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**A Micro Loop Heat Pipe Thermal Management System with an Interline
Optimized Evaporator and a Coherent Porous Silicon Wick**

by

Lilla Marion Safford Smith

A dissertation submitted in partial satisfaction of the
requirements for the degree of
Doctor of Philosophy

in

Engineering - Mechanical Engineering

in the

Graduate Division

of the

University of California, Berkeley

Committee in charge:

Professor Albert P. Pisano, Chair
Professor Liwei Lin
Professor Amy E. Herr

Summer 2015

**A Micro Loop Heat Pipe Thermal Management System with an Interline
Optimized Evaporator and a Coherent Porous Silicon Wick**

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Lilla Marion Safford Smith

Abstract

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Doctor of Philosophy in Engineering - Mechanical Engineering

University of California, Berkeley

Professor Albert P. Pisano, Chair

Throughout the history of electronics, technology has been driven by shrinking devices and increasing performance. This progress has hit a wall that is limiting everyone, and this surprising limitation is thermal management and overheating. Cooling systems for electronics used to be one of the last design elements considered, fins, fans and graphite sheets were sufficient solutions, but that is no longer the case. The technology landscape has changed, and a new wave of solutions must be created and implemented quickly, or innovation will come to a standstill. The solution proposed in this research is the micro hybrid loop heat pipe (HLHP). This device fits in with the recent innovations in using phase change devices, such as micro heat pipes and micro vapor chambers, but uses cutting edge silicon heat transfer technology to improve upon these other approaches.

The system is a three layer stack with an interline, optimized evaporator surface and a coherent porous silicon wick, sometimes referred to as a multi-scale pore membrane (MPM) [1]. Connecting the enhanced silicon components is an aluminum housing to act as the piping. The total stack thickness is less than 3mm, and has the potential to be slimmed to 1-2 mm. The hybrid system is tested at fluxes of 25 W/cm² and higher. The evaporator structure has heat transfer coefficients as high as 50,000 W/m² K. The work in this dissertation is, to the authors knowledge, the first loop heat pipe that is able to dissipate this amount of heat with such a thin form factor and utilizes a never before tested evaporator arrangement. Data showing a functioning system, the heat transfer rates of that system, the thermal resistance, and other metrics of this cutting edge device are presented in this work.

The path of the research from an initial design to a final system, gives rise to an important discussion of how this device was created. On this path, background research was done that lead to increased focus on each component of the final system. The wick, the evaporator, new possibilities in laser fabrication techniques, and the sealing of the device were all thoroughly explored before the system was built. Once the groundwork of the component research was laid, modeling of the system was conducted to try and determine the limitations and room for improvement in future iterations of this design. Finally, the system was built, assembled

and tested, creating the first micro loop heat pipe (μ LHP) of this size scale. In addition, this device, shown here in its 3mm version, could be built out of silicon (or other materials) and shrunk to under 1mm without changing any of the channel depths or geometries. The structure was designed to be less than 1mm thick and was made thicker for ease of fabrication. Information on all of the component advancements and the process for creating a system that can handle up to 12W with 50W/cm² heat flux is presented here.

To My Family and Friends

No one can do this alone and I have had a wonderful group of people supporting me and keeping me going through it all! The universe has blessed me, and in return I wrote this really long PhD for you.

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Chapter 1

Introduction

Since the beginning of integrated circuits technology, transistor dimensions have steadily decreased. As Moore's law predicted, large amounts of power are being applied to smaller and smaller areas. This results in an increased quantity of waste heat in confined spaces that initially wasn't a problem. Today, management of heat is growing to be one of the largest problems in electronics. Thermal management, which used to be an afterthought in electronics design, has moved to the forefront. Current thermal solutions are almost exclusively materials focused. Better thermal contact materials and improvements to graphite sheets have increased thermal conductivity to the levels of diamond [2], but are much cheaper and can even be flexible. These solutions allow electronics to dissipate heat, but only for short periods of time, leaving power cycling and performance throttling as the main mechanisms to keep the overall temperature down. This is a passable solution for now, but simple thermal conduction can only do so much, and material development can only go so far. Consequently, there is great need for solutions that can dissipate much higher heat at high flux.

One future solution is the introduction of phase change devices to increase heat dissipation and heat absorption. In most electronics systems there are two key factors that impact a device's thermal profile; one is the small area in which the heat is produced, the other is that the dissipation of that heat is often limited to what is provided by the ambient environment. This can be a huge problem because the heat removal method must be able to handle a large amount of heat flux and many devices cannot be coupled to large scale heat sinks. Heat flux is the amount of heat energy transferred through a surface per unit area (W m^{-2}), so even if the heat output is limited, if the area is small then the heat flux can be large. To handle this heat flux, a device that can absorb a lot of heat is needed. The second problem is that even if the heat can be absorbed, it needs to be moved to a large area where it can be removed by ambient conditions, this is why good heat dissipation is important.

Phase change devices are systems that use the change in state of a fluid to absorb and dissipate heat quickly and efficiently. The fluid goes from a liquid state to a vapor state and in that process absorbs a lot of energy or heat. The vapor travels rapidly to another location in the system due to volume expansion during boiling, where it converts back to a liquid and dumps that heat. These systems can come in many shapes and sizes, but they

have the unique ability to absorb heat in a small area and then move that heat and dissipate it, usually over a large area. Phase change devices are not new, but improvements to these systems will make them the preeminent thermal solutions as material improvements fail to keep up with the thermal demands of modern electronics.

1.1 Motivation and Market Trends

The need for efficient high flux absorbing type devices is mostly driven by recent market changes. According to the BCC, in January 2013 the global thermal management market was 7 billion in 2013 and is expected to grow to 10 billion by 2017. Of that market, hardware solutions like the one shown here make up 80% of the market. That market does not include thermal management substrates or interfaces, like graphite. A key thing to note is that this hardware market includes both large scale and small scale heating solutions and is mostly made up of heat sinks and enclosure fans, while phase change devices are just a small subset of the larger industry. Heat pipes and other cooling devices make up only 1% of the hardware market and this segment is expected to grow to 3% by 2017 [3]. However, this market study did not take into account any of the cooling solutions used in smart phones, so the true numbers may be higher than this data implies [3, 4].

Outside of the market data, it is also important to view where the market is going and what the trends in technology will be, so that new solutions can be catered to the needs of industry. One of the next big market trends according to Frost and Sullivan is the move to 3D-ICs. This is expected to be important in the consumer electronics industry and it also has potential in computers and network systems [5].

3D-IC technology is often confused with 3D packaging and 2.5D interposers. This stems from the fact that all electronics chips must communicate with off-chip signaling to work [5]. There are many combinations of packaging that stack or attach chips to substrates to achieve this signal generation. The difference is that 3D packaging has low levels of actual chip integration, in this case individual dies are made, packaged and then the packages are connected with wire or flip chip bonding. 2.5 D interposers are dies set side by side on an interposer and they use silicon vias through the interposer to connect the adjacent chips. These vias are usually called TSVs or through substrate vias. Actual 3D ICs on the other hand, stack multiple device layers that can use TSVs to form interconnects between the layers[5]. To make stacked chips, processes can be done in parallel or sequentially. Two chips can be made and etched with TSV's and then stacked and finally packaged, or one chip can be made and the additional layers can be deposited on top and then vias can be made. Both of these methods would count as 3D-IC technology as long as the chips are created with interconnected vias.

The technology shift towards 3D ICs is driven by the possibilities of increased performance, functionality, power savings and decreased footprint[5]. Instead of making each chip faster the number of chips can be increased to improve speed and by stacking the new chips electronic devices as a whole can still stay small. From a thermal standpoint, the heat gen-

eration with multiple high power chips on top of each other is enormous. The upside is that this push towards highly integrated chips makes an all silicon directly integrated cooling solution ideal. If the next generation micro loop heat pipe (μ LHP) devices can be made to act as interposers then the potential compatibility with 3D ICs is excellent.

1.2 μ LHP Operation and Components

As previously mentioned, phase change devices are not new, but they have the potential to be the next meaningful solution for thermal management issues. One of the historical limitations of these devices has been the difficulty of applying the classic miniaturization model while maintaining high performance. In this model, a previously working macro scale design is shrunk in scale, and then all the problems that arise from the shrinking process are managed independently. This model can work, but it limits form factors and starts the design process in a small box. The approach shown here, rejects this model from the outset. The micro loop heat pipe (μ LHP) presented in this work was designed with the micro scale in mind, so that physical effects that occur only at that scale, can be used to their full advantage. Each component of the device was designed to work at the micro scale so that the final system can be built thinner without losing its great performance.

The μ LHP device was designed to be a three layer device with two focus areas: the evaporator, and the coherent porous silicon (CPS) capillary pump, called the wick. The system functions as a loop heat pipe, see figure 1.1. First, there is a wick that provides capillary pressure to circulate the fluid which then goes through a phase change from liquid to vapor in the evaporator. It then travels down vapor channels to a condenser where it condenses back into liquid and travels down liquid lines back into the compensation chamber where the liquid is then pumped back into the wick. This happens continuously throughout the functioning of the device. Overall the device is meant to be 1 mm or less in thickness and dissipate 100+W. To enable this, the evaporator and wick were designed to function as efficiently as possible.

The initial plan for the device was to make all three layers out of silicon, so that it could be the substrate for electronics to be built upon, but as research progressed, this plan was altered to utilize specifically engineered silicon components in conjunction with a metal frame. This hybrid loop heat pipe design (HLHP) allowed for critical micro components that required tight tolerances and were individually designed to be tested to be integrated into a larger device. The aluminum frame allowed for the larger piping to be made in an easy to machine material saving on fabrication time. The μ LHP design shown in figure 1.1, has a CPS wick (discussed in section 3.2) and an interline optimized evaporator (discussed in section 3.1), with all of the components including the fluid channels and the vapor channels made of silicon. This device could be considered the ideal or final goal device, but it would be difficult to fabricate and bond. Moreover, each design iteration would be highly time consuming. This design also has issues with sealing the test devices that can produce reliability issues during testing and evaluation, see section 3.4. To speed this process up, each component

was designed and optimized individually. Parts of the system were then tested together to verify they worked in conjunction before the full system was assembled in aluminum. The aluminum allowed for changes to the system with the added advantage of rapid turn around and preservation of the precise silicon components. In the work presented here, the components have been tested individually and the system has been built and tested as a whole demonstrating proof of principle so that future work can be undertaken to make an all silicon device.

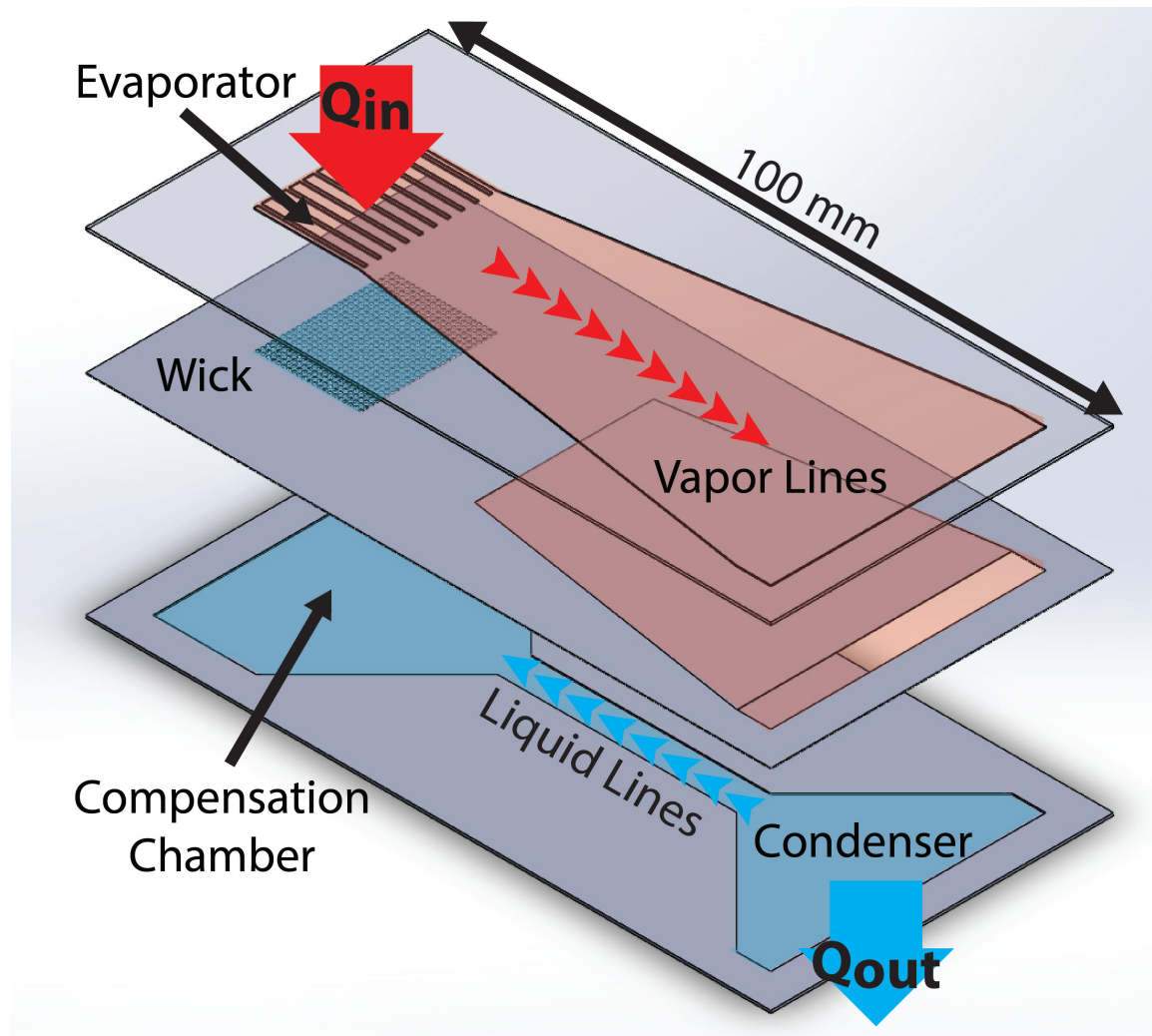


Figure 1.1: Three layer stack of final proposed silicon μ LHP.

1.2.1 CPS

The coherent porous silicon (CPS) wick with zinc oxide nanowires is the pump of the system. Surface tension forces in the small capillaries create pressure that pushes fluid through the

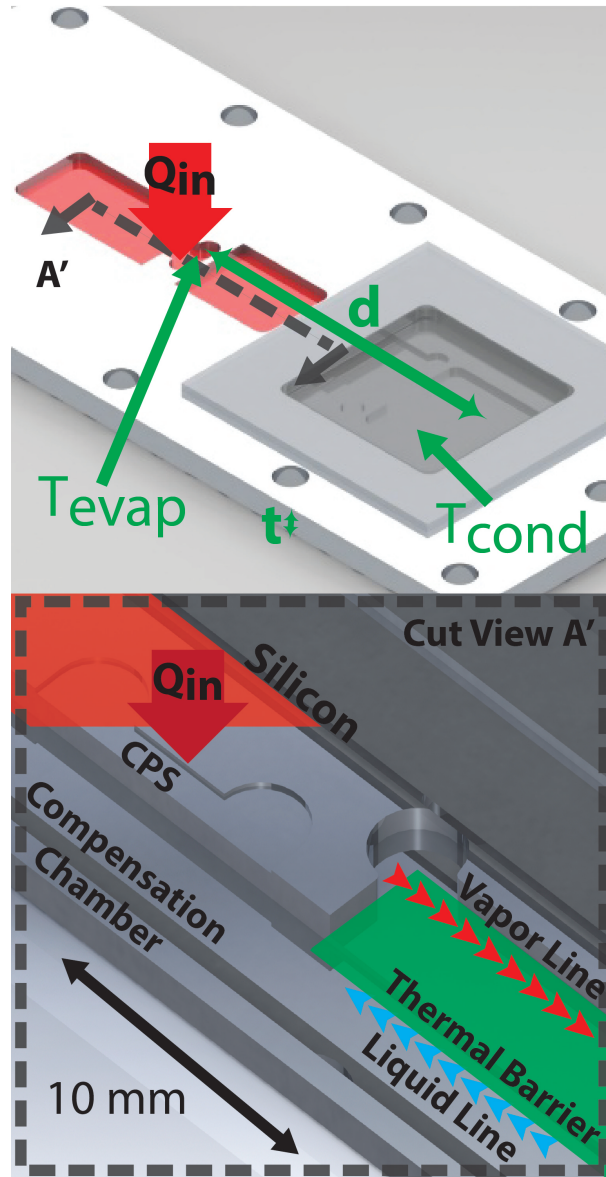


Figure 1.2: Hybrid loop heat pipe design including internal view with area for the CPS wick and a thermal barrier to separate the vapor and liquid flow.

wick. Once the channels are full, those same forces make it hard for the fluid to be pushed back through the wick. The zinc oxide nanowires then help the liquid flow out of the wick onto the evaporator surface where the liquid can undergo phase transition. The important design factor in this component is that it creates more pressure across the wick than is lost from frictional forces during the rest of the fluid circulation.

1.2.2 Interline Evaporator Region

The interline optimized evaporator region is where the heat is absorbed, while the fluid goes through the phase change process. The concept behind this design is that, in order to have boiling on a surface, the liquid needs to be above its boiling point. If the liquid is above its boiling point the surface must be at a higher temperature. This is not ideal; and is inefficient, as most of the energy goes into heating large amounts of liquid. This causes the liquid to superheat above the boiling point and increases the temperature of the surface. The interline region is an area where the fluid is spread thin by capillary forces so there is less liquid to heat, so boiling is suppressed in the thin film and instead evaporation occurs efficiently. Details of this process are discussed in section 3.1, but in short, an evaporator with no bubbles doesn't need as much energy to cause phase change and can create vapor at a lower temperature. This makes for more efficient heat absorption.

1.2.3 System Sealing

The system cannot work if the fluid is leaking, but there is no simple microfluidic sealing technique. The risk of device failure due to leaking required the creation of a unique bonding solution for silicon to microvalves for this specific system. This aspect of the research is not part of the HLHP function, but it needed to be understood in order for future devices to function and for test devices to produce reliable data. In addition, fluid loss prohibits circulation, yet a permanent sealing solution would limit the options for determining filling ratios and the ideal internal pressure. The solution to this problem was the development of a new sealing method, which is explained in section 3.4.

1.2.4 Design Goals

The design goals for the μ LHP system.

