CPU Cooling in Data Center Using a Thermosiphon Loop with Tapered Open Microchannel Manifold (OMM)

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By: Aranya Chauhan

A Thesis Submitted in Partial Fulfillment of the Requirements for 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

May 3rd, 2017
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ABSTRACT

The efficient cooling of servers in data center offers a unique challenge to reduce the worldwide energy consumption and liquid inventory of working fluid. Presently, single phase cooling techniques are widely used for CPU cooling in data centers. Such techniques are proving to be inefficient, as the heat flux generated in CPU cores is very high, limiting the clock speed of the processors. Also, single phase coolers require external pumping power adding cost to the system. The great potential of a thermosiphon system as a replacement of currently used cooling techniques is studied in the presented work.

A thermosiphon loop using two-phase heat and mass transfer process uses latent heat of the working fluid. The latent heat is much more efficient than sensible heat improving the heat dissipation ability of the system. The thermosiphon system is a gravity driven loop thus reducing the power consumption and the cost of the system. However, the system performance is limited by Critical Heat Flux (CHF) and Heat Transfer Coefficients (HTC). An increase in CHF offers wide temperature operating range while the HTC defines the efficiency of the process. In the proposed design of the cooling solution, a manifold with a taper is employed over the heater surface to guide vapor away from the surface along the flow length. The
incoming liquid flows over the heating surface unobstructed developing separate liquid-vapor pathways.

Two taper angles, 3.4° and 6° in the manifold are tested for the benchtop configuration of the thermosiphon loop. A heat transfer coefficient of 27.3 kW/m²°C and 33.4 kW/m²°C was achieved for 3.4° and 6° taper angles respectively. The heat transfer performance was analyzed with HFE7000 as the working fluid. The performance of the benchtop thermosiphon loop was evaluated for three liquid fill volumes resulting in three different liquid heads available in the thermosiphon loop. Based on the benchtop thermosiphon loop performance a new cooler was designed and built for CPU cooling in RIT’s data center. The performance of the new thermosiphon loop used in CPU cooling was compared with the air and water based coolers currently used in the data center. The maximum CPU temperature achieved for thermosiphon loop was 84.4°C under the stress test. The maximum CPU temperatures for air based and water based coolers were 82.6°C and 63.4°C respectively under the stress test.

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TABLE OF CONTENTS

ABSTRACT ......................................................................................................................... 3

LIST OF FIGURES .............................................................................................................. 8

LIST OF TABLES .................................................................................................................. 11

NOMENCLATURE .................................................................................................................. 12

1. INTRODUCTION ............................................................................................................ 13

1.1 Basics of Pool Boiling Heat Transfer .............................................................................. 17
1.2 Basics of Flow Boiling Heat Transfer .............................................................................. 21
1.3 Thermosiphon loop .......................................................................................................... 23

2. LITERATURE REVIEW .................................................................................................... 25

2.1 Pool boiling in microchannels ......................................................................................... 25
2.2 Flow boiling in microchannels .......................................................................................... 28
   Flow boiling using tapered manifold with open microchannels .......................................... 32
   Bubble expansion in tapered manifold ................................................................................. 35
   Effect of flow rate in tapered manifold configuration ......................................................... 36
2.3 Gravity driven boiling systems ......................................................................................... 37

3. EXPERIMENTAL SETUP ............................................................................................... 40

3.1 Benchtop Thermosiphon ................................................................................................. 40
   Evaporator .......................................................................................................................... 42
   Condenser ........................................................................................................................... 44
   Degassing System ............................................................................................................... 46
3.2 Thermosiphon loop for CPU cooling .............................................................................. 47
   Evaporator .......................................................................................................................... 47
   Condenser ........................................................................................................................... 49
3.3 Air cooler for CPU cooling .............................................................................................. 50
3.4 Water cooler for CPU cooling .......................................................................................... 51
Appendix ........................................................................................................... 89
LIST OF FIGURES

Figure 1: Air cooler schematic ........................................................................................................15
Figure 2: Water cooler schematic ...................................................................................................16
Figure 3: Boiling curve ......................................................................................................................18
Figure 4: Regimes in flow boiling ..................................................................................................21
Figure 5: Thermosiphon schematic ................................................................................................24
Figure 6: Microchannels geometry .................................................................................................25
Figure 7: Pool boiling mechanism in microchannels [3] ...............................................................26
Figure 8: Sintered channels boiling performance - (a) Separate liquid vapor pathways, (b) Boiling curve for different channel size[5] .................................................................27
Figure 9: Different microchannels configuration ............................................................................29
Figure 10: Balasubramanian experimental study - (a) Expanding microchannels, (b) Pressure drop comparison[16] ........................................................................................................30
Figure 11: Expanding microchannels, (a) stepped wall microchannels [17], (b) smooth wall microchannels [18] ................................................................................................................31
Figure 12: Microchannel configurations with dimensions in mm [20] ........................................32
Figure 13: Different manifold configurations - (a) Uniform manifold, (b) Tapered manifold [23] ..............................................................................................................................33
Figure 14: Flow boiling in tapered manifold with microchannels[26] ..................34

Figure 15: Bubble growth in tapered manifold[27]...........................................35

Figure 16: HTC variation with heat flux for different flow rates [27]....................36

Figure 17: Schematic of experimental setup in Noie's work[30] .........................37

Figure 18: Gravity driven systems comparison[31]........................................38

Figure 19: Benchtop thermosiphon loop ..................................................................41

Figure 20: Evaporator in benchtop thermosiphon loop ...........................................42

Figure 21: Microchannels chip – (a) Projected area, (b) Fin and channel
dimensions .........................................................................................................43

Figure 22: Condenser in benchtop thermosiphon loop…………………………………45

Figure 23: Degassing system in benchtop thermosiphon loop.................................46

Figure 24: Evaporator in thermosiphon loop for CPU cooling .............................49

Figure 25: Air cooler for CPU cooling .................................................................51

Figure 26: Copper base in water based cooler for CPU cooling ............................52

Figure 27: Water based cooler for CPU cooling ...............................................53

Figure 28: Data acquisition system in benchtop thermosiphon loop ..................55

Figure 29: One dimensional conduction in copper column ...............................57

Figure 30: Boiling curve comparison for 3.4 and 6 degree taper angles ............64
Figure 31: Heat transfer coefficient (HTC) comparison for 3.4 and 6 degree taper angle..............................................................65

Figure 32: Boiling curve comparison for 175 ml, 250 ml and 325 ml fill volumes 67

Figure 33: Critical heat flux variation for 175 ml, 250 ml and 325 ml fill volumes
........................................................................................................67

Figure 34: Heat transfer coefficient (HTC) comparison for 175 ml, 250 ml and 325 ml fill volumes ........................................................................................................................................68

Figure 35: CPU temperature variation under stress test ........................................70

Figure 36: CPU power consumption under stress test ........................................71

Figure 37: Heat dissipation comparison for air-based, water-based and thermosiphon coolers ...............................................................................................................72
LIST OF TABLES

Table 1: Flow boiling performance of tapered manifolds ........................................33

Table 2: Heat transfer performance of different CPU cooling units .......................73
NOMENCLATURE

CHF  Critical Heat Flux, W/cm²

DT   ambient superheat, °C

HTC  Heat Transfer Coefficient, kW/m²°C

T    temperature, °C

TDP  Thermal Design Power, W

U    uncertainty

P    precision uncertainty

B    bias uncertainty

q"   heat flux, W/cm²

k    thermal conductivity, W/mK

x    distance, m

ρ    dependent property

Δx   thermocouple spacing, m

ΔT   wall superheat, °C

σ    independent variable
1. INTRODUCTION

The current trend in electronics is miniaturization. The devices are getting smaller every year and the performance expectation is increasing. In 1965, Intel co-founder Gordon Moore predicted [1] that the numbers of transistors per square inch on integrated chip will double every year, and this has been happening since then. Following such trends, the performance and size of electronic devices are now constrained by their thermal threshold. The devices have become smaller and hotter than ever before. Developing effective cooling technique to dissipate a large amount of heat from small areas is proving to be a great challenge for engineers worldwide. The present, widely used cooling techniques are not efficient enough and this greatly affects the maintenance of a data center. The data center may have multiple server rooms, each room contains several racks arranged in an array. The CPUs are installed in the racks, which generate large amounts of heat under stressful working conditions. Data centers create and manage the internet connectivity all over the world acting as the backbone of information exchange.

