at the top of the chamber syphons out uncondensed gases and also prevents pressure build-up. The liquid height \( h \) within the condensing chamber was varied to investigate the effect of ethanol head on flow boiling performance.

### 6.2. Experimental Procedure

Before the commencement of a boiling test, a predetermined volume of ethanol was filled in the condensing chamber metered with the help of a syringe pump. Tests were run at ethanol fill volumes of 90 mL and 150 mL. These fill volumes corresponded with ethanol head of 15 mm and 25 mm respectively within the chamber in addition to 150 mm of available head between the evaporator and the condenser. Once the fluid was sufficiently degassed, the main heater unit connected to the copper heater block was engaged. Ball valve V-01 was opened to allow the fluid flow from the condenser to the evaporator through the downcomer. The main heater unit was operated at a step increment of 5V (corresponding approximately to 2 W) to the point where the surface temperature reached 120 °C, at which point, the power to the main heater is switched off. Owing to the absence of a CHF mitigation loop that was present in the earlier testing, all thermosiphon tests were concluded before reaching the CHF to avoid damage to the setup due to temperature overshoot.
6.3. Degassing Procedure

An additional degassing loop, same as before, was integrated in the system. Degassing of ethanol was carried out by heating it to saturation temperature within the evaporator. Liquid-vapor ethanol mixture along with uncondensable gases rise into the condensing chamber through the riser. The uncondensable gases are separated from the liquid-vapor mixture at the condenser, where condensed ethanol vapor settles back in the chamber as liquid, while uncondensable gases continue upstream. These gases are removed from the system via the vacuum pump through the liquid-vapor separator chamber. This procedure was allowed to continue for a period of 1 hour once the fluid saturation temperature was reached at the evaporator outlet, after which, the fluid was assumed to be sufficiently degassed.

6.4. Results

To investigate the effect of taper on the flow boiling performance of thermosiphon loop, 4% taper and 6% taper manifolds are tested with the same microchannel heat sink as before with ethanol. Since, 4% taper and 6% taper showed highest performance in the gravity-driven flow configuration at lower and higher flow rates respectively, their performance was tested in thermosiphon loop where ethanol was fed at constant gravity head. It was of great interest to see how manifold taper influences fluid flow rate through the system and in turn affect the heat transfer and pressure drop performance.

Parametric study was undertaken to study the effect of ethanol fill volume (F.V.) as well on the overall performance. F.V. was varied as 90 mL and 150 mL which corresponds to an ethanol head of 15 mm and 25 mm inside the condensing chamber. This equals to total ethanol head of 165 mm and 175 mm respectively between the evaporator and the condenser.
The test matrix for thermosiphon testing was shown in table 3.

<table>
<thead>
<tr>
<th>Manifold Taper</th>
<th>4%</th>
<th>6%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fill Volume (mL)/Total head (mm)</td>
<td>90/165</td>
<td>150/175</td>
</tr>
<tr>
<td></td>
<td>90/165</td>
<td>150/175</td>
</tr>
</tbody>
</table>

Table 3: Test matrix for thermosiphon loop

6.4.1. Heat Transfer Performance

Microchannel heat sink was tested in tapered manifold geometry at constant ethanol heads of 165 mm and 175 mm. Figs. 39(a) and 39(b) show the effect of fill volume on the boiling performance of 4% and 6% manifolds respectively. Fill volume has no effect on the heat transfer performance in 4% taper manifold, however, performance was lowered as fill volume was increased from 90 mL to 150 mL in 6% taper configuration.

Figure 39: Plots showing boiling performance of thermosiphon loop with F.V. of 90 mL and 150 mL in a) 4% taper manifold) 6% taper manifold
6.4.2. *Pressure Drop Performance*

Figs. 40(a) and 40(b) compare fluctuations in pressure drop in the test section over a period of 10 sec at $q^* \approx 30$ and 100 W/cm$^2$ in 4% and 6% tapers. Large variation in pressure drop with periodic negative spikes signify flow fluctuations and reversals, especially at lower heat fluxes. Thermosiphon assumed stable operation with increase in heat flux with little to no flow reversals. Flow was most stable with 6% taper manifold. Pressure drop was consistently below 3.5 kPa.