Key Performance Metrics			
	μ CLHP	Heat Pipe	Graphite Sheet
Thickness:	1-3 mm	0.7-3 mm	0.25 mm
Power Dissipation:	10-50 W	1-50 W	1-3 W

Table 1.1: Table showing performance goals of the final design.

- The μ LHP presented in this work is 3 mm thick and can handle 12 W with heat dissipation to ambient conditions (no fans or heat sinks). There is the potential to continue the development by shrinking the device to 1 mm or less and increase the power dissipation to 50W with the help of fans or heat sinks. This is discussed in Chapter 6 and 7.
- Heat Pipes come in a range of sizes from 0.7 mm to 3+ mm. The 0.7 mm devices can handle 1-3 W, but the thicker 3 mm+ devices can handle upwards of 50W with the addition of fans or heat sinks.
- The graphite sheets can be anywhere from 0.1mm to 0.25mm, but can only dissipate 3W at maximum depending on the ambient conditions.

1.3 Outline

The path of this research, from an initial design to a final system, gives rise to an important discussion on how the device was created and the principles that enable its functionality. On this path, background research was done that lead to increased focus on the performance of each component in the final system. The wick, the evaporator, new possibilities in laser fabrication techniques, and the sealing of the device were all thoroughly explored before the system was built. Once the groundwork of the component research was laid, modeling of the system was conducted to try and determine the limitations and room for improvement in future iterations of this design. Finally, the system was built, assembled and tested, creating the first micro loop heat pipe (μ LHP) of this size scale. In addition the advancement shown here, in the 3mm version, could be built out of silicon or other materials and shrunk to under 1mm without changing any of the channel depths or geometries. The structure was designed to be less than 1mm thick and was made thicker for ease of fabrication. Information on all of the component advancements and the fabrication of a system that could handle up to 12W with 50W/cm² heat flux is presented here.

Chapter 2

Device Physics

There are several key parts to understanding the specific phase change device presented in this work, these parts include: the existing types of devices in the field, the key physics in that device that define how it functions and the specific evaporation technology used called interline evaporation. Passive phase change thermal management devices come in many different configurations and can be created using a huge variety of configurations and liquids. Five common types of existing devices are: heat pipes, loop heat pipes, capillary pumped loops, oscillating heat pipes and vapor chambers. Within any of these categories there can exist many devices. The key is that all of these devices function using phase change as a driving force to dissipate heat. Phase change is a type of heat transfer that can involve large heat transfer rates with small temperature differences, as long as the temperatures are sufficient to cause a phase transformation, usually between liquid and vapor. Large heat transfer rates are driven by latent heat effects that only occur when phase change is present, specifically when liquids boil or condense. These effects are associated with the convection mode of heat transfer, because they involve fluid motion. The important variables in phase change are latent heat (h_{fg} or h_{lv}), the surface tension at the liquid-vapor interface and the density difference between the two phases. Large density differences between liquid and vapor phases are especially important because they create an additional buoyancy force. During phase change the combination of the latent heat and buoyancy effects creates heat transfer coefficients and rates that are usually much larger than regular convection without phase change. This is because the phase change process absorbs large quantities of heat and then the buoyancy forces move that heat rapidly creating stable isothermal systems with constant applied heat flux.

Understanding the functionality of phase change device requires knowledge of capillary forces, which act as a pumping mechanism to move the fluid and boiling phenomena, they are key to the transition between liquid and vapor. Capillary forces are the cause by intermolecular forces between the liquid and a solid surface. These forces are usually in opposition to gravitational forces and can cause liquids to spread over surfaces creating beneficial phase change conditions. This is a consistent and stable phenomena that can be easily calculated. Boiling on the other hand is usually a transient process that can be equated to correlations

to give estimates of expected behavior. This can be limiting if trying to use a surface that is not commonly characterized or well document in literature. Boiling can occur in several modes, the most common of which, is convective pool boiling (CPB); the most relevant to this research, is forced convection boiling or convective flow boiling (CFB). In this research convective flow boiling and interline evaporation, discussed in detail below, were used to maximize efficiency of the device.

2.1 Existing Technology

2.1.1 Heat Pipe

The TGP device is modeled after a heat pipe and uses some of the same concepts, so a basic understanding of heat pipes will aid in the overall understanding of the device. A heat pipe is a phase change thermal management device that uses latent heat to cool a hot spot. It is usually a circular pipe with a wicking material covering the inside surface and a hollow center, see figure 2.1. During operation heat is added to one side of the heat pipe, the evaporator side and removed via condensation on the other side. Once the vapor condenses it is drawn back to the hot side by the wicking structure creating a loop. This is the general operation of a heat pipe [6]. There has been a lot of research dedicated to heat pipes, specifically in improving wicking materials. All phase change devices have limits explained in (section 2.1.4.1), heat pipes have the added limitation of increasing surface resistance as applied heat increases. Surface resistance is created by having liquid move slowly in one direction and vapor move quickly in the opposite direction. These two opposing motions and interactions can create condensation before the end of the pipe, which causes evaporation to occur outside the evaporator, which chokes off flow. This is one of many issues associated with traditional heat pipes. More information on modern heat pipe research and technology can be found here.

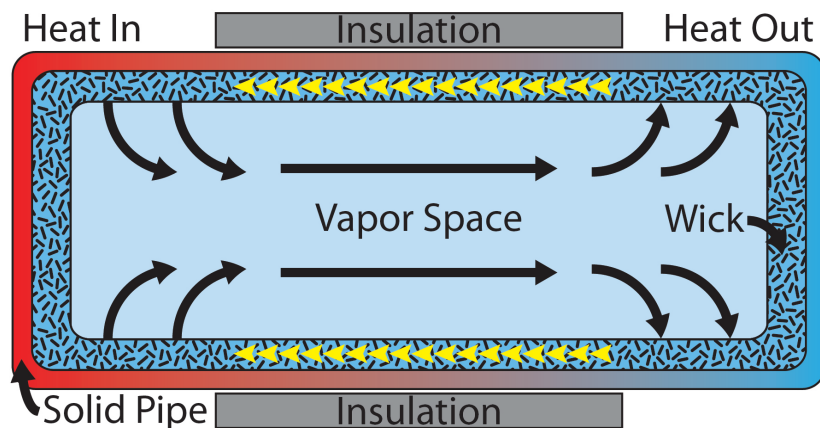


Figure 2.1: Schematic of a simple heat pipe.

2.1.2 Loop Heat Pipe

Loop heat pipes (LHPs) are passive phase change devices like heat pipes, but they contain a circulating loop that separates the liquid flow from the vapor flow. In general loop heat pipes are more difficult to make work, but they have much higher heat transport capability. The TGP device is generally based on the principles of a loop heat pipe with some adjustments. Understanding how these work and all of the different configurations of LHPs is key to understanding the design choices that dominate this research.

In operation a LHP uses the same physical processes as conventional heat pipes. The components of a working LHP are a wick sometimes called the capillary pump, an evaporator, compensation chamber, condenser, and liquid and vapor transport lines. The wick creates the capillary pumping forces that circulate the fluid around the loop, it is usually made of small channels, a porous material, some sintered metal particles or a metal mesh. The compensation chamber is usually the largest region in a LHP, by volume, and it is critical to the functioning of the device. It allows excess liquid to build up in the device after it condenses, but before it reaches the evaporator and it keeps the wick from drying out. This unique feature makes a loop heat pipe stable, because sudden increases in applied flux will not cause the wick to dry out, instead it will begin to drain the compensation chamber. This will work until pressure losses in the fluid are greater than the capillary pumping power in the wick. The condenser is where the heat is dumped, it allows the vapor to condense back into liquid. The liquid lines, which are essentially pipes or channels move the liquid from the condenser back to the compensation chamber. The vapor transport lines on the other hand allow the vapor to move and expand or decrease in pressure before reaching the condenser. More on the defining equations and the specific operation of the loop heat pipe that influenced the final design of the TGP are in Chapter 4.

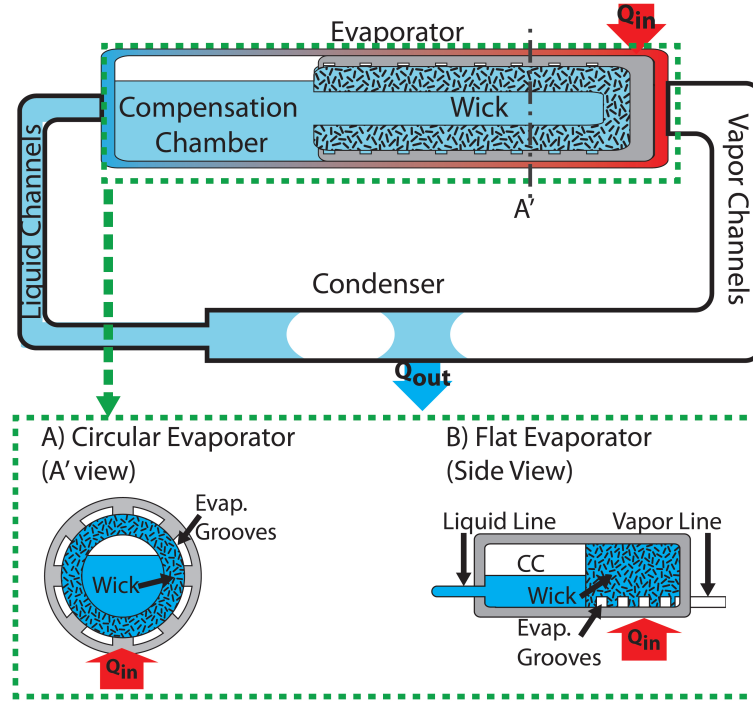


Figure 2.2: Schematic of a loop heat pipe

2.1.3 Oscillating Heat Pipe

An oscillating heat pipe (OHP) is a two-phase flow device it consists of a loop made of tubes/channels arranged in a serpentine manner see figure 2.3. The general function of an oscillating heat pipe is that the surface tension causes slugs of liquid to form in the serpentine separated by vapor bubbles [7][8]. When heat is applied to a specific region of the serpentine the fluid evaporates and the vapor bubbles grow and push the liquid column, then in the cooler condenser area the bubbles condense, pushing the fluid around the device. When the device is running there is an oscillation due to the pressure difference and as the device oscillates it also has an overall fluid circulation [9]. The fluid circulation is sometimes improved by adding a flow check valve to limit the circulation to one direction [10]. The advantages of an oscillating heat pipe are that it does not need a wick because the pressure change in the fluid causes the circulation. The device also has liquid and vapor moving in the same direction, which reduces the vapor frictional forces common in heat pipes due caused by the vapor and liquid moving in opposite directions. It is though that, because the heat pipe use capillary pumping they are limited in their possible heat flux absorption and they also have boiling and sonic limits that limit their working range [11]. OHPs on the other hand require specific minimum power level associated with a suitable temperature difference between the evaporator and condenser section to achieve startup. This means that OHP do not function well at low power inputs. Some hybrid OHPs have been created using wicks

and a serpentine flow, but these still only function well at high flux input around 100+W [12].

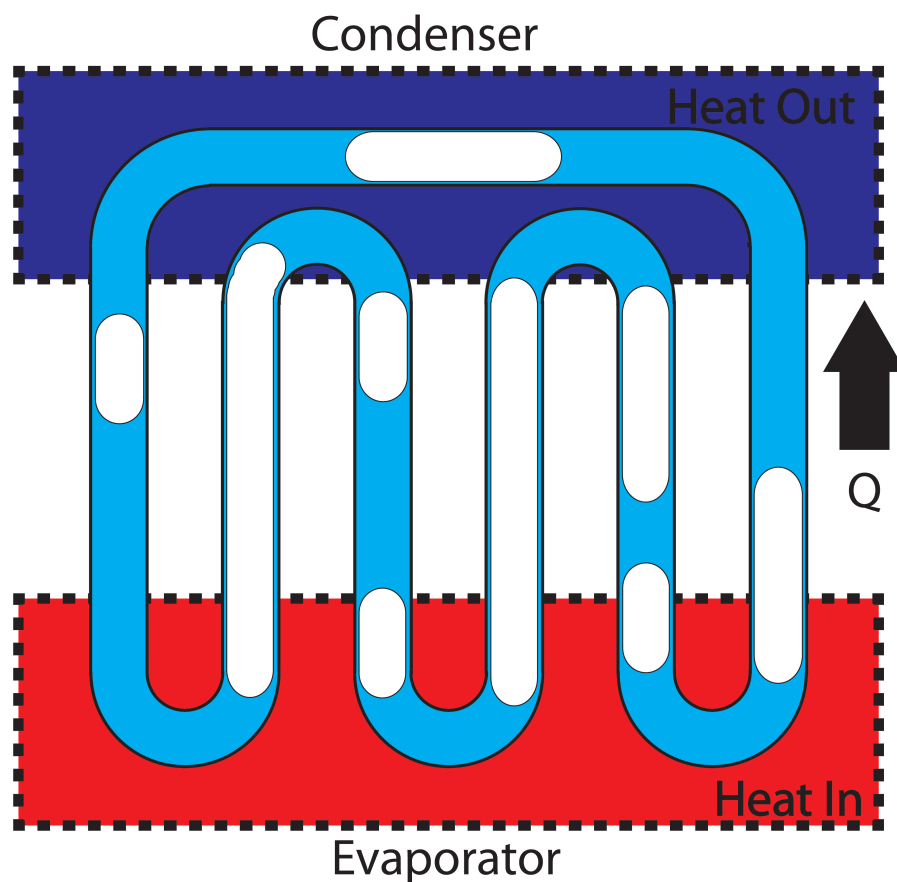


Figure 2.3: Schematic of an oscillating heat pipe.

2.1.4 Vapor Chamber

Vapor chambers are essentially flat heat pipes, instead of there being a hot side and a cold side the bottom is hot and the top is cold, see figure 2.4. The operation is the same as a heat pipe where the evaporation on the bottom causes a high pressure region that causes the vapor to move away from the evaporator and towards the lower pressure condenser at the top of the vapor chamber. With most vapor chambers the top is then attached to a metal heat sink, which removes the heat and causes the liquid to condense. The liquid is then taken back to the evaporator by the wick. The reason that a vapor chamber in conjunction with a heat sink is so useful is that it creates uniform cooling. Using only a heat sink would create hot and cold regions with a lot of thermal mismatch that is limited by the

thermal conductivity of the heat sink. A vapor chamber on the other hand, uses phase change to move heat quickly and uniformly, which allows heat to reach the heat sink with large temperature fluctuations. For more information Advanced Thermal Solutions has an excellent paper showing the advantages of vapor chambers. Figure 2.4, shows a schematic of the heat spreading difference between conventional solid base heat fins and heat fins with vapor chamber as the base.

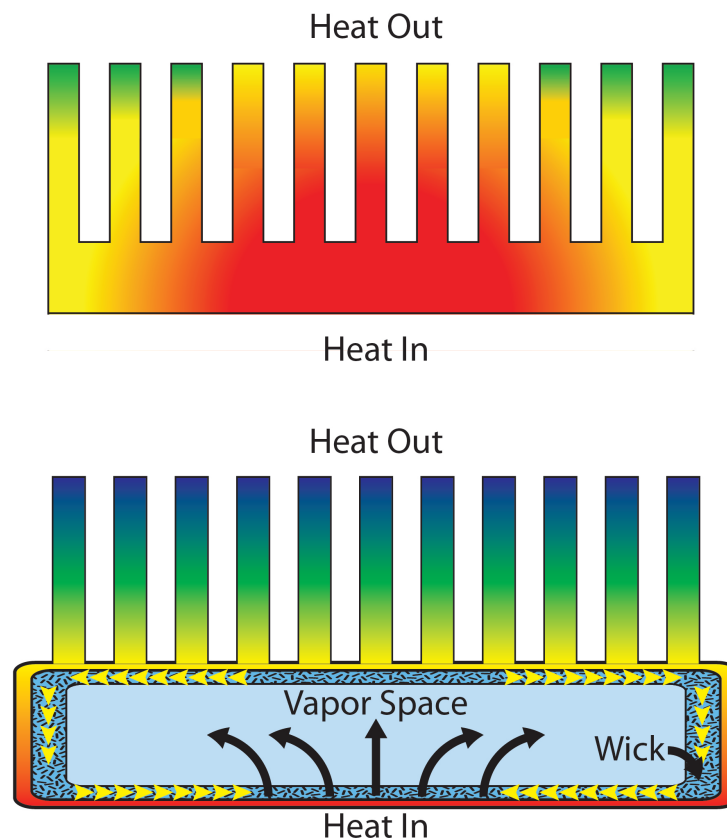


Figure 2.4: A schematic of a vapor chamber.

2.1.4.1 Limitations of Phase Change Devices

A literature review paper by Launay gives an excellent overview of the limitations of loop heat pipes that could be applied to all phase change devices. Reviewing these important limitations helps to give perspective on what is possible and not possible in a phase change device. The limits of a phase change device are: the viscous limit, the sonic limit, the entrainment limit, the capillary limit and the boiling limit [13]. The viscous limit is the point at which the viscous forces become larger than the pressure gradients that drive the

flow. This could occur in all phase change devices and is usually caused by low temperature conditions in the device, such as space or cryogenic applications.

The sonic limit is the point where the fluid, usually the vapor, reaches sonic speeds and the flow rate becomes choked. This is a real concern when phase change devices become smaller. Physically this can occur when the cross-sectional area decreases, if the mass flow rate increases as fluid travels down the length of a pipe or when fluid accelerates due to phase change when it exits the pores.

The entrainment limit is the point at which the interaction forces between the liquid phase and the vapor phase increase to the point where one fluid is changing the expected movement of the other fluid. This limit is less likely to occur in loop heat pipes, but can be common in other devices. In heat pipes this can occur when the fluid in the wick reverses directions away from the evaporator because the speed of the vapor causes shear forces larger than the capillary forces in the wick. In a loop heat pipe or vapor chamber this will most likely occur in the evaporator or primary wick. As the liquid evaporates it moves with such force it pulls droplets of liquid with it, if too much liquid is removed in the process the wick will dry out and the entrainment limit will be reached.

The capillary limit is one of the most important and must be calculated carefully in the initial design stages of a phase change devices. There is a pressure balance in the system where a positive pressure gradient is created in the wick and evaporator areas and then pressure decreases due to frictional forces in other areas of the system. This can be from frictional losses in the wick itself if the wick is long like in a heat pipe or in vapor or fluid channels like in a loop heat pipe.