In a data center, server cooling consumes lots of energy and adds significant costs to the data center management. According to the report of Natural Resources Defense Council in partnership with Anthesis, in 2013 about 3 million computer
rooms in data centers used enough electricity equivalent to annual output of 34 large coal-fired power plants. The data centers in the US could reduce their electricity consumption by 40 percent[2], estimating a saving of $3.8 billion and 39 billion kilowatt-hours. According to Cool IT Systems, generally in an air-cooled data center about 40% of total electrical power is used for cooling while remaining 60% is used for servers, networking, storage etc.

Server CPUs in the data center at RIT, Rochester are cooled by small heat sinks installed on the CPUs. Heat is extracted from this heat sink by cold air circulating throughout the room containing multiple racks of servers. The heat sinks presented in this study which are used in RIT’s data center are air and water based coolers. An air based cooler as shown in Fig.1 extracts heat from the CPU surface via copper interface and then heat is transferred through copper heat pipes to an array of metallic fins. The fin array is cooled by forced convection using an attached fan, which is driven by power supply via motherboard.
Figure 1: Air cooler schematic

In a liquid based heat sink (Fig.2), the liquid flows over a copper interface placed over CPU surface similar to air based cooling technique. Water is used as the working liquid. Water flows over the cooper cold plate and is later supplied to an external radiator. The air flow through the radiator is governed by a fan attached to the radiator. Liquid circulation in this type of system is driven by a pump which increases power consumption adding cost to the technique. The currently used cooling techniques are not efficient enough and adds cost to the system maintenance. An attractive solution to dissipate large heat quantity from CPUs in data center is two phase heat and mass transfer process - boiling. Boiling is much more efficient than single phase heat transfer because it stores heat in the form of latent heat resulting in low chip surface temperatures.
Figure 2: Water cooler schematic
1.1 Basics of Pool Boiling Heat Transfer

Boiling is a two-phase heat and mass transfer process. The high heat transfer coefficients make it an attractive option for electronics cooling. A brief overview of the boiling process is given in this section.

A pool of liquid is converted into vapor by continuous heating as the liquid reaches its saturation temperature. At this point, the heating surface’s temperature is greater than the saturation temperature of liquid. The degree of wall superheat (ΔT) is defined by the temperature difference between the heating surface and the saturation temperature of liquid. The amount of heat dissipated from the heating surface per unit area is defined as heat flux and is measured in W/cm² or W/m². For electronics cooling applications, W/cm² is widely used since small surface areas are involved. The various regimes in boiling as shown in Fig.3 are – Natural convection, nucleate boiling (partial and developed), transition boiling and film boiling. The boiling curve is dependent on the liquid used and the nature of heating surface.
**Natural convection** – The natural convection region is from the origin of the graph in Fig.3 to point ‘a’. Initially, at low heat flux, the heating surface attains a small degree of wall superheat. The liquid near the heating surface is at a higher temperature than the liquid in bulk. This temperature difference develops a density gradient along the liquid column. The liquid starts circulating due to buoyancy - hot liquid from the heating surface moves upward, and cold liquid occupies space on the heating surface. This mode of single-phase convective heat transfer sustains till point ‘a’.
**Nucleate boiling** – At point ‘a’ the first bubble nucleates from the heater surface and the point is called onset of nucleate boiling (ONB). The region from point ‘a’ to ‘b’ is partially developed nucleate boiling region. The bubbles nucleate from various active nucleating sites on the heater surface, and bubble nucleation frequency increases with continuous increase in heat flux. Point ‘b’ is the transition stage from partially to developed nucleate boiling. At the transition point, bubbles coalesce vertically to form continuous vapor jets. From point ‘b’ to ‘c’ is developed nucleate boiling. Bubbles start coalescing horizontally to form the mushroom like structures with multiple vapor stems attached to the heating surface. At point ‘c’ critical heat flux (CHF) is achieved resulting in a sudden increase of surface temperature as shown by the dotted line from ‘c’ to ‘e’.

**Film boiling** – At CHF a film of vapor is formed due to high bubble nucleation frequency and significant coalescence of bubbles in the horizontal direction. This vapor film acts as an insulator thereby increasing the surface temperature. At this stage radiation heat transfer becomes dominant. By decreasing the heat flux, the plot can be traced from point ‘e’ to point ‘d’. This is the point of minimum heat flux also
called Leidenfrost point. Decreasing heat flux further results in a sudden drop of wall superheat and state of nucleate boiling is attained as shown by the dotted line from point ‘d’.

**Transition boiling** – The transition boiling region is achieved by controlling the wall superheat, unlike other boiling regimes where heat flux was controlled input parameter. The bubbles occupy a large surface area in this region as shown from point ‘d’ to point ‘c’.

The nucleate boiling region is the desired operating region for the cooling application. The steep nature of graph makes heat transfer process highly efficient as high heat flux can be dissipated with low increment in wall superheat.
1.2 Basics of Flow Boiling Heat Transfer

In flow boiling, the liquid is pumped through the heating channel by an external pump. Bubbles in this type of boiling, flow along the liquid due to liquid inertia forcing the bubble to move along the channel, unlike pool boiling where buoyancy force was dominant. Flow boiling process in a channel contains different regions characterized by relative liquid vapor amount. The various regimes in flow boiling are shown in Fig.4

![Figure 4: Regimes in flow boiling](image)
The various flow regimes in flow boiling are described below

1. **Forced Convection (Liquid Flow)** - The liquid enters the channel as shown in Fig. 2 and as it travels through region 1 forced convective heat transfer takes place as external power is supplied by a pump.

2. **Bubbly Flow** – The continuous supply of heat flux activates bubble nucleation at the heating surface. Bubbles travel along the wall surface due to forced convection approaching fluid core under the influence of buoyancy as well.

3. **Slug Flow** - Liquid is continuously heated along the channel length due to which liquid core also heats up. This results in bubble expansion and finally bubbles coalesce forming vapor slugs.

4. **Annular Flow** – The continuous bubble growth and accumulation forms a vast vapor space in channel’s core and liquid is confined to the spaces between heating surface and vapor.

5. **Transition Flow** – The ongoing growth of vapor space in core of channel restricts the liquid to spaces very close to heating surface and eventually liquid layer diminishes. Also, few drops of liquid are observed in the vapor space in form of mist.

6. **Mist Flow** – Majority of channel space is occupied by vapor and liquid flow in the form of mist.
7. **Forced Convection (Vapor Flow)** – Eventually fine liquid droplets also evaporate, and channel space is completely occupied by vapor phase.

1.3 **Thermosiphon loop**

The application of thermosiphon loop with two-phase heat transfer (boiling) as a potential replacement of currently used cooling techniques is discussed in the presented work. Thermosiphon loop is a self-sustaining system containing a heating section and a condenser as shown in Fig.5. The condenser also acts as a reservoir for the working liquid. Initially, the working liquid is filled in the condenser and then supplied to the heating section due to the gravitational head. Working liquid flows over the heating surface and boiling heat transfer initiates at a certain heat flux and wall superheat. Since heat is stored in the form of latent heat, a small temperature difference is maintained between the heating surface and the working fluid. As the liquid flows over the heated surface, bubbles are generated, and a vapor column is established at the exit. Vapor rises vertically upwards into the condenser. Condenser extracts the heat from the vapor and changes it back into liquid which is further resupplied to the heating section. The system uses two-
phase heat transfer process which is more efficient than single phase process, and no external pump is required for fluid circulation in the loop. This can reduce the power requirement and the cost of the system compared to the currently used cooling techniques.

*Figure 5: Thermosiphon schematic*
2. LITERATURE REVIEW

2.1 Pool boiling in microchannels

Microchannels as shown in Fig. 6 is a parallel alternate arrangement of fins and channels. The boiling in microchannels is a very effective process compared to plain surface due to high surface area to volume ratio.