![Figure 40](image)

*Figure 40: Plots comparing pressure drop fluctuations at low and high heat fluxes in a) 4%; b) 6% manifold taper*

6.5. *Discussions*

6.5.1. *Effect of Taper on Boiling Performance*

Figs. 41(a) and 41(b) show the effect of taper on flow boiling and heat transfer performance of thermosiphon loop at fill volumes 90 mL. Taper was seen instrumental in enhancing the performance as taper was increased from 4% to 6% yielding higher values of heat flux and heat transfer coefficient. Pressure drop fluctuations were also seen to be lower with higher taper signifying more stable flow operations, from fig. 40. Thus, increase in manifold taper seems
beneficial to the performance of thermosiphon loop, although Kandlikar et al. [14] showed that the performance deteriorated as the taper increased from 6% to 10% in flow boiling with water.

![Figure 41: Plots showing effect of manifold taper on, a) Boiling performance, b) Heat transfer performance](image)

6.5.2. **Effect of Ethanol Fill Volume**

Fig. 39 shows little to no change in the dissipation performance with increase in the fill volume. However, performance slightly diminished in case of 6% taper. This might be due to incomplete degassing of fluid, leading to lower heat transfer due to the presence of dissolved gases. However, it was too early to make any judgements based on the limited data pool available and thus, further testing with different fill volumes was expected to give a clear picture of its effect on performance.

6.5.3. **Gravity-driven setup versus Thermosiphon loop**

Figs. 42(a) and 42(b) compare the boiling performance in 4% and 6% manifolds in thermosiphon loop with that of gravity-driven system at 20 and 40 mL/min in the same configuration. For 4% manifold, flow boiling performance of thermosiphon loop completely overlaps with that at 20 mL/min. While, for 6% manifold, it prominently improved over that at 20 mL/min and fell slightly
short of performance curve for 40 mL/min. Due to the absence of a flow meter on the thermosiphon loop, to prevent additional pressure losses associated with it, this superimposition of data gives a good understanding on its behavior in tapered manifold geometry. This highlights the pressure recovery characteristic of tapered manifold in enhancing fluid flow rate through a system, where fluid was fed from a fixed head. Thus, in addition to enhancing the heat transfer while keeping the pressure drop to a minimum, taper was seen to be beneficial in self-regulation of fluid flow rate by allowing easy vapor escape and liquid replenishment. This provides great promise for further testing of tapered manifolds to develop more compact, self-regulated cooling modules like thermosiphon loops and achieve better performance in this configuration.

An approximate flow rate in the system was estimated from Figs. 42(a) and 42(b) and is tabulated below. This assumption allows us to calculate the exit vapor quality using Eqn. 8.

<table>
<thead>
<tr>
<th>Tapers</th>
<th>Flow Rate (mL/min) (Approx.)</th>
<th>Exit Vapor Quality (x)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4%</td>
<td>20</td>
<td>0.31</td>
</tr>
<tr>
<td>6%</td>
<td>35</td>
<td>0.22</td>
</tr>
</tbody>
</table>

*Table 4: Estimated exit vapor qualities at the highest heat flux*

This knowledge of the exit quality enables us to compute the head loss in the downcomer to overcome two-phase flow in the riser. The net driving head caused by the difference in density between the liquid in the downcomer and the vapor/liquid mixture in the riser must be able to overcome the pressure drop caused by the mass flow, for maintaining fluid circulation [32]. The net two-phase fluid density in the riser in computed using the homogeneous flow model which assumes the two-phase flow as a single phase possessing mean fluid properties. The mean fluid density is computed as,
\[ \frac{1}{\bar{\rho}} = \frac{x}{\rho_g} + \frac{(1-x)}{\rho_f} \]  \hspace{1cm} (14)

where, \( \bar{\rho} \) - mean fluid density, \( \rho_g \) – vapor density, \( \rho_f \) – liquid density

The gravitational pressure change in the riser is,

\[ \Delta p_{riser} = \bar{\rho} g H_{riser} \]  \hspace{1cm} (15)

Thus, the net head provided by the system is,

\[ \Delta H_{actual} = \frac{\Delta p - \Delta p_{riser}}{\rho_1 g} \]  \hspace{1cm} (16)

<table>
<thead>
<tr>
<th>Taper</th>
<th>( \Delta H_{provided} ) (mm)</th>
<th>( \Delta H_{actual} ) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4%</td>
<td>165</td>
<td>163.8</td>
</tr>
<tr>
<td></td>
<td>175</td>
<td>173.8</td>
</tr>
<tr>
<td>6%</td>
<td>165</td>
<td>163.37</td>
</tr>
<tr>
<td></td>
<td>175</td>
<td>173.37</td>
</tr>
</tbody>
</table>

**Table 5: Corrected driving head with tapered manifolds**

Table 5 compares the total head with the actual head provided by the thermosiphon loop for fluid flow after taking into consideration the head loss to overcome two-phase pressure drop in the riser tube.