The boiling limit is a limit that becomes proportionately less important as the devices get thinner. Boiling requires the surface to be at a certain temperature and a region for the vapor bubbles to escape. In a heat pipe this can be limiting, because the hottest region is at the wall creating bubbles at the bottom of the wick where it is difficult for the vapor to escape. Ideally nucleation will only occur at the top of the wick; wick design can limit this effect. In a loop heat pipe the heat travels through the evaporator to the wick and boiling can occur of either surface, but is usually occurs at the surface leaving space for vapor to easily escape. The boiling limit becomes less important with small wicks and thin evaporators because there is less nucleate style boiling and more direct evaporation from interline effects, which are described in section 2.3.

2.2 Key Physics

2.2.1 Surface Tension and Capillary Effect

Surface tension is a key element of micro fluidic flows and ultra-thin passive thermal management system like the μ LHP utilize the advantages of additional energy created at an interface between two phases. Surface tension is a phenomena where there is extra Gibbs free energy because molecules in an interfacial region (the boundary between liquid and

gas) behave differently from their bulk states. One way of thinking of surface tension is in terms of the system as a whole, having more energy if an interface exists. That interface adds potential energy, as if the surface had an additional force on it that is minimizing the surface area at the interface. This force is analogous to a membrane being stretched by tension. When liquid is in contact with the surface the equilibrium relationship at the interface is determined by the opposing forces between the materials and can be defined by the Young-Laplace equation.

$$P_1 - P_2 = \gamma_{12} \left(\frac{1}{R_1} + \frac{1}{R_2} \right) \quad (2.1)$$

Where R_1 and R_2 are the radii of curvature of the interface in two orthogonal directions. The sign of the radii will change depending on where the surface is curved up or down. When the surface is circular or in a round tube the radii are the same and the equation simplifies.

$$P_1 - P_2 = \frac{2\gamma_{12}}{R_1} \quad (2.2)$$

The simplified equation is also the pressure drop relationship across the interface for liquid in a round capillary. This model with that added interfacial force shows the benefits these interfaces. The capillary pressure increases as the radius of a capillary decreases making this a useful method for setting up a specified pressure drop to start circulation in a μ LHP. The wick uses these capillary forces to create a large positive pressure so that as liquid evaporates the vapor moves towards the lower pressure area near the condenser. This starts and maintains the loop in the system, where vapor moves away from the evaporator and then once it condenses back into liquid it is pulled into the wick by the small pores. This is a physical behavior that only works with small flows and the system described in the work presented here, uses this to its best advantage to increase the functionality of the system.

2.2.2 Phase Change and Boiling

Understanding boiling and phase change phenomena starts with a more realistic view of fundamental thermodynamics. Starting with a system containing a pure and isotropic substance in an isolated environment that is free from external forces (including work and chemical reactions, ect.) [14]. Approximating this system, can be done using several phase diagrams to show the pressure (P), temperature (T) and specific volume (v) ranges, that cause this substance to exist in its three possible phases. For the purposes of boiling and phase change only the liquid and gas phases are shown in the diagrams. A series of P-v, T-v and P-T graphs can show the simplified thermodynamic relationship between the liquid and gas phases. The P-v diagram shown in figure 2.5 is simplified and in order to understand phase change a molecular view of the interaction between liquid and vapor is more accurate and gives a clearer qualitative analysis.

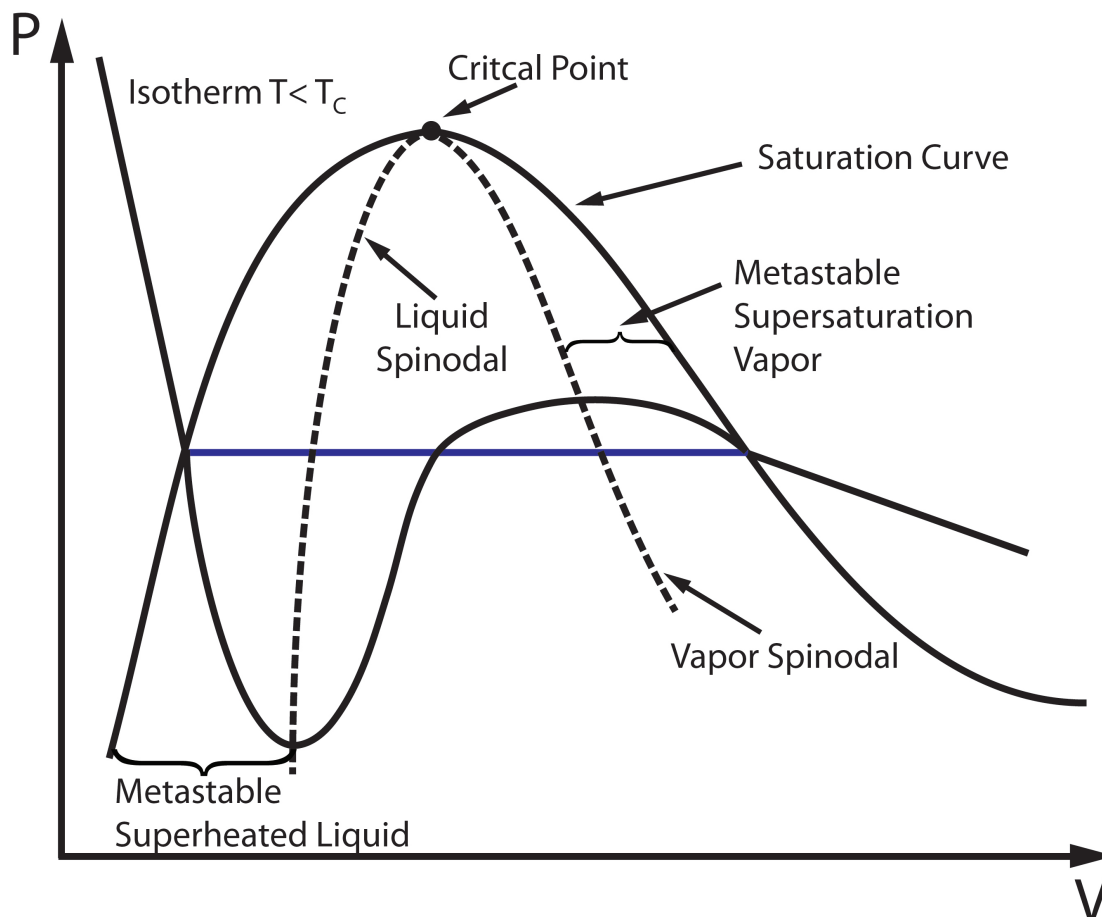


Figure 2.5: P-v diagram of a fluid with a temperature isotherm that captures the more realistic behavior of the fluid using the Van der Waals equation of state.

At the molecular level the region where bulk fluid and bulk vapor interact is called the interfacial region and has a lot of complexity in it. In this area there are many forces acting on each molecule that change the overall fluid behavior from the classical model to a more statistical thermodynamic model. The interactions that take place are described in great detail in other resources, but the conclusion of a molecular model analysis leads to idealized property relations for a van der Waals fluid. (For a derivation of this look in Carey [14]). With this model the shape of the isotherms in a P-v diagram shifts to incorporate molecular effects in nanoscale control volumes, see figure 2.5. What this diagram shows, is that the assumption that properties are well defined is incorrect and that properties of fluids can fluctuate due to random molecular motion. The van der Waals equation of state can be written in its dimensionless form:

$$P_r = \frac{8T_r}{3v_r - 1} - \frac{3}{v_r^2} \quad (2.3)$$

Where T_r , P_r and v_r are properties normalized with their critical point, shown here:

$$T_r = \frac{T}{T_c}, \quad (2.4)$$

$$P_r = \frac{P}{P_c}, \quad (2.5)$$

$$v_r = \frac{v}{v_c} \quad (2.6)$$

In the van der Waals equation T_r defines a dimensionless isotherm that when below the critical temperature has a local minimum and maximum, seen in figure 2.5 [14]. This evaluation of the interfacial region is important because from this and understanding of the non-equilibrium metastable states can be determined and these states represent the ranges where phase change and boiling begins. The region where the system can be in a metastable state is determined by the limit of intrinsic stability or the spinodal limit which is within the vapor dome on the spinodal curve, see figure 2.5 [14]. The spinodal curve is determined by differentiating the van der Waals equation and setting $(\partial P / \partial v)_T = 0$, for more on how this is done see Carey [14]. Again these metastable regions are important because within them is usually where droplet formation either evaporation or condensation occurs. Physically what typically occurs is there is liquid on a solid surface and heat is transferred through that surface and the liquid immediately adjacent to the surface may approach and then rise above the saturation temperature. The hottest liquid will be at the surface and thus a vapor embryo will form and nucleate boiling will begin. One key take away from the metastable states is that usually the liquid is superheated before boiling begins meaning that the boiling surface has to be above the saturation temperature. This is not an ideal condition for a thermal management device, nucleate boiling is actually avoided and the surface in the evaporator is designed to create thin film evaporation or interline evaporation. In the following sections convective pool boiling and convective flow boiling are discussed, these are two methods of heterogeneous or surface boiling. These types of convective boiling and the fundamentals associated with them were the foundation of the evaporator design used in the μ LHP.

2.2.2.1 Convective Pool Boiling (CPB)

Pool boiling is characterized by mostly still bulk fluid, any motion in the fluid is caused by free convection and mixing from bubble growth and movement. Pool boiling is usually categorized by the temperature of the bulk fluid, either subcooled or saturated pool boiling

can occur. Subcooled boiling is characterized by the bulk fluid temperature being below the saturation temperature of the liquid. This can cause the surface bubbles to condense back into the liquid after they evaporate. One can imagine this as a loop, the surface bubbles are created and destroyed continuously keeping the energy at the surface of the fluid instead of moving away as is desirable in phase change. Eventually the increase in energy will cause some of the vapor to not re-condense, but this is a less stable and less efficient form of boiling. Saturated boiling, uses liquid that slightly exceeds the saturation temperature of the liquid, not so much that the liquid fully evaporates, but so that any bubble that forms will detach and be removed from the surface by buoyancy forces. This is more ideal and common any paper using CPB will publish the amount of saturation used on the surface in addition to other data.

In literature there are many studies on CPB, usually the goal of this work is to increase the critical heat flux (CHF). CHF determines the point at which the surface will dry out. Dry out occurs when nucleation becomes unstable and a vapor layer forms over the surface causing the surface temperature to skyrocket. In most cases this is equivalent to system failure or breakdown. The pool boiling regimes are categorized by a boiling curve that consists of four modes. Further information on these modes can be found [15], these modes are similar to the more relevant flow boiling modes that are explained below. No more than a basic understanding of pool boiling is necessary to understand the rest of this work. This is just a small portion of all of the information on pool boiling that is available, other resources can be found [16][17].

2.2.2.2 Convective Flow Boiling (CFB)

Flow boiling in contrast to pool boiling, has a direct motion of the bulk fluid flow in addition to the flow from buoyancy. Another way to think of this is a type of boiling with forced convection over the boiling surface. In the case of flow boiling, geometry is important. Fluid can flow externally over a heated plate or wire, or internally in a pipe. Another common name for the internal flow boiling found in a pipe or duct, is two-phase flow. This is due to the characteristic rapid change from liquid to vapor. The physical difference of the external and internal flows is the location of the heat source. External flow has heat entering from one source usually on the side or center of the flow, where there is still bulk fluid motion surrounding the heat source, see figure 2.6. Internal flow is constrained in pipes of varying cross sections, where the heat source is the entire outside of the pipe, see figure 2.7.

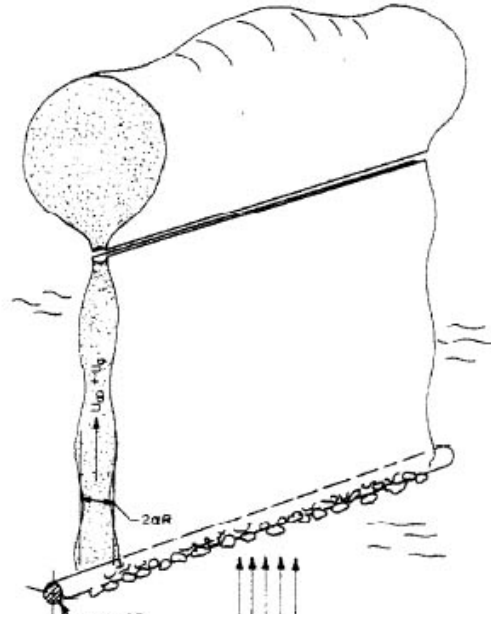


Figure 2.6: Convective flow boiling model over a wire from 1970s [18]

External flow was experimentally tested back in the 1970s, both high flow velocity and subcooling are known to increase the critical heat flux (CHF) in these cases. For certain geometries there are known correlations between the critical heat flux and a dimensionless constant called the Weber number, shown below [19].

$$We = \frac{\rho_v V^2 L}{\sigma} \quad (2.7)$$

This constant (We) is the ratio of inertial forces to surface tension forces. In this equation, L is the length of the heat source. A common geometric alteration can be made for a cylinder in cross flow, where L is replaced with D , the diameter of the cylinder. This constant is substituted into various correlations to determine the critical heat flux based on the bulk fluid velocity. All of these correlations assume subcooled liquid and no extensive vapor generation. External flow is not a flow type that occurs in the TGP device and is commonly divided into boiling modes, which is common is pool boiling and two phase flow, more information on this type of flow can be found in [19]).

The other type of CFB is characterized by an pipe with internal flow and constant heating of the pipe walls, called two phase flow. This scenario experiences may modes of boiling and vapor formation including, liquid convection, subcooled boiling, saturated boiling and complete vapor. During internal two-phase flow the initial liquid and final vapor stages have the same mode of forced convective heat transfer, often with different heat transfer

coefficients, but the type is the same. The challenging aspect is the change between the two phases. Figure 2.7, shows the different regimes and it is clear from the heat transfer coefficient graph on the right that a liquid film that is on the verge of transitioning into mist has the best heat transfer and it can be concluded that this is generally because there is increased evaporation in thinner films of liquid. This technique of trying to increase overall heat transfer coefficients by maximizing evaporation was used in the creation of the evaporator for the μ LHP device; creating a lot of thin film or interline regions is one way to maximize evaporation.

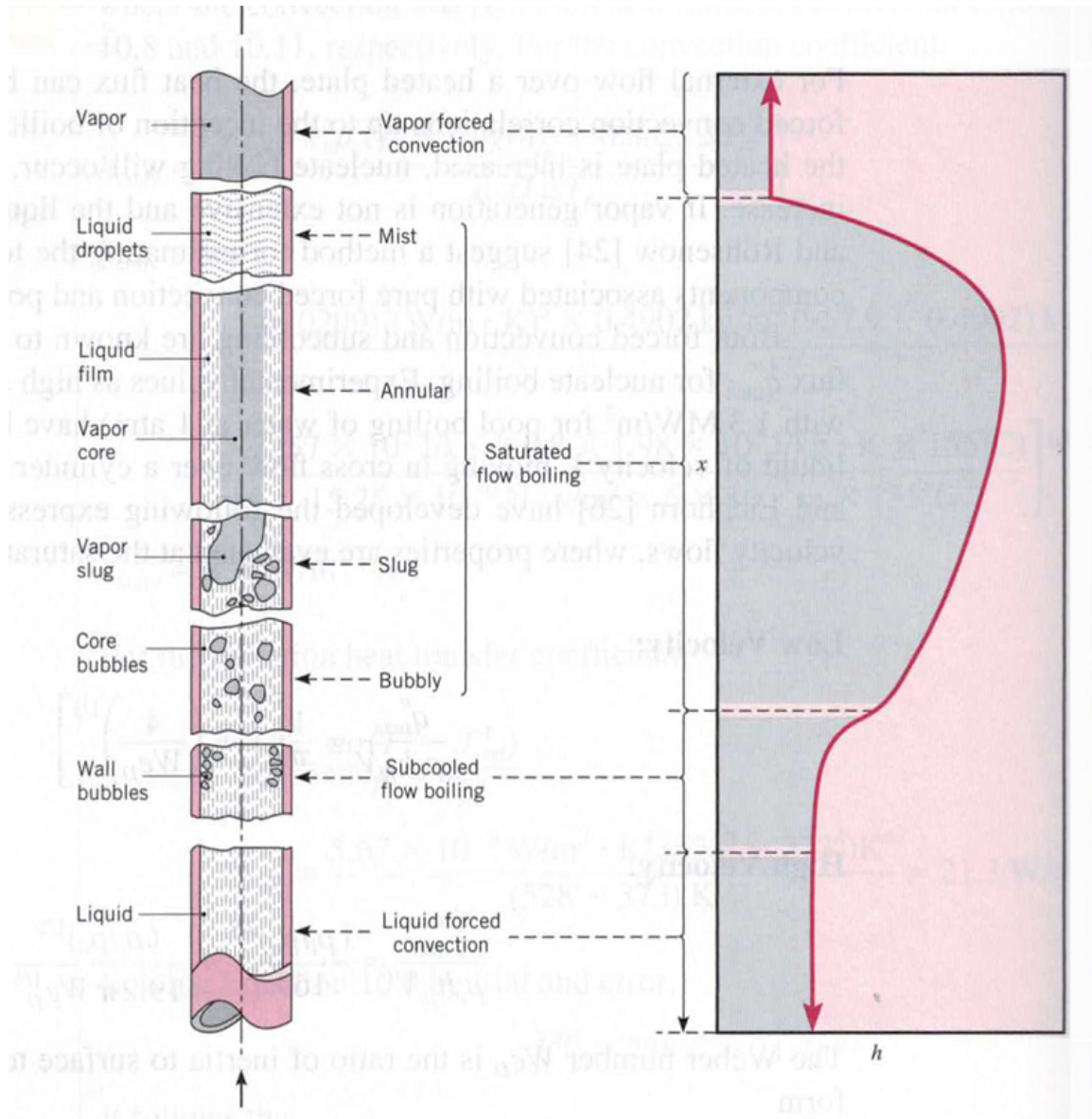


Figure 2.7: Image of the different regimes present during convective boiling in a pipe [19].

2.3 Literature Review on CPB

Most of the relevant past work that was reviewed focused on convective pool boiling (explained in section II.2.2.2) and the structures and methods used to enhance pool boiling. This topic was reviewed because there is a massive amount of research on this subject. As was explained in the previous section the TGP doesn't use CPB it uses CFB, which has different requirements, but the geometries used in the CFB tests were selected from CPB

research. It was determined that CPB results were a good starting point, as they were the closest flow correlation in recent research. The review focused on two areas of CPB research, surface structures, like pin fins, and surface roughness.