![Microchannels geometry](image)

*Figure 6: Microchannels geometry*

Cooke and Kandlikar [3,4] presented the bubble growth and departure mechanism in their experimental study as shown in Fig.7. This suggests a liquid-vapor interaction during boiling in microchannels making it an efficient mode of heat transfer. The bubble nucleates at the bottom surface and moves to the channel side wall. The bubble moves to fin top and grows completely before departure from fin top. This allows constant rewetting of the bottom surface. Critical heat flux (CHF) of 244 W/cm² was achieved using a microchannel chip with a heat transfer coefficient (HTC) of 269 kW/m²K [3].
Jaikumar and Kandlikar [5] achieved a CHF of 420 W/cm\(^2\) at wall superheat of 1.7°C with HTC to be 2.9 MW/m\(^2\)°C using sintered microchannels. Channel sizes – 300 µm, 500 µm and 762 µm were studied. The best boiling performance was observed for 300 µm as shown in Fig.8 (b). The separate liquid-vapor pathway was proposed as the heat transfer mechanism responsible for high HTC. The vapor column rises from the channel whereas liquid is resupplied from fin tops into the channels as shown in Fig.8 (a).
Figure 8: Sintered channels boiling performance - (a) Separate liquid vapor pathways, (b) Boiling curve for different channel size[5]

Other surface enhancements like tall porous structures[6,7], bi-conductive configuration[8], pores and tunnels[9], nano-micro ridges[10,11] and wicking microstructures [12,13] are also proven to be efficient ways to increase the heat transfer performance.

Pool boiling proves to be an attractive solution for high heat dissipation while maintaining very low surface temperatures but requires significant working fluid inventory.
2.2 Flow boiling in microchannels

Flow boiling is a heat and mass transfer process where liquid flows over a heated surface driven by an external pump. Although, single phase heat transfer as studied by Colgan et al.[14] can dissipate heat fluxes of over 1kW/cm² but it also results in high chip temperature and large pumping power is required to drive the working fluid. Hence flow boiling is a very promising field for such applications. The surface containing microchannels enhances the boiling performance due to the high surface area to volume ratio provided by microchannels for heat transfer[15]The microchannel chip configuration can be open or closed type as shown in Fig.9. The open microchannels have space available over the fin top whereas in closed microchannels a cover plate is placed on the fin surface. The closed microchannels allow independent fluid flow in channels. Vapor generated during boiling resists the flow of liquid at the inlet in closed type microchannels configuration. Whereas open microchannels provide extra space in the vertical direction for smooth bubble flow, thus developing more stable and efficient heat transfer process.
Balasubramanian et al. [16] used closed expanding type of microchannels in their flow boiling study as shown in Fig. 10(a). Heat flux of 120 W/cm² was achieved at a surface temperature 122°C with expanding microchannels compared to 128°C in the case of straight microchannels for the same heat flux. The maximum pressure drop decreased from 0.037 bar to 0.015 bar using expanding microchannels offering more stability and less pumping power. The expanding microchannels provided extra channel space along flow length helping in bubble expansion along channels.
Mukherjee and Kandlikar [17] in their numerical study proposed a concept of stepped wall microchannels providing increasing cross sectional area along the flow path to minimize reversible flow. Further, a smooth diverging microchannel [18] was proposed considering the ease in manufacturing process.
Lu and Pan [19] conducted an experimental flow boiling study using 10 microchannels with diverging cross sections. The mean hydraulic diameter of each channel was 120 µm and with a uniform depth of 76 µm. The channels had a diverging angle of 0.5°. It was concluded that diverging microchannels have superior stability than uniform cross section microchannels.

Single phase heat transfer process in diverging microchannels studied by Prajapati et al.[20] compared the performance of uniform, diverging and segmented microchannels configuration as shown in Fig.12. The hydraulic diameter for all microchannel configurations was 522 µm and number of channels 12. The segmented microchannels provided the highest heat transfer coefficient (HTC) for
mass fluxes 130 kg/m²s, 194.7 kg/m²s, 260 kg/m²s and 324.5 kg/m²s compared to uniform and diverging configuration. The highest HTC ~ 14000 W/m²K was achieved for mass flux 130 kg/m²s.

Figure 12: Microchannel configurations with dimensions in mm [20]

Flow boiling using tapered manifold with open microchannels

Kalani and Kandlikar [21,22] studied flow boiling and showed the surface containing microchannels with a tapered manifold as shown in Fig. 9(b) achieved higher heat flux and heat transfer coefficient compared to uniform manifold Fig.9(a).
Figure 13: Different manifold configurations - (a) Uniform manifold, (b) Tapered manifold [23]

Heat transfer coefficient of about 277.8 kW/m²°C was achieved using a tapered configuration. Three manifold types were studied and a 6% taper provided the best performance with CHF 281.2 W/cm² with pressure drop 3.3 kPa. The performance comparison for three different manifold types is shown in Table 1.

Table 1: Flow boiling performance of tapered manifolds

<table>
<thead>
<tr>
<th>Manifold</th>
<th>Heat Flux (W/cm²)</th>
<th>Wall Superheat (°C)</th>
<th>Pressure Drop (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td>283.2</td>
<td>13</td>
<td>62.1</td>
</tr>
<tr>
<td>Taper A (2%)</td>
<td>265</td>
<td>14</td>
<td>7.5</td>
</tr>
<tr>
<td>Taper B (4%)</td>
<td>239.1</td>
<td>8.6</td>
<td>6</td>
</tr>
<tr>
<td>Taper C (6%)</td>
<td>281.2</td>
<td>10.1</td>
<td>3.3</td>
</tr>
</tbody>
</table>
Also, the microchannel chip design over conventional plain chip design reduces the pressure drop across the channel length significantly [23]. Reduction in the channel length and number of channels in microchannel chip also lead to more stable flow [24]. The plain chip shows no effect of increasing taper height on the heat flux and heat transfer coefficient. But the boiling performance improves with taper manifold compared to uniform design. The tapered manifold configuration show improved heat flux, heat transfer coefficient and less pressure drop. The focus of this section is on the tapered microchannel design.

The uniform manifold compared with tapered manifold shows similar performance [21,22] at low heat flux dissipation values. But at higher heat flux values tapered manifold outperforms the uniform manifold [22,25]. The tapered manifold shows reduction in pressure compared to the uniform manifold.

*Figure 14: Flow boiling in tapered manifold with microchannels*[26]
**Bubble expansion in tapered manifold**

The vapor bubble in the uniform manifold microchannel expands on the microchannel surface and causes dry out state. This dry out results in high pressure drop, low heat transfer coefficient and early CHF state [27]. The tapered manifold is a type of design which provides a much smoother flow of vapor-liquid combination by creating separate liquid vapor flow paths. The increasing cross section area allows bubble growth in the vertical direction thus avoiding a dry out condition to a large extent. The bubble in this type of manifold flows above the liquid region because of a density difference since taper manifold provides much extra vertical space compared to the uniform manifold. Also, such type of bubble growth and flow provides nucleating sites for new bubbles under fully developed bubbles (Fig.15), thus making the bubble formation continuous and faster[28]

![Figure 15: Bubble growth in tapered manifold](image-url)
**Effect of flow rate in tapered manifold configuration**

Kalani and Kandlikar[27,28] explained that quick removal of bubbles can also be achieved by increasing the Reynolds number. The CHF of 1.1 kW/cm² was achieved at wall superheat of 43°C for Reynolds number 1642 using a 6% tapered manifold with microchannels. Increasing the Reynolds number provides more inertia to the liquid enabling liquid to carry bubbles more efficiently. But as the mass flux increases it results in larger pressure drop hence taper manifold is very beneficial as it provides pressure recovery. High Reynolds number may cause vapor blanketing. Vapor blanketing is a state where high liquid inertia forces prevent the bubbles from emerging into the taper region thus decreasing the heat transfer performance as shown in Fig.16

![Figure 16: HTC variation with heat flux for different flow rates [27]](image)
2.3 Gravity driven boiling systems

Buchling and Kandlikar [29] used a tapered manifold over copper chip containing microchannels and were able to dissipate 217 W/cm$^2$ at the wall superheat of 34°C. The pressure recovery along the flow length due to a continuous increase in cross section area helps in developing a smooth flow. The reduction in instabilities helps in improving the efficiency of the system.

Noie[30] conducted an experiment with an integrated thermosiphon loop setup (Fig.17) – the heating section and condenser are present in one unit. For aspect ratios 7.45, 11.8 maximum heat transfer rate and least mean evaporator temperature were obtained at 90% and 60% filling ratios. It was concluded that heat transfer rate and mean evaporator temperature depend on aspect ratio and filling ratio.