Figs. 42(a) and 42(b) also highlights the ability of thermosiphon loop to provide a superior performance than the gravity-driven system even with a smaller available head. Figs. 37(a) and 37(b) predicted a performance close to 100 W/cm² for a head of 0.2 m at 20 mL/min. However,
results have shown a higher performance. This may be due to: a) the reliance of thermosiphon on liquid-vapor buoyancy difference for fluid flow which allows easy escape of vapor through the riser tube, thus, allowing better replenishment of liquid to the heated surface. The vapor escape was also assisted by the taper present on the manifolds. Flow through the gravity-driven system was driven solely by gravity and the performance primarily depended on the liquid inertia, i.e. fluid flow rate. At lower flow rates, vapor accumulation on the heated surface was seen as the main cause for diminished performance along with low heat transfer coefficients associated with them. b) The varying degree of subcooling. The liquid temperature in the reservoir and thus, the liquid subcooling was seen to increase with increase in heat flux and wall temperature. The fluid temperature in the downcomer varied between 35 °C to 65 °C over the range of the entire test. Higher inlet fluid temperature promotes better heat transfer coefficient and thus, better thermal performance.

Figure 42: Plots comparing boiling performances of thermosiphon loop and gravity-driven system in, a) 4% taper, b) 6% taper
6.5.4. **Effect of Flow Orientation**

An additional study was carried out to observe the effect of flow orientation on the performance of the thermosiphon loop by allowing the fluid to flow upwards on the heat sink surface against gravity. Figs. 43 (a) and 43 (b) compare the heat transfer performance at horizontal flow (0°) and vertical upflow (90°) orientations in 4% manifold at 90 mL fill volume. The plots show similar performance at these orientations.

High speed images shown in Fig. 44 show downstream expansion of nucleating bubbles. The bubble marked in the dark circle expands as it moves downstream before exiting the test section. It was seen to rise above the channel height into the gap provided by the manifold taper. The upward bubble growth into the manifold space allows for improved liquid supply to the nucleating site, as liquid can fill the bottom of the channels as the bubbles continue to grow. The marked and its adjacent bubbles show a pointed front as they move downstream which shows their natural propensity to expand downstream. Due to low fluid inertia at low flow rates, the bubbles rise and expand solely by the action of buoyancy forces in this orientation. The ability of OMM to provide separate liquid-vapor pathways complimented by upward flow orientations is a promising pathway for future research into enhancing performance of thermosiphon loop.
Figure 43: Plots comparing a) boiling curve, b) heat transfer coefficient at horizontal (0°) and vertical upflow (90°) orientations for 4% taper manifold.

Figure 44: High-speed images of downstream expanding bubble in the vertical upflow orientation taken at 50 ms intervals. 4% taper manifold, $q'' = 40$ W/cm$^2$, $\Delta Ts = 20$ °C, flow from bottom to top.
7. CONCLUSIONS

OMM was tested in a tapered manifold geometry in a gravity-driven flow boiling system at low flow rates of 10, 20 and 40 mL/min with ethanol. To investigate the effect of heat sink orientation on heat transfer and pressure drop performance, the heat sink orientation was varied as horizontal flow (0°), vertical downflow (-90°) and vertical upflow (90°) with respect to the horizontal. Based on the heat transfer and pressure drop performance plots acquired through testing, a thermosiphon loop was developed and testing with a small driving head. Following conclusions were drawn through the experimental results:

a. OMM showed remarkable performance at low flow rate, with maximum heat dissipation of 157.6 W/cm² at 43.9 °C wall superheat, while maintaining a low pressure drop of 3.15 kPa at 40 mL/min. Even at the lowest flow rate of 10 mL/min, a heat flux of 90 W/cm² at wall superheat of 41.5 °C was obtained, accompanied by a tremendously low pressure drop of 1.12 kPa. This ability of OMM to provide superior heat transfer and pressure drop performance makes it a good choice in developing a compact, self-sustaining flow boiling system with a small driving head for electronics cooling applications.