2.3.1 Surface Structures

An excellent review in 2004 by Honda gives a nice background of many different enhanced boiling surfaces. All of these surfaces are pool boiling surfaces, with the goal of increasing the critical heat flux (CHF). The critical heat flux is the amount of flux that once added cause the evaporation equivalent of system failure i.e. dry out. The range of techniques used to increase the critical heat flux described in this paper is diverse. The size of the surface features vary, but the smallest pin fins are roughly 60 microns high and 10 microns thick. These are similar to the final dimensions chosen for the evaporator pin fin tests. The real take away of the paper is the conclusion that at that time micro-pin fins were the most effective structures for enhancing nucleate boiling and increasing the CHF [16]. Since the paper was published work in surface roughness has increased the CHF even high than the pin fins ever did. The same paper also addresses an issue that has yet to be a problem in the μ LHP system, which is incipient superheat. This is when a surface must be heated well above the boiling point of the liquid due to high surface wettability. Another way to think about this is that if there is no contact angle between the liquid and the surface, it becomes more energy favorable for the liquid to stay on the surface rather than change phase. The review discusses methods of combating this effect with micro-roughness, micro-pores or micro-reentrant cavities. These issues will not be addressed unless incipient superheat becomes a problem.

2.3.2 Surface Roughness

Pin fins were an extensive topic of research in CPB, but as nano research became more popular the shift moved to using surface roughness to increase the CHF. Previous work done by Jung in 2006 shows the effects of submicron scale roughness [17]. The take away from this work is that roughness increases the possible nucleation sites. It follows then, that more nucleation sites increases the boiling area and the nucleate boiling heat transfer coefficients. This conclusion was supported by the research and it also shows that the critical heat flux of these surfaces increases linearly with effective boiling area [17]. Other papers have emerged in recent years including promising work from Wang in 2012 that essentially made the roughest surface to date and this pushed the CHF higher than ever before [8]. This technique of creating rough surfaces was not utilized in the evaporator design, but the research shows that the RMS roughness of a surface does effect critical heat flux especially when subcooling is involved. Surface roughness is an area where the split between CFB and CPB becomes more defined. Rough surface create a lot of flow resistance and would most likely increase pumping requirements in the TGP, which is undesirable. So, although this research create critical heat fluxes that were higher than ever before the work would

theoretically not transfer to the μ LHP as well. This area of could be explored in the future, but has not been explored in this work.

2.4 Literature Review on Interline Region

The theory of interline evaporation comes from nucleation of bubbles on a surface. The interline evaporation region shown in figure 2.8 is the region that the majority of evaporation occurs in during the formation of vapor bubbles. The figure shows two red highlighted regions that are interline regions. These regions have an ideal thickness that allows for a large amount of evaporation in these regions. The thinner region closer to the center of bubble, called the absorbed region, has strong surface tension and Van der Waals forces that require a lot of energy to detach from the surface. Alternatively the thicker regions closer towards the bulk fluid, called the macro region, can still evaporate, but the fluid must reach a high temperature. The added superheat requires more energy, which is undesirable. As one can imagine, with one region being just right for evaporation this region has been recently studied under the hopes of creating more efficient evaporators. These studies influenced the design of the pin fins chosen to the evaporator study.

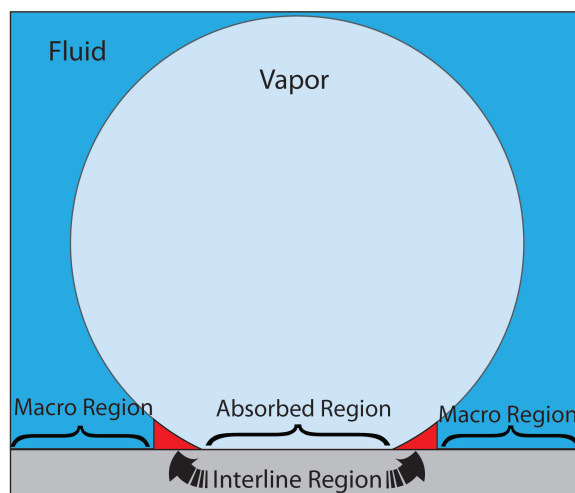


Figure 2.8: Image showing the 3 different regions of evaporation in a bubble.

2.4.1 Interline Behavior

The evaporator design was focused around identifying the behavior of the interline region and incorporating that research into the design. The behavior of the interline region was studied at Purdue by Dhavaleswarapu with hexane in 2009 [20]. There were three important conclusions from this work. Two of them were that the shape of the meniscus changed with different heat fluxes and evaporation rates. The other conclusion was a measurement of the

percentage of the overall evaporation that was occurring in the interline region. (Note: The interline region is not well defined geometrically, but for this paper they define it as the region from the tip of the visible meniscus up 50 microns to the bulk fluid, sometimes called the sub-region.)

The images from the work at Purdue show that the interline region behaves favorably for high flux devices with lots of evaporation. Figure 2.11A shows this favorable behavior. It shows that as the evaporation rate increases the interline region increases or spreads further down the channel. Figure 2.9B, shows that the region also increases the contact angle along the sidewall as heat flux increases. These graphs provide a better picture into the behavior of the interline region. As evaporation rates increase the liquid does not dry up and shrink, but grows. That is exactly what a high flux device would need to avoid drying out. Additionally, as heat flux increased the contact angle increases, which causes increased capillary pressure and creates a greater draw of fluid. Understanding the behavior of the interline region is important to increasing the thermal efficiency of the device. The interline region reacts favorably to high heat fluxes and evaporation rates. This makes it an ideal region to focus this evaporator design around. If a surface can increase the formation of these interline regions, it should create good capillary forces and high evaporation rates in a device.

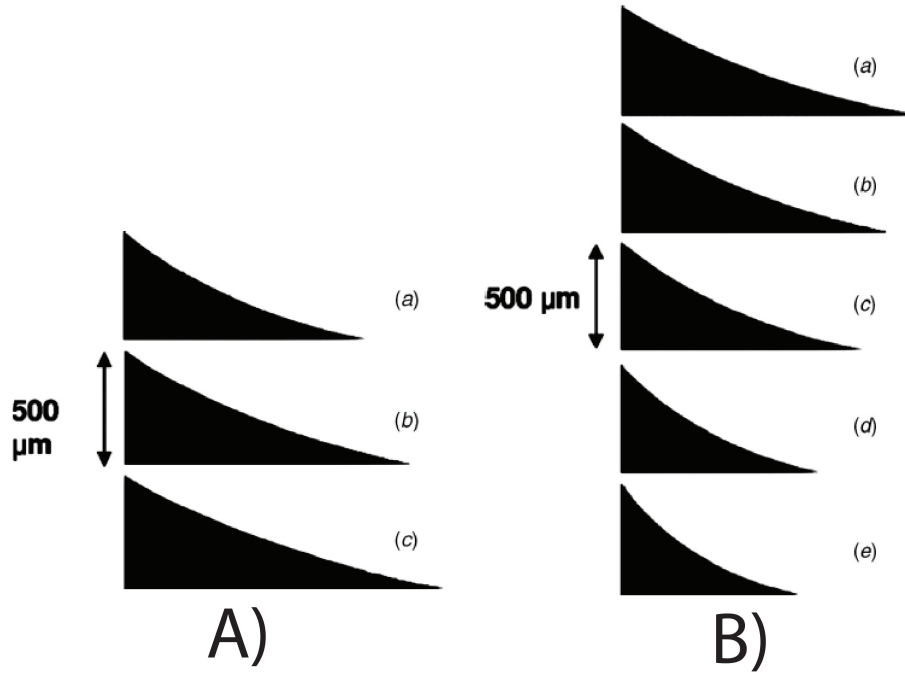


Figure 2.9: A) Two dimensional meniscus images at a heat flux of 990 W/m^2 and different evaporation rates: a) $8.35 \times 10^{-7} \text{ kg/s}$, b) $8.8 \times 10^{-7} \text{ kg/s}$, and c) $9.35 \times 10^{-7} \text{ kg/s}$ [20]. B) Two dimensional meniscus images at an evaporation rate of $8.35 \times 10^{-7} \text{ kg/s}$ and different heat fluxes: a) 810 W/m^2 , b) 950 W/m^2 , c) 990 W/m^2 , d) 1030 W/m^2 , and e) 1190 W/m^2 [20].

The conclusion that increasing the interline region will increase the efficiency of the device if further supported in the same work, at Purdue. Figure 2.10 shows the percentage of evaporation in the interline region as a function of heat flux and evaporation rate. The table clearly illustrates that an increase in both heat flux and evaporation rate increase the evaporation percentage. This information confirms that creating large interline regions is the key to efficient evaporation. These conclusions led to further research on what geometries dimensions were necessary to maximize the interline regions.

q_s''	Ψ		
	$8.35 \times 10^{-7} \text{ kg/s}$	$8.8 \times 10^{-7} \text{ kg/s}$	$9.35 \times 10^{-7} \text{ kg/s}$
810 W/m ²	66	67	69
950 W/m ²	70	73	78
990 W/m ²	68	73	78
1030 W/m ²	76	79	84
1190 W/m ²	77	80	87

Figure 2.10: Variation in the percentage contribution of the sub-region heat transfer to the total (%) with heat flux (q_s) and evaporation rate (ψ) [20].

2.4.2 Interline Geometry and Dimensional Research

The other paper that was integral in the formation of the interline design models was a paper by Nilson from Sandia National Laboratories [21]. This paper was focused on creating steady evaporating flows with continual evaporation and high capillary pressure. The importance of these variables in the scheme of the device was to create a surface that draws the fluid along, limiting dry out and improving overall device function. These issues are important in CFB, but not in CPB because the pins cannot stop the flow or the device will dry out. The graph shown in figure 2.11 demonstrates that certain geometric ratios provide more continuous evaporation. Interestingly, all of the numbers that asymptote to the center line in the figure, actually experience a region of suppressed evaporation. This is where the contact angle stays constant and the fluid has no pressure difference to drive the flow. This creates a lot of resistance in the channel and could slow the flow. Another way to look at this is by considering the research from Purdue that explains that increased evaporation causes the meniscus to spread [20]. The spreading is fluid motion and any region of suppressed evaporation is causing less flow by limiting spreading. In addition, there is only so much pressure created in the CPS pumping system, so any added resistance in the evaporator is undesirable. This paper explained that by incorporating the optimal ratio of depth to width, of the channel, the resistance of the flow in the channel should be lower. This ratio has been incorporated into the pin fin surface structure designs. One limitation of this work is that it runs these models only for rectangular channels and the evaporator tests incorporate two additional types of pin fins, square and oblique. After a thorough exploration of other literature there is no other leading work on the ratio of dimensions for these pin fins, so it was decided that the same ratio would be kept for the other pin fins in order to maintain consistency. This interline theory lead to the dimensions of three evaporator surfaces.

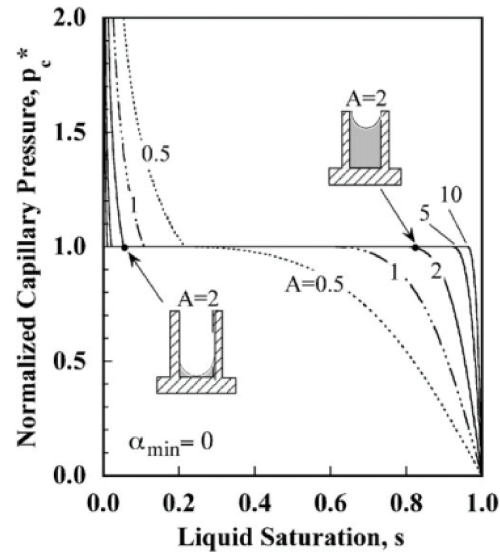


Figure 2.11: Variation of normalized capillary pressure with fractional saturation (liquid volume fraction) for a minimum contact angle of 0° . Plateau corresponds to dead zone. $A = D/H$ [21].

Chapter 3

Preparatory μ LHP Component Development and Fabrication

3.1 Interline Optimized Evaporator Development and Fabrication

As described in section 2.2.2.2, the interline optimized evaporator work focused on creating an efficient evaporation surface to work in a convective flow boiling (CFB) environment. The work tested four surfaces with features on the scale of 10 microns and flow rates less than 3 mm/s. The four surface types and the configuration of the testing can be seen in figure 3.1. The flow rates were chosen because they realistically simulate the kinds of flow rates found in the CPS wick. The tests were performed in an open loop, which used a high speed camera and an IR camera to record visual and temperature profile data of the flow behavior over time. Heat transfer coefficients and data on the isothermal stability of the surfaces were also collected. The ends of the chips were open to atmosphere, creating an open loop system; the steam created in the evaporator was not collected or condensed in these experiments. The evaporation surfaces were tested to the point of dry-out at three different pressure potentials: 150, 650, and 1150 Pa, at a range of powers from 25 W/cm² to 50 W/cm² [22].

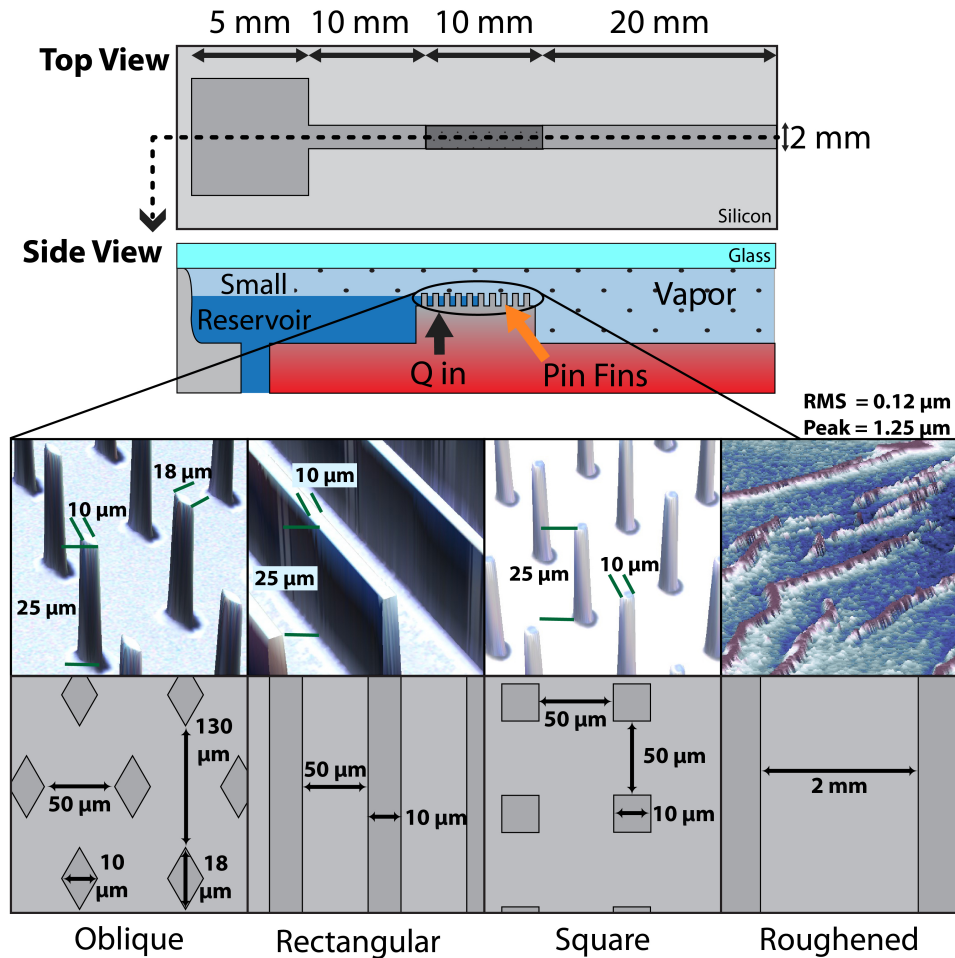


Figure 3.1: Schematic showing the pin fin tested chips, with a sideview of the liquid filling process. This image also includes 3D laser confocal images of the pin fin surface and top down schematics with measurements of the features.

The results from the experiments did not exactly match the expected flow behavior based on convective flow models. Actual observed flow patterns avoided most bubble or slug flow, and transitioned straight from liquid forced convection to annular flow. As evaporation started, an interline region formed at the leading edge in front of a larger meniscus, shown in figure 3.2. One purpose for using pin fins is this interline leading edge, which is efficient at converting liquid to steam without creating bubbles, as explained in section 3.1. The pin fins testing looked at three main features to determine what the best surface was stability, time to dry out and heat transfer coefficient. The final conclusion of the work was that at higher loads, the blank control surface had the best performance and the longest times before drying out. Additional testing supported conclusions that, under more realistic operating conditions, complex surfaces may be preferable due to their relative stability.

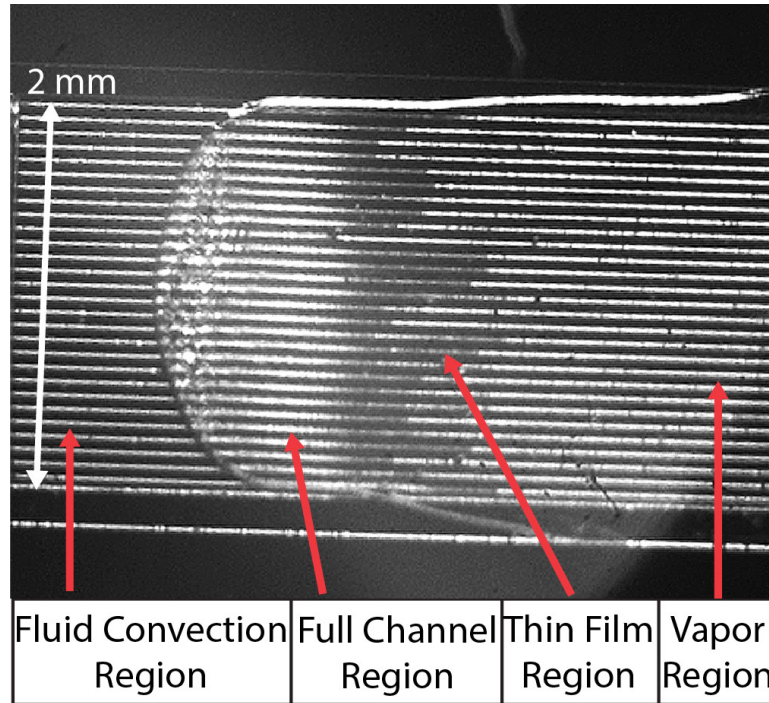


Figure 3.2: Experimental image of the rectangular evaporator surface with developed thin film region.