![Figure 17: Schematic of experimental setup in Noie’s work][30]
Panse and Kandlikar[31] used a 6% tapered manifold with a microchannels copper chip in a gravity driven flow boiling system. The critical heat flux (CHF) of 136 W/cm² at wall superheat of 42.7°C with ethanol as working liquid was achieved. They maintained a very stable fluid flow with pressure drop 4kPa near CHF. The performance of thermosiphon loop was compared with gravity driven flow boiling system as shown in Fig.18

![Figure 18: Gravity driven systems comparison][31]

Palm and Khodabandeh[32] and Furberg et al.[33] showed the dependency of heat transfer coefficient on reduced pressure. It was concluded that heat transfer coefficient increases with an increase in reduced pressure as at higher pressure more cavities are activated which enhances nucleate boiling. Multiple evaporator
experimental study containing rectangular channels with hydraulic diameters varying from 1.2 – 2.7mm concluded that larger diameter channels generally perform better.

Webb and coworkers [34,35] used a thermosiphon system with R-134A and water, heat transfer coefficient values over 60 kW/m²°C were achieved. Porous metallic coatings were used by Tuma[36] in a thermosiphon system to achieve heat transfer coefficient~ 100 kW/m²°C. But the performance was decreased with larger heaters. Moura et al.[37] used circular cavities in a thermosiphon system and during the stress test they were able to reduce the CPU temperature by 26°C.
3. EXPERIMENTAL SETUP

The experimental study for thermosiphon loop is conducted for two types of setups. The first type of setup is benchtop thermosiphon where a thermosiphon loop is tested in a controlled environment. In the second type of thermosiphon setup, the system is built for actual CPU cooling application. The loop is built such that it can be mounted on the motherboard of the server. The two types of cooling units – air based and water based also setup on the mother board for respective experiments. The air based cooler, water based cooler and second type of thermosiphon loop are tested on the same CPU server.

3.1 Benchtop Thermosiphon

The benchtop thermosiphon system used is similar to the setup used by Panse and Kandlikar [31]. The benchtop thermosiphon loop is set external to the CPU cooling setup, to study heat transfer process in a controlled environment. The major sub-systems in the benchtop thermosiphon loop are – 1) Evaporator, 2) Condenser, and 3) Degassing system. The base of the condenser is kept at the height of 0.2 m from the heating surface in the evaporator. The schematic of benchtop thermosiphon loop is shown in Fig.19.
Figure 19: Benchtop thermosiphon loop
Evaporator

The evaporator contains a heating block and tapered manifold along with other components as shown in Fig. 20.

![Diagram of evaporator](image)

Figure 20: Evaporator in benchtop thermosiphon loop

A cube shaped copper block is used as the heating unit. The copper block is heated by joule heating using four cartridge heaters (120 Watts each) inserted in each face of the block. The four cartridge heaters are connected to an external TDK – Lambda DC power source via a junction box. The copper block has an integrated chip containing microchannels at the top. The heat is conducted to the copper chip through a copper column having cross section 10 mm x 10 mm. Three calibrated
K-type thermocouples are inserted 5 mm inside the copper column placed equidistantly, 3 mm apart. The chip dimensions are 15 mm x 15 mm and microchannels projected area is 10 mm x 10 mm. The microchannel’s fin width, channel depth and channel width are 200 µm each. The chip surface not containing the microchannels is insulated using a teflon tape thus enabling heat transfer through 10 mm x 10 mm projected area of microchannels only.

![Diagram of microchannels chip](image)

Figure 21: Microchannels chip – (a) Projected area, (b) Fin and channel dimensions

A manifold with certain degree of taper is mounted over the microchannel chip. The tapered section in the manifold provides 100 mm² projected area over the microchannels for fluid flow. The two types of manifold used for various tests had 3.4° and 6° taper angle along the flow length of 10 mm. The 3.4° and 6° taper provides additional 5.94 mm² and 10.5 mm² cross sectional area at the exit of flow length (10 mm) compared to inlet. The inlet and outlet of manifold are inclined at
45° with the manifold base. Quarter inch union fittings are used at both ends of the manifold for extended pipe connections. A highly compressible silicone gasket of thickness 200 µm was used to seal the interface between copper chip and manifold. The manifold is made up of polysulfone material, it’s glass transition temperature is 185°C and can be easily machined.

**Condenser**

A cylindrically shaped condenser as shown in Fig.22 is used in the benchtop thermosiphon loop made up of 7 mm thick steel. The inner diameter of the condenser is 91 mm and height is 160 mm. A copper coil with outer and inner diameter to be 3.3 mm and 1.5 mm respectively is used as the heat exchanger. The coil contains 12 turns with each turn’s internal diameter being 30.5 mm. The heat exchanging copper coil is connected to an external chiller. The chiller supplies water at a constant temperature of 15°C to the copper coil for the entire test duration for all experiments. A pressure gauge and K-type thermocouple are mounted on the top surface of the condenser. The thermocouple is extended till the bottom of the condenser to measure liquid’s inlet temperature. Two valves are also attached at the top surface. One valve is used to feed the working liquid inside the chamber before each experiment and the other valve is used for connecting condenser chamber to the degassing sub-system.
The condenser is sealed at the bottom using a steel cover plate with an O-ring at the interface. The bottom surface is attached to main condenser body using four aluminum double clamps. The double clamps are symmetrically placed along the circular periphery for even pressure distribution. The attached surface at the bottom contains two ports, one each for downcomer and riser. The riser is extended up to 81 mm inside the condenser with a bend. The bend prevents the condensed liquid from copper coil from escaping into the riser.

*Figure 22: Condenser in benchtop thermosiphon loop*
**Degassing System**

The degassing system as shown in Fig.23 includes a liquid-vapor separator, a centrifugal pump and a fume hood. The working liquid may contain dissolved gases which alters the boiling performance thus degassing is performed before experiment. The outlet from the condenser is connected to a liquid-vapor separator. The liquid-vapor mixture in the separator is segregated due to buoyancy and vapor is driven away by the centrifugal pump to an external fume hood.

*Figure 23: Degassing system in benchtop thermosiphon loop*
3.2 Thermosiphon loop for CPU cooling

The computer engineering department of the Rochester Institute of Technology provided the server computers used in RIT’s data center. The central processing unit (CPU) is an Intel i7 processor, i7-930 and has high thermal design power of 130 W. A thermosiphon loop is suggested as a potential replacement for the existing water and air based cooling techniques.

**Evaporator**

The heat from the CPU is transferred to an external evaporator assembly similar to the benchtop thermosiphon’s evaporator. The schematic, CAD assembly’s exploded view and final machined evaporator assembly used in CPU cooling are shown in Fig.24. A copper interface containing microchannels with a projected area of 32 mm x 38.5 mm is placed between the CPU and the tapered manifold. The microchannel chip contains a thin circular projection at the bottom making whole chip thickness equal to 7 mm. The microchannels’ fin width, channel depth and channel width are 200 µm each. The channels are machined by a CNC machine on the top face of the copper interface. A tapered manifold made up of Lexan material is used to contain and guide the working fluid over the heating surface. Two taper angles 3.4° and 6° are tested and the performance is evaluated later in the results.
section. The tapered section has a projected area of 38.5 mm x 32 mm with a continuous taper along the 38.5 mm length. The taper provides an additional 72.96 mm² and 129.4 mm² cross sectional area for 3.4° and 6° taper angles respectively at exit over the flow length of 38.5 mm.