b. The ability of OMM with tapered manifold to promote easy vapor escape and liquid replenishment to the heated surface was well established through literature to provide high heat transfer performance with increase in manifold taper downstream. However, at low flow rates of 10 and 20 mL/min, higher taper gradient proved detrimental due to low liquid inertia, which eventually led to excess vapor accumulation in the manifold recess. This was the leading cause of flow and pressure fluctuations in case of 6% taper manifold, which consistently performed lower than the 4% manifold at these flow rates.
c. Flow orientation had prominent effect on the boiling performance with both plain and microchannel heat sinks showing superior heat dissipation performance in horizontal flow orientation (0°) as compared to vertical downflow orientation (-90°). Moreover, the effect of flow orientation was prominently seen on the performance of tapered manifolds, with tapered manifolds performing poorly against the uniform manifold in the vertical downflow orientation. The predominance of buoyancy force acting on the liquid-vapor interface against the bulk flow direction was seen as cause behind the diminished performance in the vertical downflow orientation. On the other hand, the change in flow orientation had insignificant effect on pressure drop over the entire range of heat flux with similar pressure drop values recorded at both orientations at comparable heat fluxes. Flow could not be sustained in the vertical upflow orientation due to excess vapor accumulation leading to system clogging.

d. A self-sustaining, two-phase thermosiphon loop was developed based on the performance trends from testing of gravity-driven flow boiling system and was tested in OMM. A maximum head flux of 136 W/cm² at 41 °C was obtained with 6% taper. The performance was seen to increase with increase in taper gradient from 4% to 6%. The pressure recovery effect of tapered manifold resulted in a remarkably low pressure drop. The maximum pressure drop was consistently below 3 kPa. The use of OMM with higher taper gradients was a promising pathway for developing more compact cooling modules and enhance heat transfer in future works.

e. The pressure recovery effect of the tapered manifold geometry led to enhancement of fluid flow rate at higher heat fluxes, thereby improving the heat transfer performance. The heat transfer performance of the thermosiphon loop was seen to exceed that of the gravity-driven flow boiling system even with a smaller available head. A flow instability was observed at low heat fluxes.
These can be easily avoided by placing a one-way valve that was commonly employed in many thermosiphon loops to prevent large pressure excursions.

This present study of OMM in a thermosiphon loop confirms its ability to provide high heat dissipation even with a small available head. This configuration can be further explored by varying and optimizing parameters like microchannel geometry, taper gradient, manifold gap height and liquid driving head for further enhancement of heat transfer performance while making the system more compact.
8. FUTURE WORK AND RECOMMENDATIONS

8.1. Manifold Design

In the present study, flow boiling performance of a gravity-driven system was tested at different flow orientations. The inlet and outlet ducts of the manifolds used were primarily designed for testing flow boiling in the horizontal orientation, however, vertical orientations were tested by altering the piping layout in and from the manifold to optimize the flow without drastic pressure drop penalty. However, flow could not be sustained in the vertical upflow orientation and led to clogging of the system due to backflow from the outlet duct and thus, vapor accumulation at low flow rates. Hence, for future testing in the vertical orientations, it is necessary to customize the manifold inlet and outlet duct design to sustain low flow rates at higher heat fluxes and pacify back flows in this orientation.

The pressure recovery achieved with tapered manifolds was instrumental in enhancing fluid flow rate through the thermosiphon loop and eventually the heat transfer performance. Results have shown higher taper gradients improving heat flux and minimizing pressure drop. Thereby, higher taper gradients can prove beneficial in enhancing the performance further in the thermosiphon loop. This low pressure drop configuration can assist in designing a more compact thermosiphon in the future.

8.2. Parametric Study

A heat flux over 100 W/cm² was dissipated with the thermosiphon loop with a small driving head below 0.2 m, promising further improvement in the performance. Parameters like microchannel geometry, manifold gap height, pipe diameter, liquid driving head, flow orientation and system pressure can varied to investigate their effect on performance.
9. REFERENCES


APPENDIX

Appendix A: Ethanol Properties, Flammability and 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.

Auto-ignition of ethanol at high heater temperature may also be concern. Auto-ignition temperature is defined as the lowest temperature at which a substance will spontaneously ignite, without any external source of ignition. The auto-ignition 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.
Appendix B: 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 by [31]. 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. 45(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 45: (a) Heat loss simulation model, (b) Plot of heat loss as a function of wall superheat
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. 45(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. (17),

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

\[ = q^{\text{measured}} - (0.0308\Delta T_{\text{sat}} + 1.675) \quad (17) \]

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.