The stability of the evaporation in the chip was an unexpected criteria added after the data was obtained, due to the sometimes large and unexpected fluctuations in the chip temperatures. There were essentially two types of temperature fluctuations, there were continual oscillations and sporadic peaks above the average. The continual fluctuations can be seen in Figure 3.3. These oscillations were small for some chips, but as the fluxes increased larger more noticeable oscillations were observed. For the most part any chip with continual fluctuations at larger temperature differences were in or close to a failure mode. That being said the allowable amount of fluctuation must be determined by the operating guidelines. The sporadic fluctuations were difficult to distinguish from the dry out conditions, but once the devices reached a temperature above 200°C testing was stopped regardless of fluctuations or dry out for the safety of the experimental apparatus.

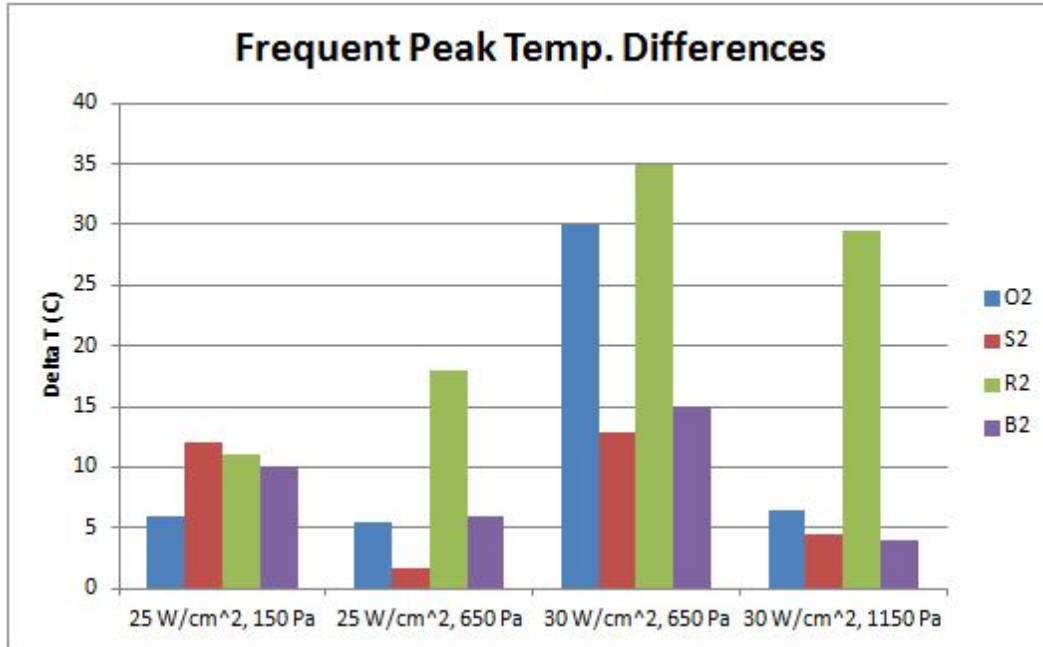


Figure 3.3: Graph showing the temperature range of fluctuations that occur repeatedly.

Dry out was another metric tested for in the interline optimized chips. This occurs when all of the water in the chip evaporates away and the temperature stops regulating itself and increases drastically. For the most part dry out occurs when thermal runaway is witnessed. The easiest way to spot thermal runaway is to look at the slope of the dry test, where there is no evaporation, just increasing temperatures. The slope of this dry test line, shown in Figure 3.4, is when thermal runaway is occurring. In this graph, it is clear that the rectangular pin fin, in green, goes into thermal runaway almost immediately, as the slope of the pink line and green line are identical. The green line representing the rectangular chip does not recover and dries out in less than 5 minutes. The oblique chip, in black, follows a similar pattern, but the square chip, in red, has spikes of thermal runaway with short recovery periods before thermal runaway occurs again. This spiking behavior delays thermal runaway for several minutes. Overall, this spiking means that dry out can be determined by matching the slope of the two lines, but true dry out is actually when the recovery from thermal runaway stops. This is a difficult metric to monitor and runs the risk of melting the test set up, so a simple cap was put on the chips, that once the temperature was above 140 the chips were dried out. (Note that the graphs do not show the temperature increase up to 140 if the increase is smooth. This allows for the more interesting lower temperature oscillations to be clearer.) Figure 3.5, shows the time until dry out for the 4 chips at 1150 pressure potential and three different power fluxes. It is clear from this graph that the point of dry out is not a gradual slope, but a steep one. At a set pressure potential in only 10 W/cm² a chip can teeter between working well for 60+ minutes and drying out in 5. This

dry out limit will be important when setting operational limits.

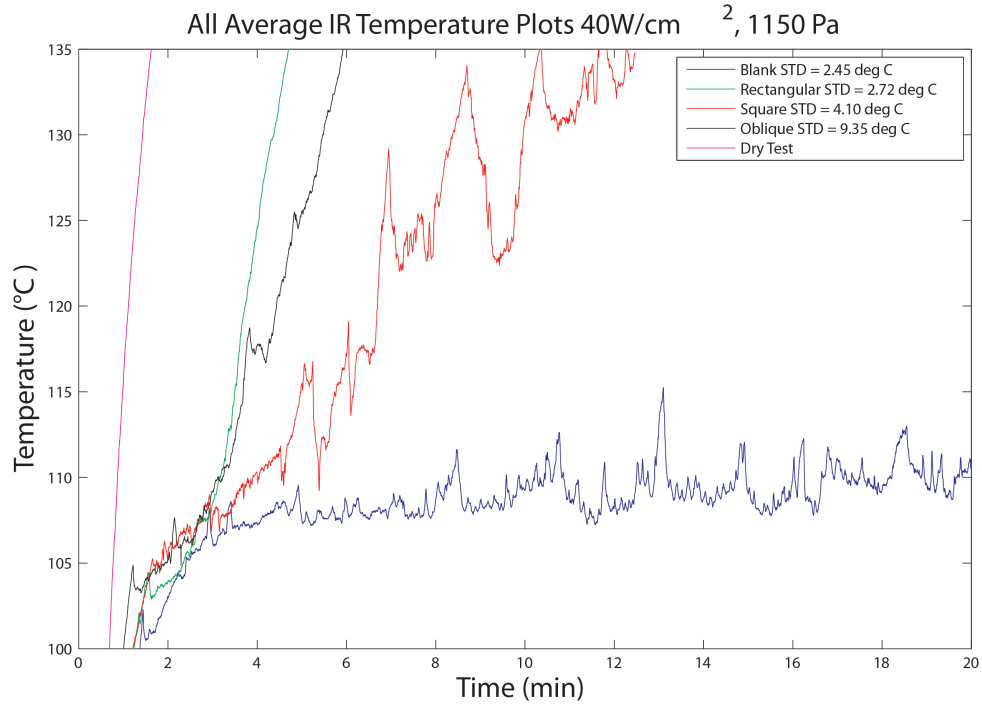


Figure 3.4: Graph showing results from the 4 chips at 40W/cm² and 1150 Pa.

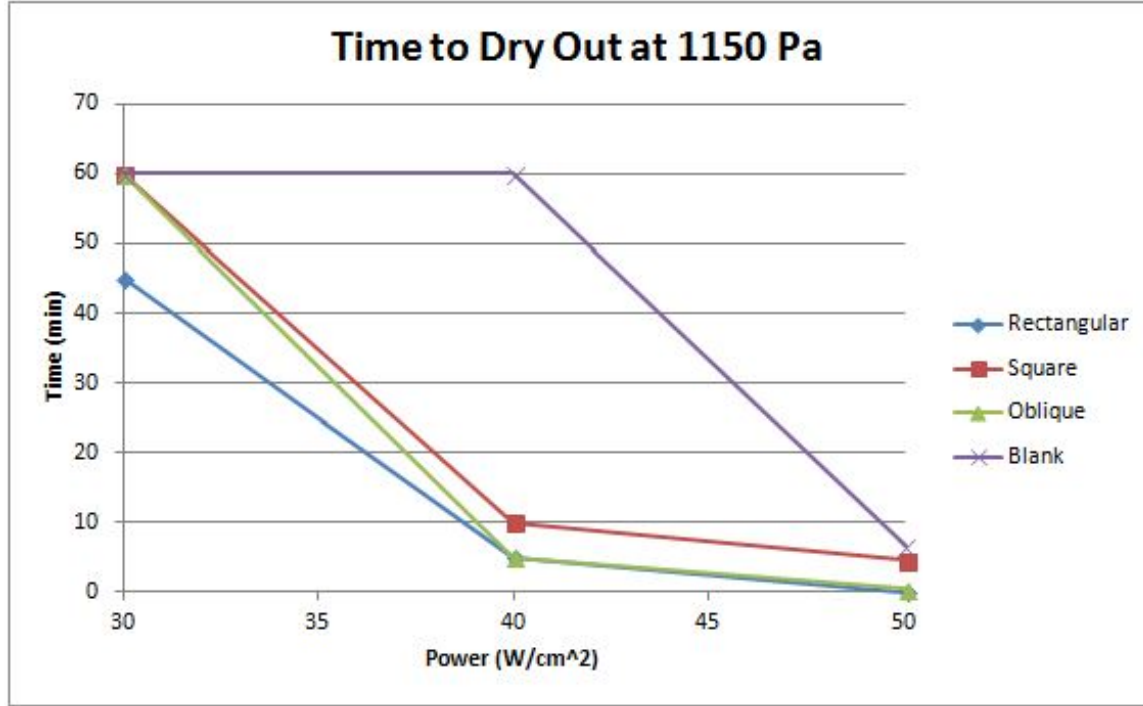


Figure 3.5: Graph of time until dry out for each chip at 1150 Pa and 3 different powers.

The final metric was calculated from the data to be the convective heat transfer coefficient (h) across the evaporator surface. This coefficient is an experimentally determined parameter it has units of $W/cm^2 K$ and its value depends on many characteristics of the fluid and the flow including the surface geometry, the fluid properties and the type of flow. The equation used to calculate h is shown here:

$$h = \frac{q}{T_s - T_{sat}} \quad (3.1)$$

Where q is the applied heat flux in (W/cm^2), T_s is the surface temperature and T_{sat} is the saturation temperature at atmospheric pressure. Once calculated the heat transfer coefficient can be used to approximate the flux capabilities of the surface when integrated into a full system. Figure 3.6 shows the calculated heat transfer coefficients at different fluxes. The graph shows a linear relationship between increasing heat flux and heat transfer coefficient. In a simple system the heat transfer coefficient can be approximated as a constant, but in complex system like this one the heat transfer coefficient changes. For these enhanced evaporation surfaces the heat transfer coefficient can be approximated using the following equation derived from the experimental results.

$$h = C_1 Re Pr \frac{A}{L} + C_2 \quad (3.2)$$

Where Re is the Reynolds number, Pr is the Prandlt number and C_1 and C_2 are experimentally determined constants found in table 3.1. The number in the table show that the roughened surface has the largest C_1 constant, which corresponds to the steepest slope meaning that it shows the highest heat transfer coefficient as flux is increased. This is one of the many reasons that the roughened surface was determined to perform the best in these tests.

Table 3.1: Heat Transfer Coefficient Constants

Surface	C_1	C_2 (W/cm ²)
Roughened	219	10099
Square	94	15640
Oblique	57	13080
Rectangular	27	13619

This table shows the values of C1 and C2 that can be used in equation 3.2.

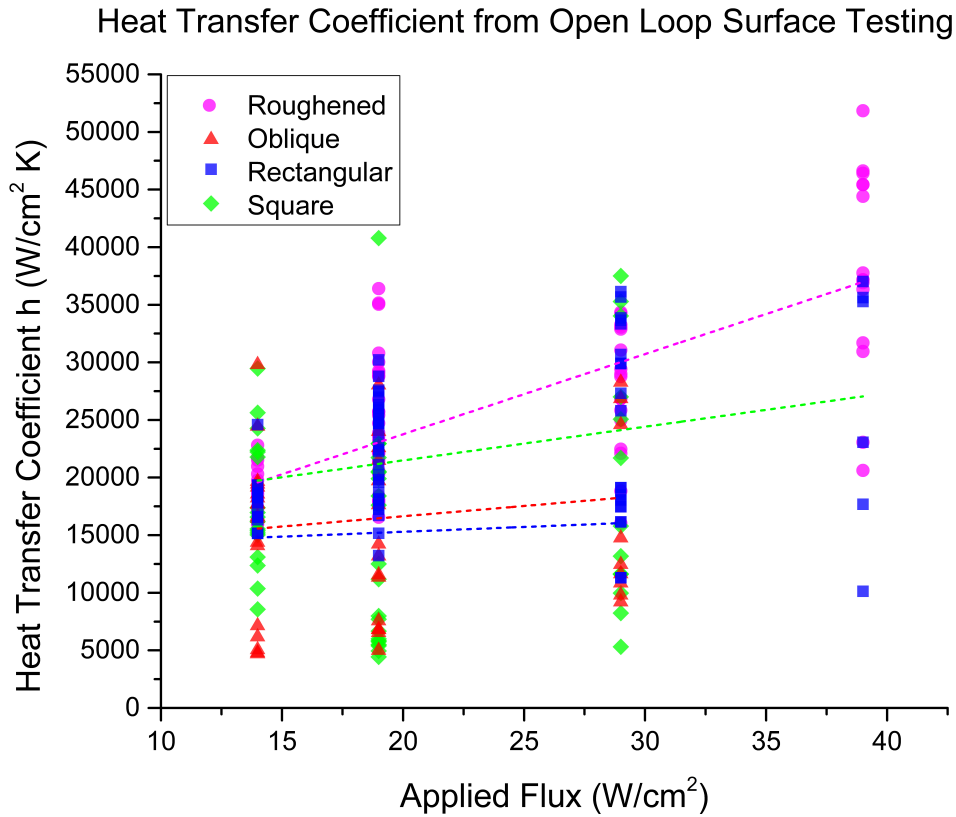


Figure 3.6: A graph of the heat transfer coefficients of the pin fin surfaces.

From these experiments it seems clear that the best structure is a roughened surface with no defined structures at all, depending on the desired operation of the chip. The roughened chip out performs all of the others in its ability to withstand high heats as well as time before dry out. The roughened chip also had low standard deviation between trials, so it performed the most consistently. The square chip is the second best shape, it performs well in the lower power regimes, with lower potential pressure. The one benefit of the square surface structures was seen in the $25 W/cm^2$ tests at 650 Pa. In this test the square chip had the least fluctuations, as well as the smallest standard deviation in its temperature. At this operational interval this chip is able to keep one temperature for more than an hour with a max deviation from the mean of 3 degrees and an average deviation of 1 degree. This could be useful depending on the application of these surfaces and the required operation interval.

The operation interval defines the conditions that a surface would work in, and the desired operational conditions. It has been established that the blank chip has the best performance for the metrics defined in this paper, it lasts the longest before dry out, it has minimal

fluctuations and it can withstand the highest tested fluxes. This chip clearly meets the requirements, but as this project continues on, other applications should be considered. One set of applications are things that require elevated temperatures, but need constant reaction conditions even as the reaction produces heat. In a case like this, another surface structure might be more beneficial. The square chip is an obvious example, because as was previously mentioned in a certain condition, it could cool 25 W/cm^2 with 1 degree accuracy with an max fluctuation of less than 3 degrees. Another possible scenario is a system with a lot of large temperature fluctuations that needs to be kept from exceeding a certain threshold, while still reacting to a variety of thermal fluctuations. In a case like this, the oblique surface, which has the ability to recover from large temperature fluctuations without drying out, may be the best option. Most of these operational intervals are yet to be fully explored, but as they arise there will be certain surfaces that can achieve the needed conditions. Application focused testing will be the next likely step for the research and that is continued by the surface integration into the hybrid μ LHP device.

3.2 CPS Development

As explained in the outline of the TGP device the pumping mechanism to make the device circulate is a specially design CPS wick with zinc oxide nanowires. The development of this wick was done by Dr. So and the specifics of this project can be found [23]. Overall he created CPS with $4.5 \mu\text{m}$ diameter pores that reached a depth of $150 \mu\text{m}$ and porosity of $11,000/\text{mm}^2$, see figure 3.7 [1]. The zinc oxide nanowires penetrated $5\text{-}10 \mu\text{m}$ into the pores and prevented the liquid in the pores from pinning at the top of the pores instead liquid would pool on top of the CPS. This novel approach, design and creating of a wick with nanowires allows fluid to spread across the entire surface without added external pressure [1]. This is a huge improvement over traditional wicks because usually the liquid stays trapped in the pores and the wick cannot spread liquid to a surface that it is not in direct contact with. Additionally, in this work wicking performance was verified and the volumetric flow rate through the typical membrane wick was experimentally measured and compared with the improved wick showing a flow rate improvement of 12%. This work became a key precursors to creating the hybrid μ LHP and will eventually be a key component of the future all silicon high performance cooling devices.