A machined aluminum mounting ring is used to hold the manifold over the copper interface. The copper interface, gasket, mounting ring and tapered manifold are held together using #4-40 x 3/8” socket cap screws arranged in a circular pattern. The tapped holes with each having diameter 2.8 mm can be seen in manifold Fig.24(b) The evaporator assembly is installed on the motherboard using the mounting bracket.
Condenser

The condenser of thermosiphon loop for CPU cooling was same as the one which was used for benchtop thermosiphon system.
3.3 Air cooler for CPU cooling

The air cooler used for the baseline tests and comparison is a 296 g, 2.4 W SilenX EFZ-80HA2 air-based CPU cooler. This air cooler is used in RIT’s data center for server cooling. The base of the air cooler is placed on the CPU’s top surface for heat conduction. High quality thermal paste – arctic silver, is used to minimize the contact resistance. The base dimensions of the air cooler are 35.5 mm x 45 mm for a CPU surface measuring 32 mm x 34.5 mm. The base of the air cooler contains a 4 x 8 array of fins over a projected area of 35.4 mm x 35.4 mm. Each fin’s width, thickness and height measurements 4.1 mm, 1.5 mm and 10 mm respectively. Heat from the air-cooler’s base is dissipated by this fin array and also the majority of heat the is transferred to another group of fins arranged linearly over the cooler base using four copper heat pipes, each having 6 mm diameter. This fin arrangement has a projected area of 86.7 mm x 76.7 mm containing 36 fins with each fin thickness to be 0.6 mm. The vertical group of fins is cooled by an external 80 mm diameter fan consuming 2.4 W from motherboard.
3.4 Water cooler for CPU cooling

A 2.5 W Dell Alienware Area 51 W550R PP749 water-based CPU cooler is used for comparison and baseline tests. This water based cooler is used for server cooling at RIT’s data center. The base of the cooler has a cooper surface which is placed over the CPU surface (32 mm x 34.6 mm). Arctic silver thermal paste is used to minimize the contact resistance between cooler base and the CPU. The copper surface’s shape and dimensions for the water cooler are shown in figure Fig.26.
The water is driven over the copper base using a pump. The pump and base are integrated as single unit as shown in Fig.27. The cold-water flows over the copper surface inside the integrated unit and hot water is supplied to a radiator containing a fin array. The radiator has a projected area 110 mm x 118 mm with fin arrangement as shown in Fig.27. The radiator has 12 columns of fins with each column having 98 fins. The total surface area of fins is 1478 cm². The thickness of each fin is 0.2 mm. A fan is used to force air through the radiator via fin array for cooling hot-water supplied from heated copper surface. The cooler consumes a total of 2.5W power from motherboard.
Figure 27: Water based cooler for CPU cooling
4. EXPERIMENTAL PROCEDURE

The experimental procedure is explained for two setups. The first setup is a benchtop thermosiphon loop where the experiment is conducted in a controlled environment. The second set is a thermosiphon loop on the actual server CPU used in the RIT data center. The CPU cooling test is executed in two phases – a) baseline test and, b) stress test

4.1 Benchtop thermosiphon

- The desired volume of the working liquid is filled in the condenser. Three different volumes tested are 175 ml, 250 ml and 325 ml. The liquid height in the condenser was kept below the exit of the riser. The working liquid used in the presented work is a refrigerant - HFE 7000.

- The whole test setup is operated by a Labview VI program using National Instruments cDAQ – 9174 and MOD – 9211 at the interface (Fig.28) between test setup and Labview program.

- The main DC power supply is switched on and program is initiated in Labview. Initially 4 Watts power is supplied to the four cartridge heaters inserted in copper heater block for degassing.
• The valve connecting degassing subsystem and condenser is opened and the pump is switched on. The degassing is performed for 20 minutes, the valve is closed and pump is switched off.

• The voltage from DC power supply is increased in the steps of 2V initially and later at higher heat fluxes the step size was reduced to 1V. A smaller step size was used at higher heat fluxes to trace more accurate boiling curve. The data was recorded at steady state for 10 seconds with step size of 0.2 sec.

• The experiments were conducted till critical heat flux (CHF) was achieved.

![Diagram of the degassing system](image-url)

*Figure 28: Data acquisition system in benchtop thermosiphon loop*
4.2 Data center CPU cooling using thermosiphon

The CPU cooling test is executed in two major steps. Initially baseline test is conducted to develop power consumption plots when CPU is under no stress. The stress test is conducted later, where normalized power consumed by CPU under stress is calculated comparing with baseline.

**Baseline Test**

The power is supplied to CPU tower through an uninterruptible power source (UPS) which displays the power consumption by whole system. The CPU tower contains CPU, graphics processing unit (GPU) and cooling system as the main power consuming parts. Baseline testing is done by turning the system on and not putting any stress on the CPU. The power consumption is recorded from the UPS for 3 hours for all the cooling setups tested. The three different types of systems tested as explained in Section 3 are - 1) Air based cooler, 2) Water based cooler, and 3) Thermosiphon loop

**Stress Test**

The baseline test provides power consumption under no CPU stress condition. CPU is brought to its extreme performing capability by initiating a stress test. The stress test is performed by a stress package executed in Linux. The code simply calculates
multiple roots continuously to put CPU under stress thus increasing the temperature of all four cores. The stress test is performed for all three cooling configurations.

4.3 Data Reduction

The data reduction is done for benchtop thermosiphon experimental study and CPU cooling tests. The three CPU coolers — air cooler, water cooler and thermosiphon cooler data is reduced based on baseline and stress tests.

Benchtop thermosiphon

The Data reduction here was the same as that used by Panse and Kandlikar[31].

![Figure 29: One dimensional conduction in copper column](image)

Heat flux through the copper block was determined through Fourier’s law for 1D conduction:
\[ q''_{\text{block}} = -k_{\text{Cu}} \frac{dT}{dx} \]  

(1)

Where, \( q''_{\text{block}} \) is the heat flux, \( k_{\text{Cu}} \) is the thermal conductivity of copper and \( \frac{dT}{dx} \) is the temperature gradient along copper column.

The temperature gradient was found using Taylor’s backward difference formula as expressed in Eqn. (2).

\[ \frac{dT}{dx} = \frac{3T_1 - 4T_2 + T_3}{2\Delta x} \]  

(2)

The three temperatures - \( T_1, T_2, \) and \( T_3 \) used to find the temperature gradient correspond to three thermocouples in the copper column shown in Fig.29 and \( \Delta x \) is the spacing between thermocouples, 3 mm. A previously performed heat loss analysis based on superheat is included in the data reduction [38]. Heat loss for each data point is calculated using the respective wall superheat values. Heat loss is deducted from the calculated heat flux value to determine the effective heat flux.

\[ q''_{\text{effective}} = q''_{\text{block}} - q''_{\text{loss}} \]  

(3)

The surface temperature \( (T_s) \) is then derived from the effective heat flux, top thermocouple temperature, and the distance between the top thermocouple and the surface, \( x_1 \).

\[ T_s = T_1 - \frac{q''_{\text{effective}}}{k_{\text{Cu}}} x_1 \]  

(4)
Finally, the heat transfer coefficient ($HTC$) is found using the heat flux and wall superheat, $\Delta T_{sat}$ i.e. the temperature difference between the surface and the fluid saturation temperature.

$$HTC = \frac{q''_{effective}}{\Delta T_{sat}}$$ (5)

To compare the results of the benchtop thermosiphon to the CPU coolers, a theoretical CPU temperature is derived from the surface temperature found with the benchtop thermosiphon and thermal resistances associated with the thermosiphon CPU cooler. There are two resistances accounted for: resistance from conduction through the copper chip, and a bulk resistance resulting from contact as well as internal CPU packaging. This bulk resistance is found experimentally to be roughly 2.7 cm$^2$K/W by measuring the temperature of the copper protrusion on the chip, and then using the CPU temperature and heat flux to determine this value. The theoretical CPU temperature is then calculated as follows:

$$T_{CPU} = T_s + q''_{effective}(R''_{Bulk} + \frac{x_2}{k_{Cu}})$$ (6)

Where $R''_{Bulk}$ is the combined CPU packaging and contact resistance, and $x_2$ is the 7 mm distance heat is conducted across in the copper plate of the thermosiphon CPU cooler.
**CPU cooling stress tests**

The temperatures of four CPU cores and the power consumption by the server are analyzed using a MATLAB code. The average of the four CPU core temperatures is used to plot the final heat dissipation curve. It was not possible to determine the exact power consumption only by the CPU thus the value is estimated based on the total power consumption. The average power consumption over a 3-hour period, while the CPU was under no load, is found first. This average power is subtracted from the total power consumption during the stress test to estimate the power draw of the CPU.

**4.4 Uncertainty Analysis**

The uncertainty accounts for all the errors associated with experimental data. The thermosiphon study is conducted for two configurations, the benchtop system and server CPU cooling. The uncertainties for both the setups are explained in the section.
**Benchtop Thermosiphon**

Uncertainties associated with the experimental data for benchtop thermosiphon loop are calculated. The formula used to compute uncertainty values is given below:

\[
U_p = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial p}{\partial \sigma_i} U_{\sigma_i} \right)^2}
\]  \hspace{1cm} (7)

Where \( p \) is any property dependent on independent variable \( \sigma \) over \( n \) samples.

Both precision and bias errors were taken into account.

The total uncertainty associated with heat flux, \( q'' \) is calculated using the following Eqn. (8), where \( \alpha = 3T_1 - 4T_2 + T_3 \)

\[
\frac{U q''}{q} = \sqrt{\frac{U_{kCu}^2}{k_{Cu}^2} + \frac{U_{\Delta x}^2}{\Delta x^2} + \frac{9U_{T_1}^2}{\alpha^2} + \frac{16U_{T_2}^2}{\alpha^2} + \frac{U_{T_3}^2}{\alpha^2}}
\]  \hspace{1cm} (8)

The total uncertainty in measuring the heat transfer coefficient was calculated using the following Eqn. (9)

\[
\frac{U_h}{h} = \sqrt{\frac{U^2_{q'}}{q'^2} + \frac{U^2_{T_s}}{\Delta T_{sat}^2} + \frac{U^2_{T_{sat}}}{\Delta T_{sat}^2}}
\]  \hspace{1cm} (9)

The total uncertainty was calculated from bias and precision errors,
\[
\frac{U_{\Delta p}}{\Delta p} = \sqrt{B_{\Delta p}^2 + P_{\Delta p}^2}
\]  

(10)

All uncertainty computations were done for 95% confidence interval. The uncertainties associated with critical heat flux and heat transfer coefficient are shown as error bars in plots presented in results section.

**CPU Uncertainty**

The temperature uncertainty of the CPU was determined by the standard deviation of the temperatures while the CPU was under no load. As a validation of the CPU heat generation, the values found in stress testing were compared to the Thermal Design Power (TDP), i.e. the maximum heat generated, of the CPU used here, and Intel Core i7-930. The TDP value for this CPU is 130 W, correlating to 11.78 W/cm\(^2\) over the 32x34.5 mm\(^2\) chip surface. The maximum heat flux found in the stress tests performed here was 11.94 W/cm\(^2\), very similar to the theoretical maximum.
5. RESULTS
The results for benchtop thermosiphon loop and thermosiphon for CPU cooling is presented in this section. The benchtop thermosiphon loop results are presented for two different tapered manifolds and later pressure head study results are also presented. The CPU cooling study includes baseline and stress test results. Two single phase coolers performance is compared with thermosiphon loop for both tests.

5.1 Benchtop Thermosiphon

The benchtop thermosiphon loop testing is done in two phases. In phase 1 two taper angles in the manifold 3.4° and 6° are tested. Respectively heat transfer performance was evaluated based on the heat flux dissipation limits and heat transfer coefficient values. Whereas, in phase 2 the boiling performance was studied for three liquid filling volumes in the condenser. The three filling volumes studied are 175 ml, 250 ml and 325 ml for a 6° tapered manifold.

Boiling performance of refrigerant HFE 7000 for constant volume

The boiling performance of refrigerant HFE 7000 is evaluated in this section. The constant fill volume 175 ml is used for all experiments. The total head between liquid’s top surface in the condenser and microchannel chip in evaporator is 0.227
m. The two tapered manifolds tested had 3.4° and 6° taper angles for flow length 10 mm. The two taper angles, 3.4° and 6° provide additional 0.59 mm and 1.05 mm height at the exit.

**Effect on critical heat flux (CHF)**

The highest critical heat flux was achieved 44 ± 8 W/cm² at wall superheat of 16.2°C for 3.4° taper angle. The critical heat flux for 6° degree taper was 34.4 ± 2 W/cm² at wall superheat of 10.3° C. Fig.30 shows the compared boiling curves of 3.4° and 6° tapered manifolds.

![Figure 30: Boiling curve comparison for 3.4 and 6 degree taper angles](image-url)
Effect on heat transfer coefficient (HTC)

The highest heat transfer coefficient was achieved $33.4 \pm 2$ kW/m$^2$ °C at CHF for 6° taper angle. At CHF, 3.4° degree tapered manifold attained heat transfer coefficient of $27.1 \pm 5.2$ kW/m$^2$ °C. Fig.31 shows the heat transfer coefficient variation with heat flux for 3.4° and 6° tapered manifolds.

![Heat transfer coefficient (HTC) comparison for 3.4 and 6 degree taper angles](image)

**Figure 31: Heat transfer coefficient (HTC) comparison for 3.4 and 6 degree taper angles**
**Boiling performance of refrigerant HFE 7000 for different filling volumes**

The boiling performance of benchtop thermosiphon loop is evaluated for different filling volumes in the condenser using a 6° taper angle in the manifold. The three volumes studied are 175 ml, 250 ml and 325 ml developing total head of 0.227 m, 0.239 m and 0.251 m respectively at the manifold inlet. The performance is studied based on the respective boiling curves and heat transfer coefficient variation with heat flux.

**Effect on critical heat flux (CHF)**

The highest critical heat flux achieved was 40.5 ± 6 W/cm² at wall superheat of 17.8° C for 325 ml fill volume. The critical heat flux for 175 ml and 250 ml were 34.4 ± 2 W/cm² and 36.9 ± 3.2 W/cm² at wall superheat of 10.3° C and 16.4°C respectively. Fig.32 shows the different boiling curves for respective fill volumes. The linear increment in critical heat flux with increasing fill volume is shown in Fig.33.
Figure 32: Boiling curve comparison for 175 ml, 250 ml and 325 ml fill volumes

Figure 33: Critical heat flux variation for 175 ml, 250 ml and 325 ml fill volumes
**Effect on heat transfer coefficient (HTC)**

The highest heat transfer coefficient was achieved $33.4 \pm 2$ kW/m$^2$ °C at CHF for 175 ml fill volume. At CHF, 250 ml and 325 ml fill volumes attained heat transfer coefficients of $22.5 \pm 2$ kW/m$^2$ °C and $22.7 \pm 3.9$ kW/m$^2$ °C. Fig.34 shows the heat transfer coefficient variation with heat flux for different fill volumes.

![Figure 34: Heat transfer coefficient (HTC) comparison for 175 ml, 250 ml and 325 ml fill volumes](image)
5.2 Data center CPU cooling using thermosiphon loop

The data center’s CPU was tested using three different cooling techniques, which are fan cooling, liquid cooling and thermosiphon loop. The fan cooling and liquid cooling units are presently used in data centers. Thermosiphon loop was built for CPU based on the preliminary tests conducted on benchtop configuration. A baseline test was conducted to measure the amount of power consumed by CPU tower under no stress condition. The baseline test was conducted for all three cooling units.

The thermosiphon test was initially conducted for 6° taper angle. The 6° taper angle was chosen because it has higher heat flux coefficient than 3.4° tapered manifold as concluded from benchtop thermosiphon tests. Although, 6° taper had lesser CHF than 3.4° taper but heat flux dissipation requirement in CPU cooling was not more than 12 W/cm². It is observed in 6° tapered manifold that bubbles didn’t leave the region between microchannels and tapered manifold surface, instead fully grown bubbles stayed in the region. The bubbles stayed in the region as CPU reached steady state during stress test. Therefore, 6° taper was not tested further in thermosiphon loop for CPU cooling; instead 3.4° taper angle was used in the
manifold for CPU cooling application. The detailed explanation is provided later in the conclusion section.

The stress test was initiated after baseline completion and average CPU temperature as shown in Fig.35 was recorded. The power consumption by CPU under stress test condition was measured and is shown in Fig.36

![Figure 35: CPU temperature variation under stress test](image-url)
Figure 36: CPU power consumption under stress test

The predicted thermosiphon performance based on benchtop results and contact resistance offered in CPU cooling along with heat dissipation performance of three cooling units is shown in Fig.37. The predicted CPU temperature is calculated using Eqn. (6). The temperature difference between CPU core and ambient air is defined as DT in °C.
The maximum average CPU cores temperature, heat flux (power consumption/CPU area) and maximum temperature difference between CPU and ambient air (DT) for three different cooling units is shown in Table 2. For calculating DT, the ambient air temperature was considered to be 29°C. The value is estimated on the basis of average room temperature in actual data center considering hot spots.
Table 2: Heat transfer performance of different CPU cooling units

<table>
<thead>
<tr>
<th>Cooling Unit</th>
<th>CPU Core Temperature (°C)</th>
<th>Heat Flux (W/cm²)</th>
<th>DT (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air cooling</td>
<td>83</td>
<td>11.9</td>
<td>54</td>
</tr>
<tr>
<td>Water cooling</td>
<td>63</td>
<td>10.3</td>
<td>34</td>
</tr>
<tr>
<td>Thermosiphon cooling</td>
<td>84</td>
<td>11.9</td>
<td>55</td>
</tr>
</tbody>
</table>
6. DISCUSSION

6.1 Benchtop Thermosiphon

The benchtop thermosiphon loop testing was done in two phases. Initially, two taper angles 3.4° and 6° were tested for a 10 mm x 10 mm projected microchannel area. In the second phase, effect of liquid volume developing pressure head at the inlet of manifold was tested.