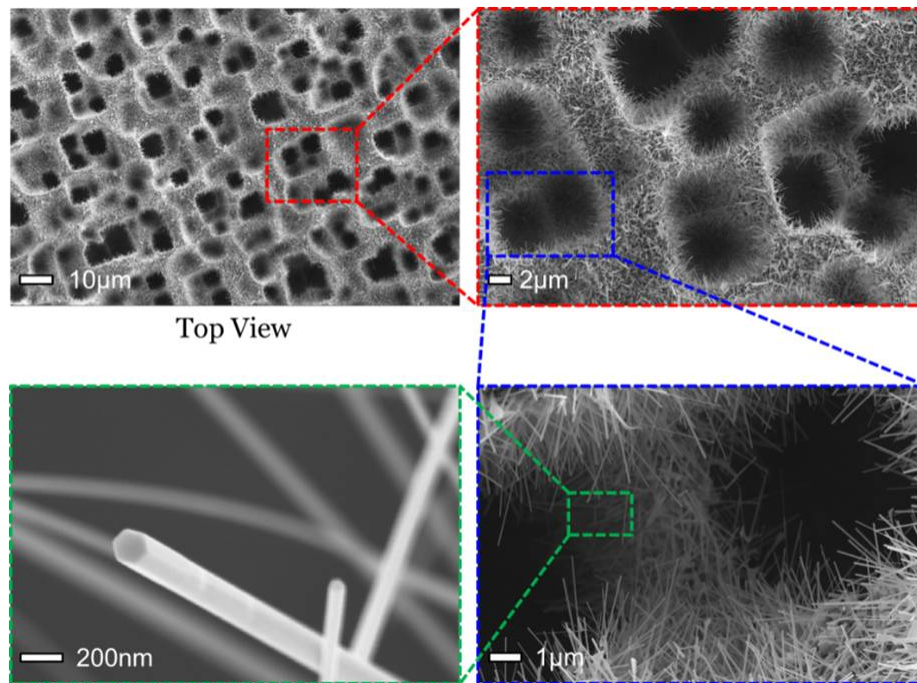


Figure 3.7: SEM images of the CPS wick with zinc oxide nanowires [23]

3.3 CPS and Interline Evaporator Integration

Loop heat pipes can have a variety of evaporator structures, to test that the novel structure created for the hybrid design would work a circulation test was started under simple conditions. In chapter 4 on system modeling there is a full discussion of start-up conditions, but to simplify the in there is mainly liquid in the wick with minimal liquid in the vapor channel getting the basic looping conditions to start is much easier. To simulate the evaporator arrangement, shown in figure 3.8, for the full system a test was devised that used a high speed camera to verify if the flow patterns were able to move through the CPS wick and into the silicon pin fin surface. The test set up can be seen in figure 3.9. In this case the wick and the evaporator were pressed together with a clamp that allowed the camera to get a side view. Under the clamp was also a heater so the evaporator could be heated and the initial stages of evaporation could be witnessed.

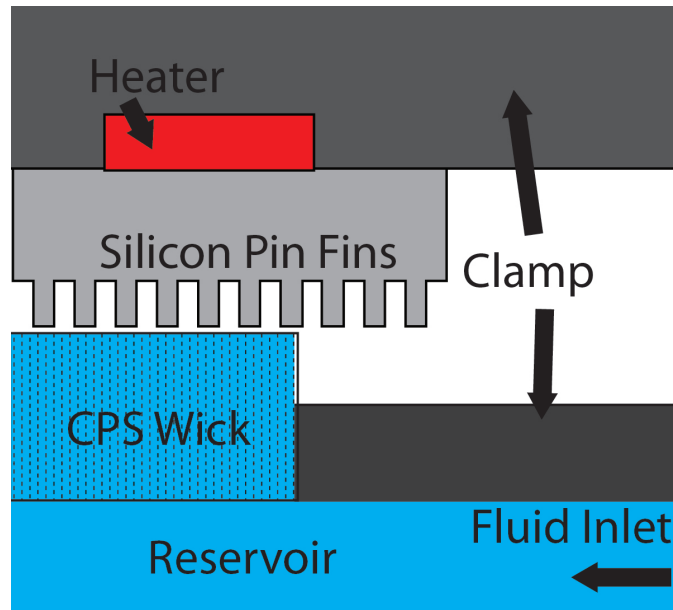


Figure 3.8: Schematic of the CPS to interline surface testing set up.

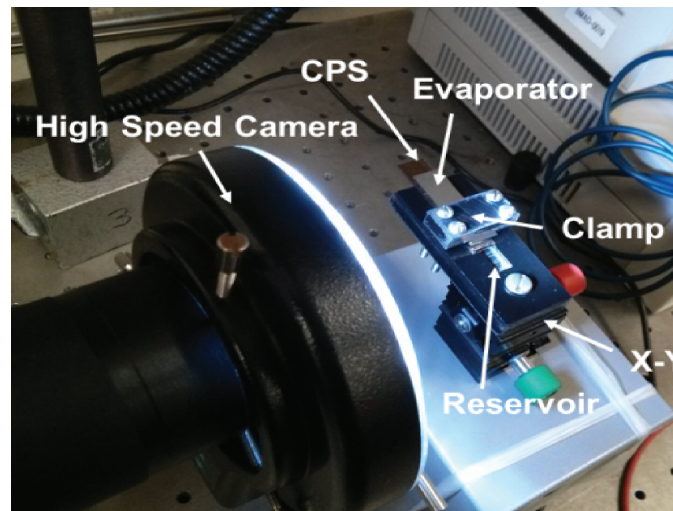


Figure 3.9: Image of the experimental set up with the high speed camera.

The first test, shown in figure 3.21, showed that when liquid was added to the reservoir it flowed through the wick (CPS) onto the evaporator surface. This process in the evaporator flow is called “bridging the gap”. This is due to the clear gap seen in the first image that in a normal non optimized wick would be difficult if not impossible to bridge because the capillary effects would stop the flow in the pores. With the addition of the zinc oxide nanowires the

liquid is brought out of the pores by competing surface energy forces and made to pool on top of the wick in this case that pooling causes liquid to touch the top surface. The third and final image shows that once the liquid touches that top surface it spreads onto the enhance evaporator surface.

After the flow tests the heater was turned on to observe the beginning of evaporation. Temperatures were monitored when the heat was applied and the chip was able to maintain a stable temperature (around 65°C) for 3 seconds and then dry out would occur and the chip would get dangerously hot. This was a repeatable although difficult test and no high speed data was ever successfully recorded from this work. This was due to the small reservoir in the experiment and it was determined that the loop did in fact start and by building this into an actual μ LHP design this problem would be solved, so that became the next steps from this research.

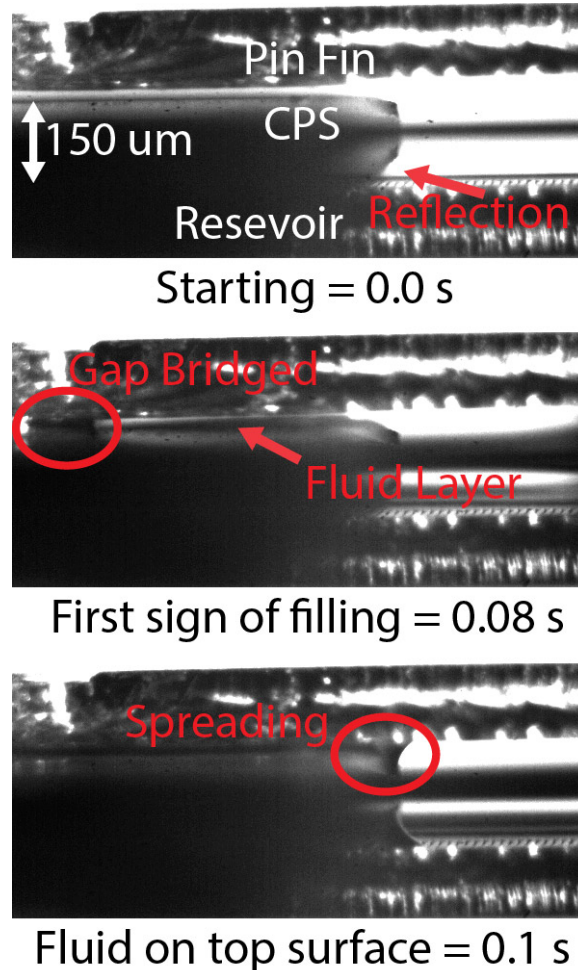


Figure 3.10: High speed camera data of the fluid traveling from the reservoir to the pin fin surface.

The goal of this evaporator design is to have the heater positioned over that spreading region rather than the wick region so that the bridge once formed is never or rarely broken. Ideally most of the evaporation should occur in the thin film region keeping the bridge region as a stable and constant refill supply. In future designs it would be interesting and useful to test the positioning of the heat to see if that affects the function of the device and the integrity of the bridge. This was overall a fairly simple test if the hybrid loop does not function at the high expected performance points it will probably be necessary to come back and tweak this experiment to be more comprehensive.

3.4 System Sealing Solution

During the initial component and system design of the μ LHP the system was meant to be entirely made from silicon and the plan was for the final non-hybrid future design to also be made of all silicon. This will make the integration of the thermal management system into CMOS chips simple with no thermal mismatch and it would also have made it easy to integrate all of the expertly designed silicon components. One prototyping difficulty with this plan is that bonding and sealing a multilayer silicon device is incredibly difficult. The plan for the future device is to try to anodically bond all three layers of silicon and if that doesn't work anodic bonding could be used to attach two layers and then glass frit or a eutectic bond could affix the third layer. This process was never attempted in the course of the research, but assuming that these issues could be worked out additional work into how to seal the devices once filled with liquid so that testing could occur, was conducted.

In order to successfully demonstrate the functionality of a LHP, the device needed to be filled with a working fluid and then sealed. A good seal between silicon and some outside material is required to prevent non-condensable gasses from entering the system as well as to prevent the working fluid from escaping. In order for this to happen the device must be sealed before the filling apparatus is removed. Additionally, the seal must be sufficiently strong to withstand the internal vapor pressure during operation of the device. These requirements created the need for a unique sealing solution that could be used during filling of the device.

Initially many sealing methods were reviewed and many tested, the methods tried and their successes and failures can be found [24]. From the conclusion of the work started by Hoople a new method of sealing devices was created and tested for the μ LHP. It was determined that the optimal sealing solution for filling and testing was to bond resealable valves to the input and output ports of the device. The resealable external valves are a major asset in the testing of a μ LHP device. With these valves devices can be sealed and the resealed if there is a problem with the first filling attempt. A device can also be filled at multiple pressures and temperatures. When the first device was designed the optimal filling method for the device was not yet characterized so the ability to attempt to fill the device multiple times with the same device saves time on fabrication and cost of production materials.

The new sealing technique proposed and tested was a eutectic bond between a silicon

wafer and steel adapter that connects to resalable valves. Simple sample versions of the full μ LHP device were deep reactive ion etched (DRIE) into silicon then bonded to glass to allow for testing of the sealing process without the potential of ruining a device that is complex to fabricate. The devices were designed to replicate the relevant aspects of the filling setup in the μ LHP and allow for a simple testing method. The device was a single channel (2.5 mm wide, 35 mm long, and 150 μ m deep) with fill holes (1 mm in diameter) at both ends. This simple geometry allowed for 14 devices to be fabricated on a single silicon wafer, compared to 2 - 3 μ LHP per wafer. The fabrication process was based on the one used to make the previous 2D silicon glass μ LHP and has been reported on previously.

The sealing process consists of two major components: a steel adapter and a miniature valve. The steel adapter is made from a 0.3" square piece of carbon steel with a M3 threaded hole. This adapter is bonded directly to the silicon side of the LHP over the fill hole. A hollow, 6mm long set screw with a silicone gasket connects the steel adapter to the miniature valve. A removable filling apparatus is screwed into the miniature valve, allowing the device to be filled with steam. Once the device is filled, the valve can be closed and the filling apparatus removed. The key bond in this system is the one connecting the steel adapter to the silicon device. Bonding is achieved by creating a metalized pad on the silicon device that, when combined with solder, forms a strong bond between the silicon and steel.

The bond pad stack, shown in Figure 3.11, consisted of layers of chrome (10 nm), nickel (200 nm), and gold (10 nm). Gold and solder form strong, reliable bonds that have been used in a wide range of applications. Gold, however, does not adhere well to the native silicon dioxide found on silicon wafers. In order to form strong bond pads, adhesion layers must be used to bond the gold to the silicon wafer. Chromium has a strong affinity for silicon dioxide and adheres well to the surface even during bonding at elevated temperatures. Nickel provides a diffusion barrier that prevents diffusion of the gold into the chromium or silicon wafer during the bonding process. While the nickel layer is not always necessary, it can improve bond reliability as well as preserve the electrical properties of the silicon wafer, if that were necessary. For information on how this bonding process was conducted see the full paper [24].

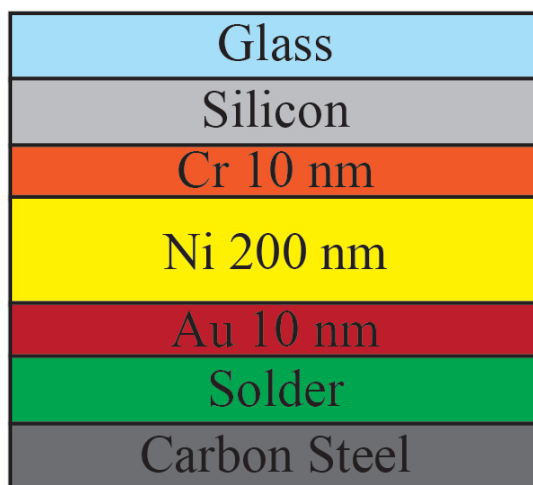


Figure 3.11: Layers of the eutectic bond used to attach the steel adapter to the silicon wafer.

In order to understand the viability of this sealing method, testing was conducted in three phases. In the first phase of testing, the adapter was bonded directly to a solid silicon wafer with no inlet holes. The valve on the steel adapter was then connected, through a regulator, to a pressurized air source. This setup was the ideal method to test the reliability of seals up to pressures of 100 psi. In order to simplify the leak detection process, the bonded assembly was submerged in water. The pressure was slowly increased and, if there was a leak, bubbles would form in the water. Testing in this manner revealed that the bond could withstand 100 psi, the maximum available pressure from the compressed air source. Bonding directly to a silicon wafer was the simplest test case, eliminating many of the complicating factors found in the next two test cases.

In the second phase of testing, a pressure leak test was conducted by attaching miniature valves to both bonding pads of the test device. One valve was sealed and the other was attached to a compressed air line. The entire device was then submerged in water and the pressure was steadily increased from 0 to 100 psi. This phase of testing was used as a quality control exercise to identify if the bonding process had failed. In this test leaks were again detected if air bubbles were observed.

Once a bond had passed the pressure leak test, a more rigorous study was conducted. The microvalves on the test devices were connected to a filling apparatus and water vapor was added to the system, see Figure 3.12. Once the devices were filled with approximately half vapor and half water the micro valves were closed and the device was detached from the filling apparatus. After closing the valves, plug screws were added to the end of the microvalves to reduce liquid losses. The device was then allowed to cool to room temperature. Finally, in order to remove any excess water that had spilled on the outside of the device during filling, the device was held at room temperature for 24 hours.

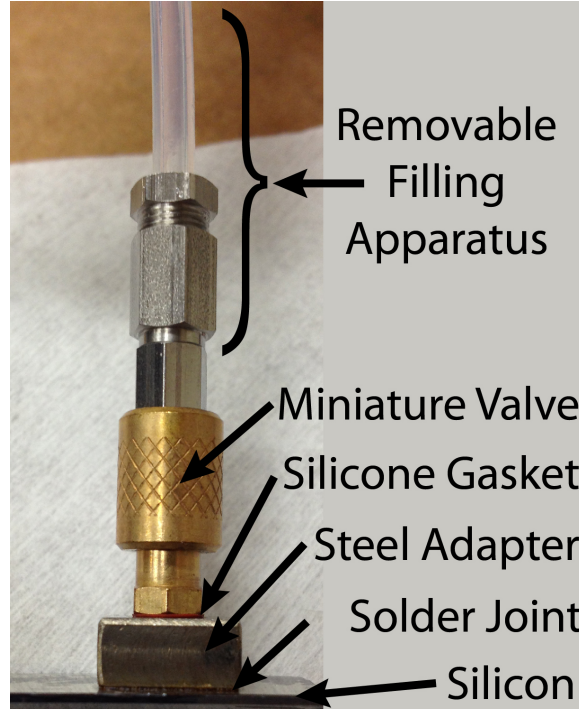


Figure 3.12: The components of the new sealing method explained.

In order to quantify the leak rate, filled devices were heated on a hot plate and the rate of water loss was quantified by tracking device weight. Empty, full, and periodic weights during the heating process were taken on a Denver Instruments PI-214 scale with 0.1 mg accuracy. By measuring the loss in weight as a function of time, which was due entirely to escaping water, it was possible to quantify the leak rate of the device. Multiple tests were conducted at two temperatures, 100C and 110C. Weight measurements were taken periodically over a 6 hour period at 100C and over a 3 hour period at 110C. From this data the percent weight loss could be calculated using the following equation:

$$WL = \frac{D_{t-1} - D_t}{W_{H_2O}} * 100 \quad (3.3)$$

where WL is the percent weight loss, D_t is the device weight at time point t , D_{t-1} is the device weight at the previous time point, and W_{H_2O} is the original weight of the water in the device. The weight loss per hour can be calculated simply by dividing WL by the duration of time that elapsed between the two measurements. In addition the dimensionless normalized fluid weight, NW, could be calculated using equation:

$$NW = \frac{D_t - D_0}{W_{H_2O}} \quad (3.4)$$

where D_0 is the weight of the empty device. Initially at time zero the device was filled with fluid, so the normalized weight was 1. As liquid was lost during testing, NW decreased. The normalized fluid weight is plotted versus time in Fig. 3.13 and 3.14.

The results of the tests revealed some leaking at both 100°C and 110°C. The trend in these graphs shows a slow leak which, over the course of 6 hours at 100°C, allows less than 5% of the water mass to escape. The slope of these lines reveals the percent weight loss per time unit. Two metrics are useful when evaluating this data, the average hourly percent weight loss (AHPL) and the cumulative liquid loss approximation (CLLA). The AHPL averages the loss rate between each of the data points throughout the duration of the test. This average can be influenced if there are large fluctuations between data points. The CLLA, on the other hand, is calculated by dividing the total fluid lost by the total test duration. The CLLA smooths the data and is more useful than AHPL for projecting device behavior hours into the future.

While the various runs show similar trends, there are a few interesting deviations. In the 100°C experiment, Tests A1 and A3 have the largest losses. In Test A3 there was a large weight loss in the first half hour, however after this initial loss the normalized weight lost per hour tracks Tests A2 and A4. This sharp initial loss is likely the result of some un-evaporated water on the external surface of the device rather than a large leak in the gas tight seal. If the initial half hour loss is disregarded as an outlier, the standard deviation of Test A3 reduces to 0.155% with an AHPL of 0.441%. This is an example of where the CLLA calculation is more representative of overall behavior. In test A3 the initial loss has a substantially lower effect on the CLLA. Test A1, on the other hand, has a much higher AHPL and CLLA throughout the duration of the test. This indicated an inferior seal when compared to the other devices.

The testing conducted at 110°C revealed higher losses than the 100°C degree tests. Intuitively this result should be expected, higher temperatures result in higher internal device pressures, which in turn expel vapor more quickly. While the losses are higher, the overall losses are still relatively low - ranging between 5% and 10% of the overall water weight. Like Test A3, Test B2 had significant losses within the first half hour that did not continue as testing continued. Again this is likely explained by the presence of water droplets on the outside surface of the device. While all of the chips were left overnight to dry, some chips may have had more liquid on the surface than others. If the initial half hour of Test B2 is disregarded as an outlier, the AHPL drops to 3.21% with a standard deviation of 1.55%. This is still a high standard deviation, but the range of the averages amongst the test is more consistent. In future testing, a low temperature bake for 30 minutes may help to reduce this source of error.