**Effect of taper angle in the manifold**

The 6° taper provides an exit cross section area of 7.94 mm² compared to the inlet area of 2 mm² whereas 3.4° taper angle provides exit cross section area of 12.5 mm² for the same inlet. The 6° taper supports more bubble growth in the manifold due to greater cross section space in the manifold. This greatly improves the efficiency of heat transfer performance of the tapered manifold as explained in the literature review[21,22]. Therefore, 6° taper performs superior to 3.4° taper.
**Effect of liquid head at the manifold inlet**

The experimental study performed using 6° taper manifold for three liquid heads at inlet showed shows continuous increment in critical heat flux with increase in liquid head. The higher head develops more liquid inertia in the heating section thus vapor bubbles are removed easily from the tapered region. But decrease in heat transfer coefficient is also seen with increase in liquid head. The increased liquid inertia along with removing bubbles from the tapered region also presses nucleating bubbles on the microchannels surface causing higher surface temperature. The same mechanism is also explained in literature where Kalani and Kandlikar[27] reported decrease in HTC at high flow rates.
6.2 CPU Cooling Using Different Cooling Techniques

Effect of taper angle in the manifold in thermosiphon loop

The 6° taper’s experimental performance is not presented because the 6° taper manifold didn’t work during the CPU cooling application. The vapor bubbles formed during boiling didn’t leave the manifold space. A stable vapor dominated region was formed near the exit in the manifold causing high surface temperatures.

The 6° taper angle provides 135.8 mm² cross section area at the exit where inlet area is 6.4 mm². The area increases about 21 times at the exit compared to inlet, whereas in benchtop configuration this area increment was only 6.25 times. This results in a significant drop in liquid inertia along flow length of 38.5 mm in CPU cooler and liquid lacks momentum to push the vapor out of the manifold. The taper angle of 3.4° is used in the manifold which increases area 12.4 times at exit. The taper angle 3.4° is chosen in order to compare the results with benchtop configuration.
**Power consumption by different coolers**

The power consumption comparison for water cooler, air cooler and thermosiphon cooler as shown in Fig.36 presents higher power consumption for air and thermosiphon cooler. The additional fans present in the CPU tower are activated if water cooler is not used for system cooling. Thus, adding more consumption by the system.

**Heat dissipation performance of thermosiphon loop**

The thermosiphon performance is predicted based on the experimental data from the benchtop configuration. A significant deviation is observed between predicted performance and actual performance. The thermosiphon doesn’t perform as predicted due to the geometry of evaporator in CPU cooling. The area in manifold in CPU cooling application increases 12.4 times whereas for benchtop configuration it was only 4. The less liquid inertia at near exit in manifold causes insufficient removal of bubbles affecting the heat transfer performance. Also the exit diameter in the manifold in order to connect manifold with tube fittings is 10mm, compared to the exit width of manifold 32 mm this causes sudden contraction at exit developing back flow of vapor.
The thermosiphon loop is not able to match the performance of water cooler also because of the high contact resistance between copper chip and CPU surface. Presently, the bulk resistance is 2.5 cm²K/W, this suggests that minimum CPU temperature for microchannel surface to reach saturation temperature of HFE 7000 must be 65.5°C. If this resistance is reduced to 1.5 cm²K/W then required CPU temperature to would be 53.7°C.
7. CONCLUSIONS

The two configurations of thermosiphon loop, the benchtop and a CPU cooler for server in data center are experimentally studied in this work. Thermosiphon as CPU cooler is compared with the presently used single phase (air and water) CPU coolers at RIT data center. The preliminary thermosiphon tests are conducted with benchtop configuration and later evaporator is designed for CPU cooling based on benchtop thermosiphon loop results.

- The benchtop thermosiphon loop study was done for two manifold taper angles, 3.4° and 6°. The critical heat flux of 34.4 W/cm² and 44 W/cm² was achieved at wall superheat of 10.3°C and 16.2°C for 3.4° and 6° taper angles. The maximum heat transfer coefficient achieved was 27.14 kW/m² °C and 33.4 kW/m² °C for 3.4° and 6° taper angles.

- The effect of liquid head at manifold inlet was studied for three different fill volumes in benchtop thermosiphon loop with 6° taper angle. The fill volumes used were 175 ml, 250 ml and 325 ml developing a total head of 0.227 m, 0.239 m and 0.251 m. The CHF of 34.4 W/cm², 36.9 W/cm², 40.5 W/cm² and maximum HTC of 33.4 kW/m² °C, 22.5 kW/m² °C, 22.7 kW/m² °C was achieved for 0.227 m, 0.239 m, 0.251 m liquid heads.
• The maximum average CPU cores temperature achieved under stress for air, water and thermosiphon cooler was 82.6 °C, 63.4 °C, 84.4 °C for heat flux dissipation of 11.9 W/cm², 10.28 W/cm², 11.94 W/cm² respectively.

• The benchtop thermosiphon loop results show the promising potential of thermosiphon system as a replacement of air or water cooling techniques for CPU cooling in data center. The bulk contact resistance and large cross section increase in thermosiphon setup for CPU cooling are limiting the effective heat dissipation capability of thermosiphon cooler.
8. FUTURE WORK

• The evaporator contains a 3.4° taper angle in the manifold in thermosiphon loop used for CPU cooling. This offers 12.4 times area increment at the exit of manifold compared to inlet. This decreases the liquid inertia significantly thus decreasing the heat transfer process. A new manifold testing is proposed with smaller taper angle to achieve higher liquid momentum force. Also, better evaporator mount can be developed to reduce the contact resistance.

• The design parameters in the evaporator like gasket thickness, inlet/outlet size and inclination need to be optimized. The gasket thickness defines the gap over the microchannels providing space for bubble growth. The outlet diameter is 10 mm compared to the exit of tapered region which is 32 mm, this causes vapor contraction and instabilities are developed in the system.

• The heater in the benchtop thermosiphon can be developed to match the geometry of actual CPU surface dimensions. The geometry of heating surface is a critical parameter affecting the performance of thermosiphon loop.
9. REFERENCES


Appendix

MATLAB code for analyzing data from CPU cooling tests

```matlab
% Thermosiphon Testing
% Data Analysis for cpu testing
close all
clear all
clc

% Read in raw data
message = sprintf('Load csv file for CPU1-4 temps and power.\nMust select files in that order.\nFile must be in current root directory.');
reply = questdlg(message, 'Load File', 'OK', 'Cancel', 'OK');
if strcmpi(reply, 'Cancel')
    return;
end
filename = uigetfile('*.csv','MultiSelect','on');
cpu1raw=csvread(string(filename(1,1)));
cpu2raw=csvread(string(filename(1,2)));
cpu3raw=csvread(string(filename(1,3)));
cpu4raw=csvread(string(filename(1,4)));
powerraw=csvread(string(filename(1,5)));

maxtime1=max(cpu1raw(:,1));
maxtime2=max(cpu2raw(:,1));
maxtime3=max(cpu3raw(:,1));
maxtime4=max(cpu4raw(:,1));
maxtime5=max(powerraw(:,1));
finaltime=min([max(cpu1raw(:,1)),max(cpu2raw(:,1)),max(cpu3raw(:,1)),max(cpu4raw(:,1)),max(powerraw(:,1))]);

l=[length(cpu1raw(:,1)),length(cpu2raw(:,1)),length(cpu3raw(:,1)),length(cpu4raw(:,1)),length(powerraw(:,1))];
s1 = size(cpu1raw);
s2 = size(cpu2raw);
s3 = size(cpu3raw);
s4 = size(cpu4raw);
s5 = size(powerraw);
a = max([s2(1),s2(1),s3(1),s4(1),s5(1)]);
catraw = [cpu1raw;zeros(abs([a 0]-s1))],[cpu2raw;zeros(abs([a 0]-s2))],[cpu3raw;zeros(abs([a 0]-s3))],[cpu4raw;zeros(abs([a 0]-s4))],[powerraw;zeros(abs([a 0]-s5))]);
```