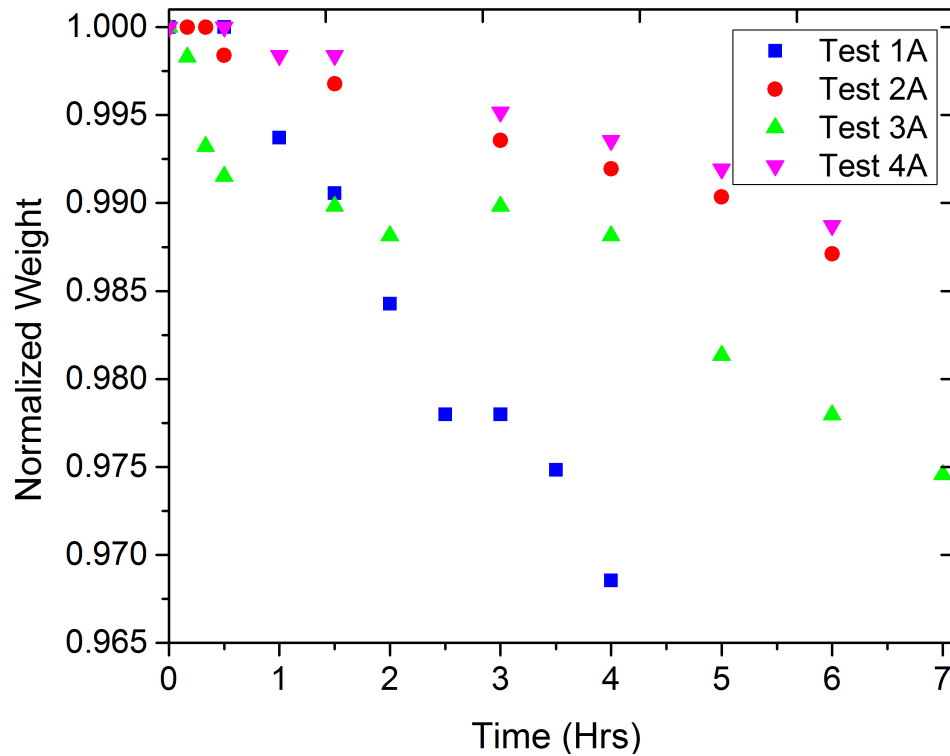


Figure 3.13: The normalized fluid weight loss for testing at 100° Celsius for 4 separate tests. The normalized weight is the percentage of fluid left in the device. It begins full with a normalized weight on 1 and as fluid escapes the percent drops and empty device would have a weight percent of zero.

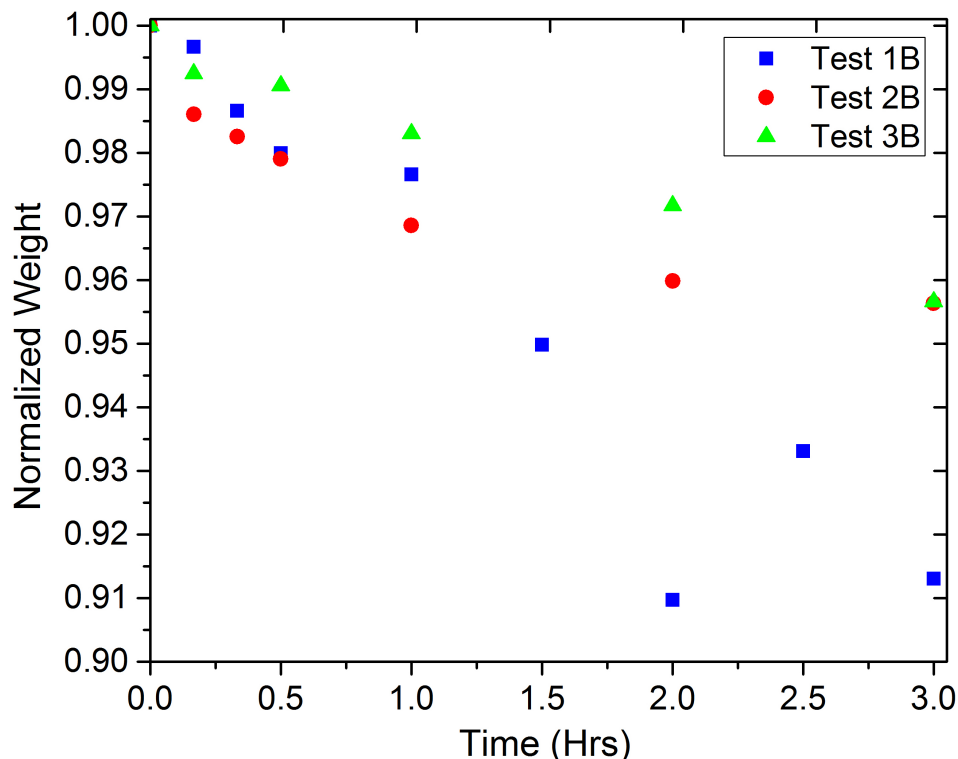


Figure 3.14: The normalized weight loss per hour for testing at 110° Celsius for 4 separate tests. For a definition of normalized weight see the caption on figure 8.

An important note to make is no non-condensable gasses were observed to enter the system during testing. The μ LHP will fail if gasses can enter the system and cause blockages. While a slow outward leak is not ideal, losing small amounts of working fluid should not negatively impact the behavior of the μ LHP devices. Overall this sealing research lead to a method that will work for testing of the device, but is only necessary in the future all silicon μ LHP. In the end for this research a hybrid device was created that used the key silicon components, but the fluid channels were made of aluminum. For this case the valves could be directly attached to the filling ports and the silicon components were attached to the aluminum with a new epoxy that was found after this sealing work was completed. The bonding technique presented here will be necessary in the next level device and could be used to find a way new way to attach the silicon to the aluminum if the epoxy doesn't work in future tests.

3.4.1 Laser Etching Silicon

One of the plans for the future creation of an all silicon μ LHP is to possible laser ablate the silicon features to increase the speed of fabrication process and allow for more device iterations. With that in, mind resolution and accuracy testing on silicon using a Universal

PLS laser etching was conducted. One of the biggest challenges with this depth testing was figuring out the resolution and the possible accuracy for the device. In order to test silicon wafers a kinematic clamp was designed and fabricated to improve the accuracy, precision and resolution. The laser ablation system was used to etch a standard 6 inch silicon wafer. In the end a resolution of 33 microns with a standard deviation of 1.7 microns was measured. In addition, with the clamp the laser etcher can provide accuracy of approximately 44 microns in X direction and 51 microns in Y direction. Also, it results in an improvement in the precision with a final standard deviation of 32 and 61 microns in the X and Y direction respectively.

For this work the laser ablation tool uses a high power class 4 green light fiber laser to break chemical bonds and remove material. To aid in the material removal, nitrogen or compressed air is used to blow the broken particles away from the material being etched. During the initial calibration of the tool, testing to determine the necessary power and speed settings was preformed. Issues with testing revealed that the wafer would move during etching making accurate calibration difficult. In order to eliminate this problem a customization of the tool has been proposed that will include the design and development of a wafer holding mechanism that will stop this movement during etching. In addition this mechanism is expected to have some type of accurate and repeatable placement to allow multiple wafers to be identically etched. Ideally these etches would be repeatable even if the wafer is removed and replaced during the process. To accomplish this goal the clamp, as seen in figure 3.15, consists of two separate pieces. This allows the bottom piece to remain stationary, while the top piece can be moved in and out to allow the wafer to be easily placed in the clamp and changed out. The top piece is connected to the bottom piece through a kinematic clamp. To avoid laser damage to the clamp, a cutting area shown in the center of figure 3.15 was removed from both pieces. This gap between the wafer and the table allows for the laser to become diffuse, preventing it from damaging either the clamp, or the table. The final clamp material was selected as aluminum, to keep the overall weight down and allow for rapid machining. Figure 3.16 shows the clamp with a wafer in it.

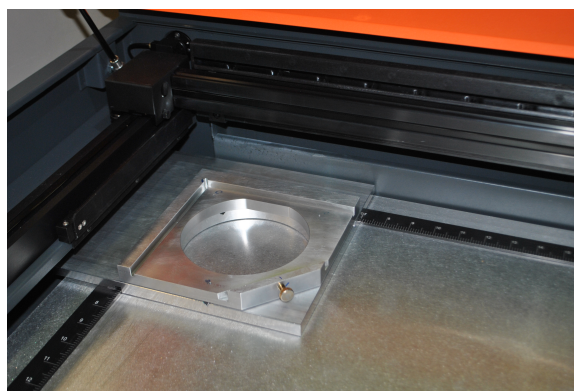


Figure 3.15: Aluminum kinetic clamp in laser bed.

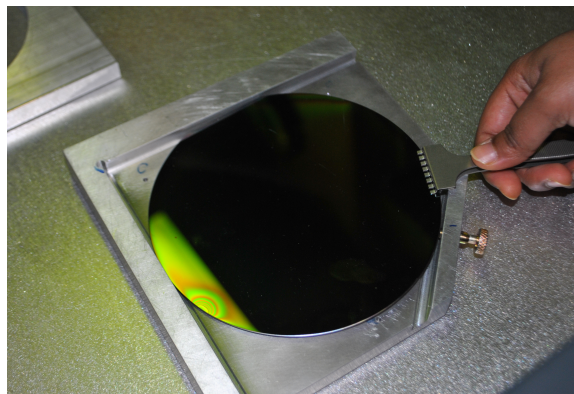


Figure 3.16: Clamp with a wafer being positioned.

Once the clamp was designed and manufactured the resolution, precision and accuracy were qualitatively characterized. In order to do this, there must first be a clear definition of resolution, precision and accuracy for this tool. The aim is to characterize these parameters under two conditions: with the clamp and without the clamp. These definitions are as follows, resolution is the smallest discernible feature spacing, or how close two features can be before they become one feature. Precision is the repeatability, or the ability to remove and replace the wafer and etch a new feature in the desired location over and over again. Finally, accuracy is being on location, or having an etch feature where it is supposed to be according to the design. To test these, a pattern was etched into one wafer. The decision was made to etch only one wafer, so as to remove any material errors. In order to do this it was assumed that the starting wafer was ideal. The etch pattern had two elements one for determining resolution and the other for accuracy and precision.

In order to determine resolution a pattern with increasing feature spacing was etched into the wafer. The laser beam itself can etch roughly 50 micron lines, but there was no data indicating how close two lines can be. The schematic below, see Figure 3.17, shows several lines with increasing spacing; the first two lines are spaced 10 microns apart, and the second set are 20 microns apart increasing in 10 micron increments. The resolution of the device will be the spacing between the first two separate lines that are completely separate from one another. For added accuracy this spacing will be measured. The final resolution is presented in the metrology and results section.

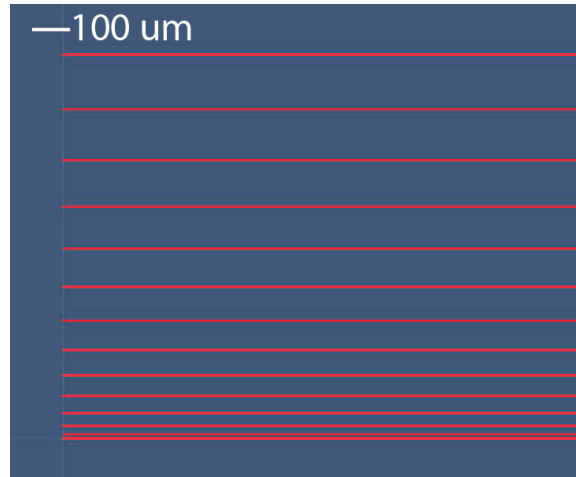


Figure 3.17: Etch pattern used to measure the resolution of the laser.

The testing method for the accuracy and precision measurements required features to be etched in both the X and Y direction. In order to measure the accuracy only one measurement was required, but in order to determine the precision several etches were required. Making several etches also has the added benefit of generating many data points and more accurate results. The etch pattern consisted of one set of reference lines and then 6 sets of 1 cm long lines that were spaced 100 microns apart. Each line was both in the X and Y direction, shown in Figure 3.18. The color variation of the lines represents a time that the wafer was removed from the tool and then repositioned back in the tool and etched again.

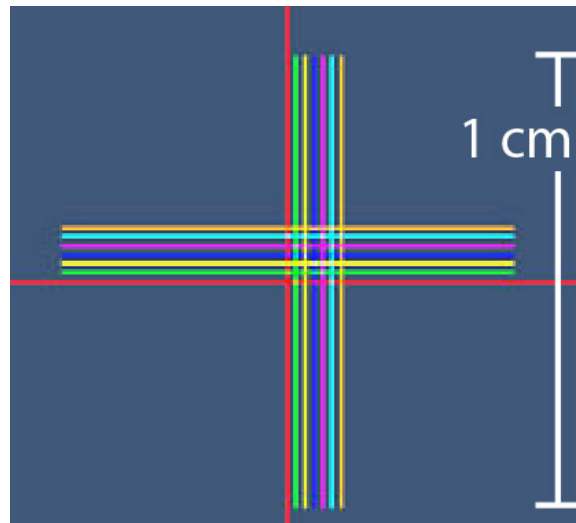


Figure 3.18: Etch marks used to determine accuracy and precision of the laser and the clamp

Three tests were performed using this etch pattern, one without the clamp and two with the clamp. In order to test the improvement etches were first made with just the wafer positioned against the sides of the tool, shown in Figure 3.19. For the second test the top of the clamp was removed, then the wafer was removed and repositioned and finally the clamp was reattached to the base. In the third test the wafer was positioned once and then the top of the clamp was removed from the base after each etch. The second test measured the system inaccuracies as a whole, while the third test looked only at the accuracy of the kinematic coupling. One flaw in these tests, were the assumptions associated with the reference line.

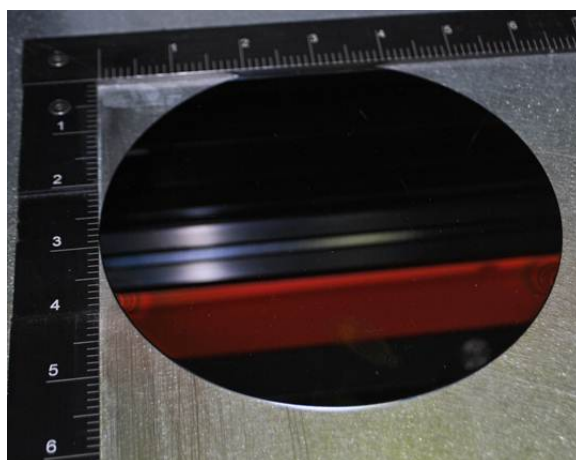


Figure 3.19: The wafer with no clamp just set into the laser.

Due to the fact that the wafer is blank and there is no easy method to measure from the edge of the wafer to widely spaced etch marks, reference lines were made. These lines were etched first along both the X and Y direction and made as close to the center as possible. The reference lines however are not perfect and are subject to error, but because they were etched at the same time, it was assumed that any error in the reference lines would be equal for all of the reference lines. This is a flawed, but necessary assumption in order to make a good approximation of the accuracy and precision. In order to support this approximation two control tests were performed. In these test the entire etch pattern was created in one etch with no removal of the wafer; any error in these patterns is mostly from the laser etcher. After all of these patterns were etched into the wafer they were measured using a high precision microscope.

Keyence VHX-500F Digital Microscope was used to measure the etch patterns. These measurements were later used in calculating the accuracy, precision and resolution of the laser etcher with the kinematic clamp. Keyence VHX-500F digital microscope is a high resolution CCD camera based system with a high intensity halogen lamp and image processing capabilities that integrate observation, recording, and measurement functions.

As mentioned earlier, etch patterns have two elements: 1 A set of equally spaced horizontal and vertical lines at a certain distance from the reference line. It is used to determine the accuracy and precision of the clamp. 2 A set of horizontal lines with varied spacing among themselves. This pattern will aid in determining the resolution of the clamp.

First, the etch patterns were made on 6 inch silicon wafers using the kinematic clamp on the laser etching tool, followed by the measurement of the etched features using the microscope. The spacing was measured between the etched vertical lines and the reference line along the X-direction (see Figure 3.20a). Using these measurements the accuracy of the features can be determined with respect to their desired position. Furthermore, a set of these measurements were made in order to generate various data points. This data was later used in calculating the precision and accuracy along the X-direction. A series of data points was also recorded by measuring the spacing between the etched horizontal lines and reference line along the Y-direction (see Figure 3.20b). This information was used in calculating the precision and accuracy along the Y-direction.

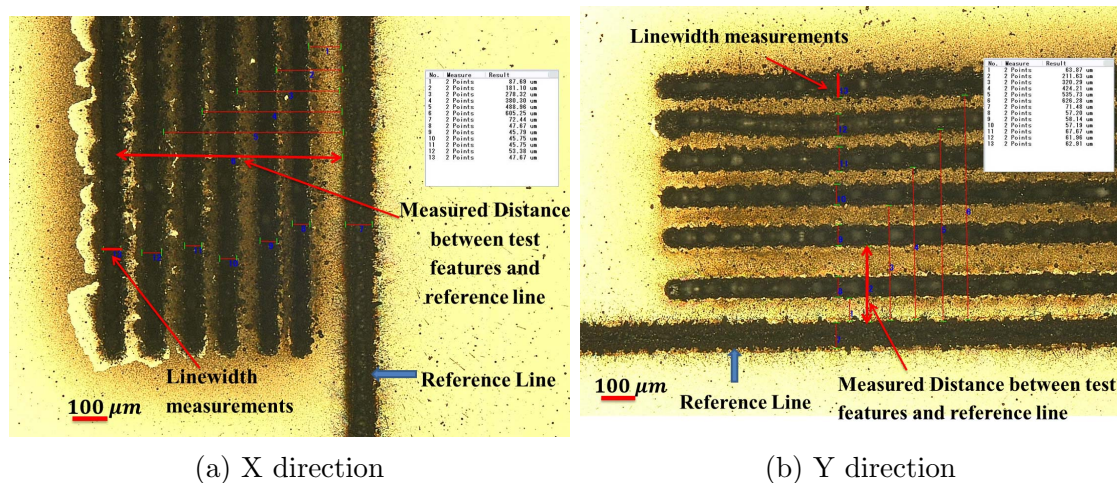


Figure 3.20: Accuracy Images for the Clamp Case

A picture of the etch pattern with both horizontal and vertical lines is shown below (Figure 3.21). This etch pattern was generated with the help of laser etcher tool using the wafer clamp. The etch patterns were also made on 6 inch Silicon wafers without using the kinematic clamp followed by the measurement of the etched features. Accuracy and precision were calculated along both the X-direction (Figure 3.22a) and Y-direction (Figure 3.22b) in a similar manner as the kinematic clamp case.