---

89
%cpuraw=cat(3,z(:,1:2),z(:,3:4),z(:,5:6),z(:,7:8));

[Y,B]=max(I);
raw=zeros(Y,6);
count=1;
starter=1;
tic
for i=1:Y
    for j=starter:Y
        if catraw(i,2*B-1)==catraw(j,9)&&catraw(i,2*B-1)<=finaltime
            raw(count,1)=catraw(i,2*B-1);
            raw(count,2)=catraw(i,2);
            raw(count,3)=catraw(i,4);
            raw(count,4)=catraw(i,6);
            raw(count,5)=catraw(i,8);
            raw(count,6)=6*catraw(j,10); % converted to watts here
            count=count+1;
            starter=j+1;
            break;
        end
    end
end
toc
raw( ~any(raw,2), : ) = [];

% Filter
% All frequency values are in Hz.
Fs = 1; % Sampling Frequency
N  = 1; % Order
Fc = 0.005; % Cutoff Frequency
% Construct an FDESIGN object and call its BUTTER method.
h  = fdesign.lowpass('N,F3dB', N, Fc, Fs);
Hd = design(h, 'butter');
% Filter all data
%raw(:,2:6)=filter(Hd,raw(:,2:6));
% Initialize matix for final use
final=zeros(length(raw(:,1)),13);
% Change epoch time to standard time
for i=1:length(raw(:,1))
    % Get unix time for current point
    unix_time=raw(i,1);
    % Convert epoch time to standard
    [y,m,d,h,mi,s] = datevec(datenum([1970 1 1 -5 0 unix_time]));
    final(i,1)=unix_time;
final(i,2:7)=[y,m,d,h,mi,s];
final(i,8)=(s+60*mi+3600*h)/3600; % total hours on day
final(i,9)=raw(i,2); % CPU1 Temp
final(i,10)=raw(i,3); % CPU2 Temp
final(i,11)=raw(i,4); % CPU3 Temp
final(i,12)=raw(i,5); % CPU4 Temp
final(i,13)=raw(i,6); % Power
end

% Time to start graph and end
StartTimePrompt = {'Enter start hour (0-24):','Enter start minute (0-59):','Enter end hour (0-24):','Enter end minute (0-59):'};
StartTimeTitle = 'Test Time';
StartTimeLines = 1;
StartTimeDef = {'0','0','23','59'};
answer = inputdlg(StartTimePrompt, StartTimeTitle, StartTimeLines, StartTimeDef);
hstart=str2double(answer{1});
mistart=str2double(answer{2});
hend=str2double(answer{3});
miend=str2double(answer{4});

% Plot Data
hFig = figure(1);
set(hFig, 'Position', [250 250 1250 500])
subplot(1,2,1)
plot(final(:,8),final(:,9))
xlim([hstart+mistart/60 hend+miend/60])
hold on
plot(final(:,8),final(:,10))
plot(final(:,8),final(:,11))
plot(final(:,8),final(:,12))
hold off
title(sprintf('CPU Temperature over time'))
xlabel('Time (s)')
ylabel(sprintf('Temperature (%cC)',char(176)))
legend('CPU1','CPU2','CPU3','CPU4')
subplot(1,2,2)
plot(final(:,8),final(:,13))
xlim([hstart+mistart/60 hend+miend/60])
title(sprintf('Power Consumption over time'))
xlabel('Time (s)')
ylabel('Power Consumption (W)')

% Determine range over which to compute average to normalize power consumption
StartTimePrompt = {'Enter start hour to calculate average:', 'Enter end hour to calculate average'};
StartTimeTitle = 'Average Time';
StartTimeLines = 1;
StartTimedef = ['0', '0'];
answer = inputdlg(StartTimePrompt, StartTimeTitle, StartTimeLines, StartTimedef);
avestart = str2double(answer{1});
aveend = str2double(answer{2});
sum = zeros(5, 1);
count = zeros(5, 1);
for i = 1:length(final(:, 1))
    if final(i, 8) >= avestart && final(i, 8) <= aveend
        sum(1, 1) = sum(1, 1) + final(i, 9);
        count(1, 1) = count(1, 1) + 1;
        sum(2, 1) = sum(2, 1) + final(i, 10);
        count(2, 1) = count(2, 1) + 1;
        sum(3, 1) = sum(3, 1) + final(i, 11);
        count(3, 1) = count(3, 1) + 1;
        sum(4, 1) = sum(4, 1) + final(i, 12);
        count(4, 1) = count(4, 1) + 1;
        sum(5, 1) = sum(5, 1) + final(i, 13);
        count(5, 1) = count(5, 1) + 1;
    end
end
ave = sum ./ count;

% Reduce temperature by average
avefinal = final;
avefinal(:, 9) = avefinal(:, 9) - ave(1, 1);
avefinal(:, 10) = avefinal(:, 10) - ave(2, 1);
avefinal(:, 11) = avefinal(:, 11) - ave(3, 1);
avefinal(:, 12) = avefinal(:, 12) - ave(4, 1);
avefinal(:, 13) = avefinal(:, 13) - ave(5, 1);

% Plot normalized data
hFig = figure(2);
set(hFig, 'Position', [250 250 1250 500])
subplot(1, 2, 1)
plot(avefinal(:, 8), avefinal(:, 9))
xlim([hstart + mistart / 60 hend + miend / 60])
hold on
plot(avefinal(:, 8), avefinal(:, 10))
plot(avefinal(:, 8), avefinal(:, 11))
plot(avefinal(:, 8), avefinal(:, 12))
title(sprintf('Normalized CPU Temperature over time'))
xlabel('Time (s)')
ylabel(sprintf('Temperature (%cC)',char(176)))
legend('CPU1','CPU2','CPU3','CPU4')
subplot(1,2,2)
plot(avefinal(:,8),avefinal(:,13))
xlim([hstart+mistart/60 hend+miend/60])
title(sprintf('Normalized Power Consumption over time'))
xlabel('Time (s)')
ylabel('Power Consumption (W)')

% Determine range over which to plot boiling curve
StartTimePrompt = {'Enter start hour to calculate boiling curve:', 'Enter end hour to calculate boiling curve:'};
StartTimeTitle = 'Boiling Time';
StartTimeLines = 1;
StartTimedef = {'0','0'};
answer = inputdlg(StartTimePrompt, StartTimeTitle, StartTimeLines, StartTimedef);
boilstart = str2double(answer{1});
boilend = str2double(answer{2});

% create boiling graph
boiling=zeros(length(final(:,1)),6);
count=1;
for i=1:length(final(:,1))
    if final(i,8)>=boilstart&&final(i,8)<=boilend
        boiling(count,1)=final(i,9);
        boiling(count,2)=final(i,10);
        boiling(count,3)=final(i,11);
        boiling(count,4)=final(i,12);
        % boiling(count,5)=(normfinal(i,8,1)+normfinal(i,8,2)+normfinal(i,8,3)+normfinal(i,8,4))/4;
        boiling(count,6)=avefinal(i,13);
        count=count+1;
    end
end
% remove zero rows
boiling(~any(boiling,2), : ) = [];

% plot boiling curve with average CPU temp
figure(3)
c = linspace(1,10,length(boiling(:,1))); scatter(boiling(:,5),boiling(:,6),25,c,'filled')
title(sprintf('Normalized Power Consumption vs Average Temperature'))
xlabel(sprintf('CPU Temperature (%cC)',char(176)))
ylabel('Normalized Power Consumption (W)')
% Normalized temps and power by maximum
% figure(4)
% plot(normfinal(:,1),normfinal(:,9)/max(normfinal(:,9)))
% xlim([1479070800 1479074400])
% hold on
% plot(normfinal(:,1),normfinal(:,10)/max(normfinal(:,10)))
% plot(normfinal(:,1),normfinal(:,11)/max(normfinal(:,11)))
% plot(normfinal(:,1),normfinal(:,12)/max(normfinal(:,12)))
% plot(normfinal(:,1),normfinal(:,13)/max(normfinal(:,13)))
% title(sprintf('Normalized CPU Temperature over time'))
% xlabel('Time (s)')
% ylabel(sprintf('Temperature (%cC)',char(176)))
% legend('CPU1','CPU2','CPU3','CPU4','Power')