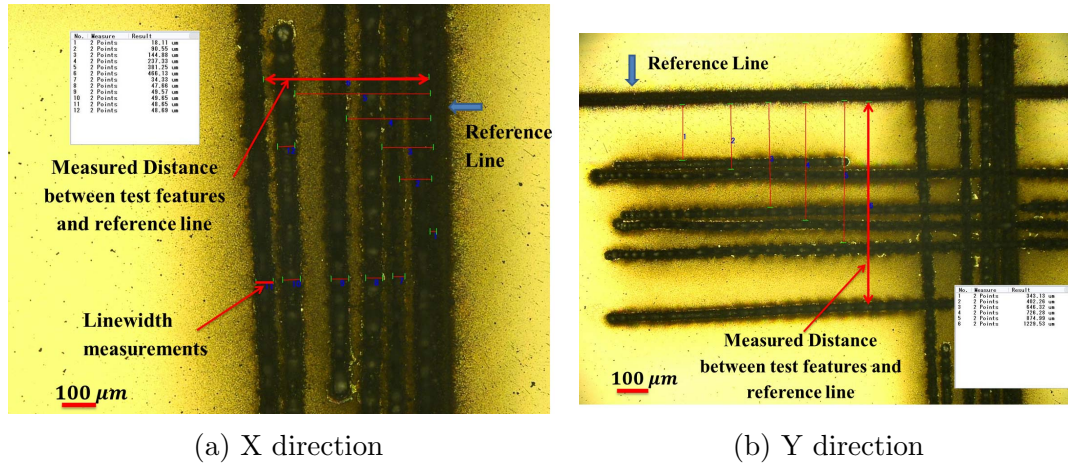


Figure 3.22: Accuracy Images for the No Clamp Case

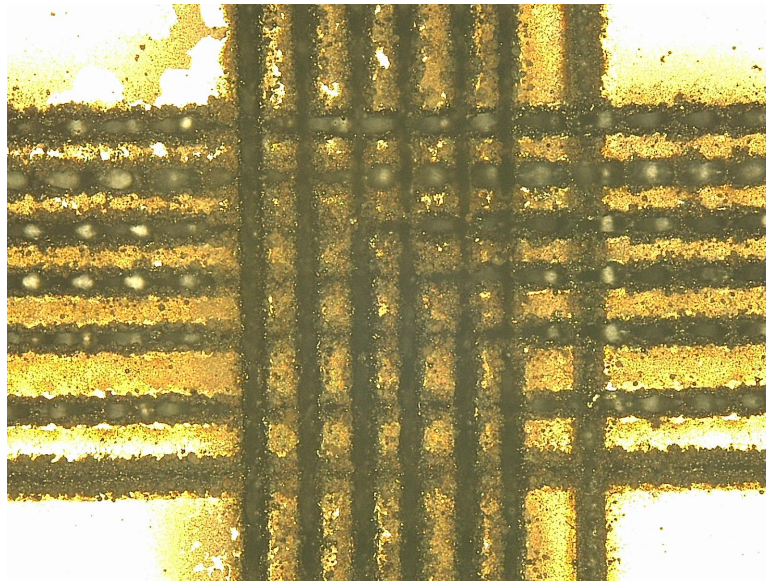


Figure 3.21: The normalized

Measurements were also made for calculating the resolution. First, a pattern with increasing feature spacing was etched into the wafer using the kinematic clamp during etching. Next, the spacing was measured between the various lines that are completely separate from one another i.e. no overlap (see Figure 3.23). This data was used to determine the resolution. Resolution was also determined for the case when no kinematic clamp was used during the etching of the pattern. Like the previous case, spacing between two distinct lines was measured for determining the resolution.

Various calculations were done to determine the accuracy, precision and resolution of laser

etching machine for both the clamp and no clamp case. The formulas used for computing these parameters are given below:

$$Accuracy = \frac{Measuredvalue}{Desiredvalue}$$

$$Errorpercent = \frac{MeasuredValue - DesiredValue}{DesiredValue} \times 100$$

$$Precision = STDEV = \frac{1}{N} \sum_{i=1}^N (x_i - \mu)^2$$

Where N= number of data points

$$x_i = MeasuredValue - DesiredValue$$

$$\mu = Meanofx_ivalues$$

The calculated values are tabulated below for both the clamp and no clamp case. Table 3.2 shows the calculated average accuracy in microns. The calculated average percent error is listed in Table 3.2. Table 3.3 shows the calculated precision values.

Accuracy (microns)		
	With Clamp	Without Clamp
X direction	44	151
Y direction	51	257
Accuracy (% Error)		
	With Clamp	Without Clamp
X direction	19	72
Y direction	21	109

Table 3.2: Accuracy data and percent error for the wafer with and without the clamp.

Precision (microns)			
	With Clamp		
	Wafer Movement	No Wafer Movement	
X direction	31	25	113
Y direction	61	44	233

Table 3.3: Precision data for the case where the clamp is moved from its base and the wafer is removed from the clamp and the case there the clamp is moved from the base, but the wafer is not moved.

The results include the average etch mark width, the resolution of the tool, as well as the accuracy and precision of the tool with and without the clamp. The average etch mark

width was found to be 51 microns with a standard deviation of 12 microns. It was also determined that the etch width did not change with or without the clamp. The resolution was 33 microns on average with a standard deviation of 1.7 microns, shown in Figure 3.23. Resolution, like etch width was not impacted by the use of the clamp. Although, the clamp did not affect the etch width or the resolution, it did improve the accuracy and precision.

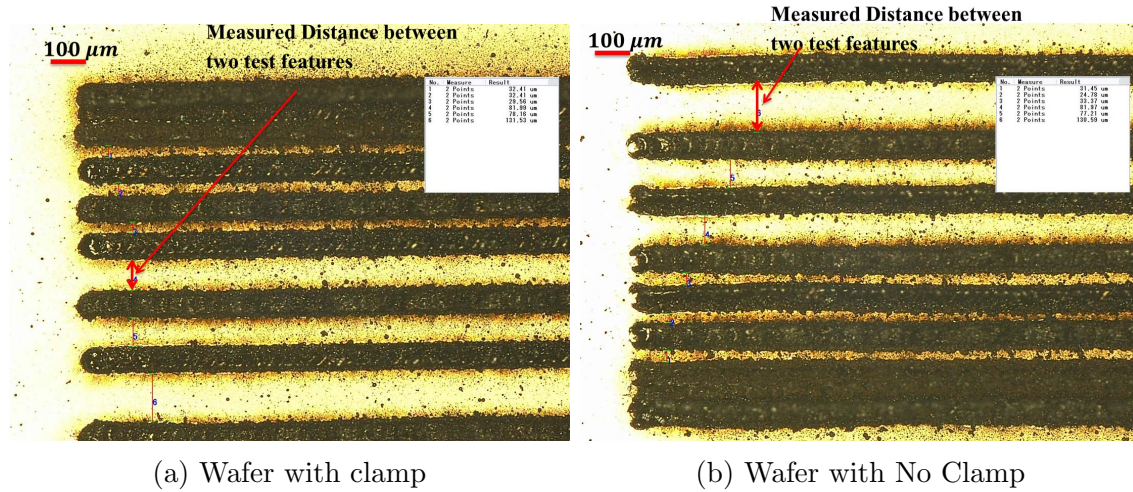


Figure 3.23: Pictures of animals

When using the clamp the laser etcher can provide accuracy of approximately 44 m in X direction and 51 m in Y direction. Using a kinematic clamp changed the accuracy and the precision. The accuracy improvement in the X and Y direction can be seen in Figure 3.24.

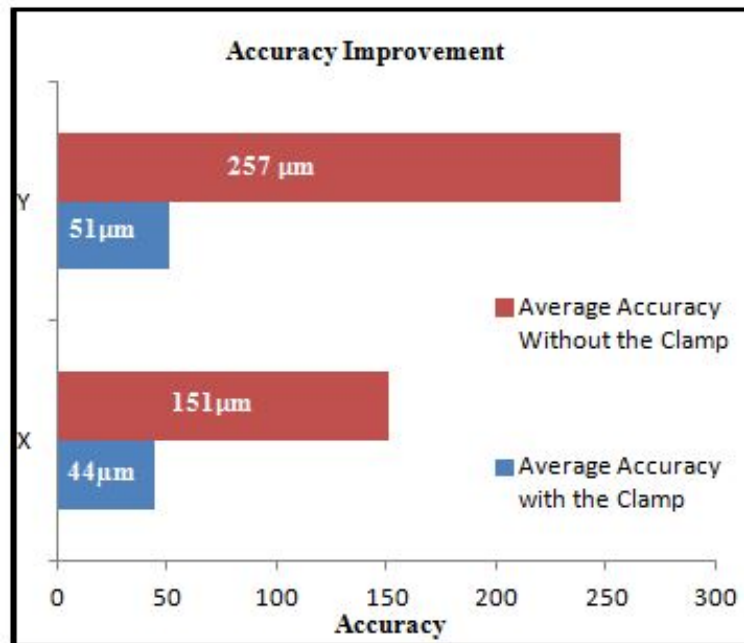


Figure 3.24: A bar graph to show improvement in the accuracy for both the X and Y directions.

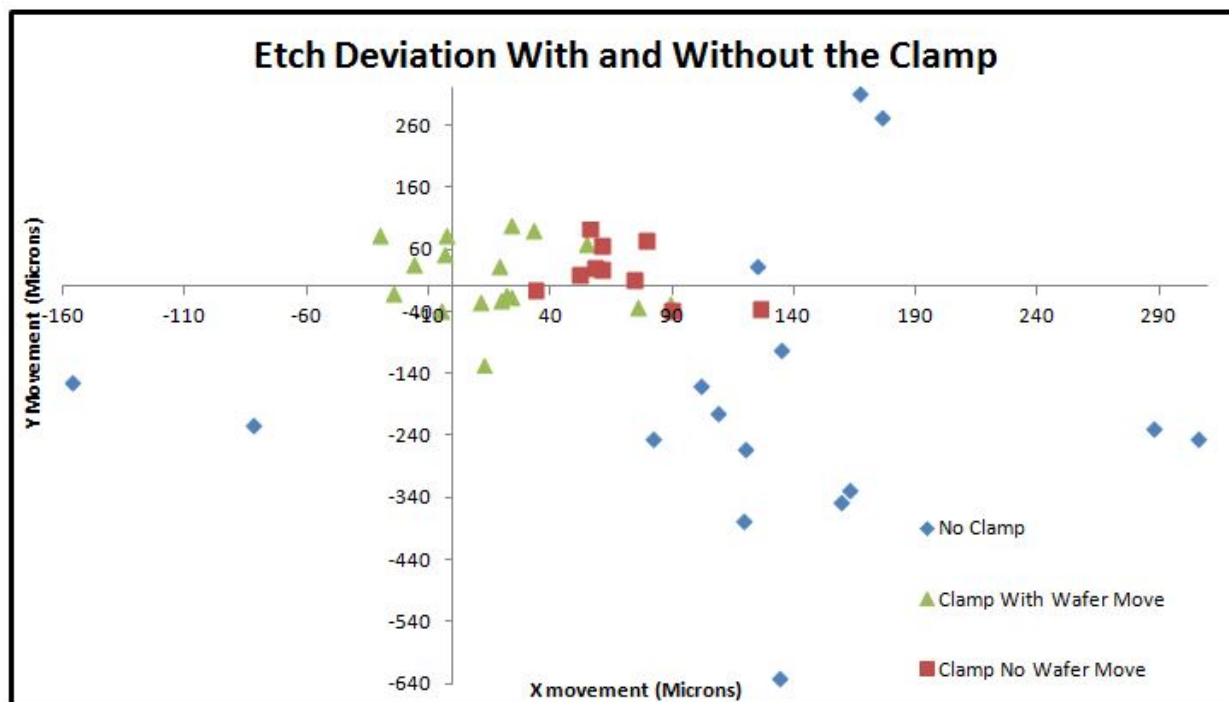


Figure 3.25: A graph showing the results of the accuracy and precision in the X and Y direction.

The results of the precision calculations are shown in Figure 3.25. All of these results show a favorable improvement in the accuracy and precision with the addition of the wafer clamp. The accuracy results show a simple conclusion, but precision results show some interesting trends. From Figure 3.25, the spread of the no clamp data is broad, showing that it is neither precise nor accurate. The data for the clamp where the wafer was moved is clustered around the center point. This shows that the clamp is somewhat precise because the spread is not large and somewhat accurate because the spread is focused around the origin. The final data point for the clamp without moving the wafer show clustered points, but they are not around the origin. This data shows that the clamp without any wafer movement is more precise, but less accurate. Most likely, this occurred due to a misalignment of the wafer in the clamp, and because it was never repositioned this error stayed constant for all of the measurements. In conclusion there are still some inaccuracies associated with the clamp, but its a great improvement.

With this data a recommendation can be made to the minimum feature size to be expected and the minimum distance between features necessary for repeatable results. The smallest etch lines that should be expected are 63 microns. In addition, the minimum line spacing that one should use it 65 microns in the X direction and 95 microns in the Y direction. These gaps account for the standard deviation found when using the clamp and removing the wafer. During normal use of the clamp, the wafer will only be secured once.

Most movement that will occur will be from the entire top piece of the clamp. If this is the intended use, it is recommended to make sure that the first placement of the wafer is as accurate as possible. These are the minimum suggested distances anything larger than these should be fairly accurate and precise. Overall it seems that the resolution of this laser system is probably too low to create the kind of features necessary for the silicon device. A new laser etching tool has recently been purchased that uses the same type of laser, but with mirror instead of stepper motor control a test like the one above should in the future be re-done for this new tool to see if it can be used for the silicon fabrication.

Chapter 4

Full System Modeling, Design and Integration

4.1 Hybrid Loop Heat Pipe Design

Up until this point in the research, has been focused on creating, fabricating and testing a series of individual components. Once that work was completed a system was created to incorporate them those components together. The system was design to have two key silicon components, the CPS wick and the interline optimized evaporator, incorporated into a larger system. As previously mentioned in section 3.3, the CPS wick and the evaporator were found to work together in an open loop test set-up, but this system is the first attempt to have the evaporated fluid condensed and re-circulated.

A full loop heat pipe system is in many ways symbiotic. All of the components must balance and work together. In order to create a loop like this system dependencies and dimensions needed to be determined. Everything from the pressure balance, to the thermodynamic balance was reviewed in order to make a cohesive system. In addition control volume calculations help determine the dependencies of different components. To help determine future system possibilities there was also extensive modeling to calculate pressure loss in these systems when certain thickness, length and width parameters were varied. This was necessary because no fluid calculation could be done with the odd geometry of the system.

Other key theoretical system level models that were reviewed were on start up conditions and evaporator arrangements. The start-up condition information give us a method for interpreting future data and the evaporator arrangements show how other loop heat pipes work and give possible alternate arrangements that the μ LHP could use.

4.2 Loop Heat Pipe Fluid Flow and Heat Transport Models

Loop heat pipe systems have specific requirements for functionality. All of the components must work together and follow specific thermodynamic, fluid flow and heat transport rules in order for the system to function. The basic principles of this operation are similar to other phase change devices, but LHPs are unique. The loop is not just driven by the evaporation-condensation cycle with a wick to balance the pressure. Separating the liquid and vapor phase transport into different lines creates an incredibly stable system, but that system requires balance to function. In addition the wick is isolated solely to the evaporator section minimizing losses in the, but limiting the options for wick refill. For each perk of a loop heat pipe there is a design limitation that makes the systems more difficult to create, but the advantages outweigh this complexity by creating a system that can handle higher heat flux with less thermal resistance.

Overall the system follows a specific control volume model that balances enthalpy change, heat loss and pressure changes. The most important balance in any heat pipe or LHP is the balance between the capillary pressure gain in the wick and the pressure drop from frictional forces in the loop. This equation looks like the following:

$$(\Delta P_{cap})_{max} \geq \Delta P_v + \Delta P_l = \frac{2\sigma}{r_{me}} \quad (4.1)$$

where the capillary pressure ΔP_{max} is the capillary pressure created in the wick and the pressure drop in the loop is caused by losses in the vapor ΔP_v and the liquid ΔP_l lines. In a larger loop heat pipe gravity can also cause pressure losses, but this device is small and the bond number can be calculated to determine whether surface tension forces or gravitational forces dominate the system. Bond number is calculated using the following equation:

$$B_o = \frac{\Delta \rho g L^2}{\sigma} \quad (4.2)$$

Where, $\Delta \rho$ is the density difference between the two phases of the fluid, g is gravity, L is the characteristic length and σ is the surface tension of the liquid. If the bond number is less than 1 then surface tension forces are the primary forces determining the flow and if the bond number is much larger than 1 then gravitational forces are the primary forces. When the number is in the middle of this range both forces are important.

The pressure balance is the critical to the function of the device. In addition to the balance between the channel losses and the wick gain there is thermodynamic temperature and pressure balance between the two sides of the wick. Figure 4.1 shows a P-T diagram of the loop heat pipe system. In this diagram points 1 and 8 show two saturation states that

are on different sides of the wick in order for these two point to be on the saturation line they have to be related by the following equation:

$$\left. \frac{dP}{dT} \right|_{T_8} (T_1 - T_8) = P_1 - P_8 \quad (4.3)$$

This equation states that for any temperature difference that exists between two point on the saturation line there is also a specific pressure difference determined by the Clausius-Clapeyron equation:

$$\left. \frac{dP}{dT} \right|_{T_8} = \frac{h_{lv}}{T_s(v_s - v_l)} \quad (4.4)$$

In this case dP/dT is the slope of the P-T line at the saturation temperature (T_8). Where h_{lv} is the latent heat of vaporization and v is the specific volume of vapor and liquid at temperature T_8 . This relationship is required for the loop heat pipe to start circulation, without this balance the system cannot expel any droplets in the evaporator and vapor lines.

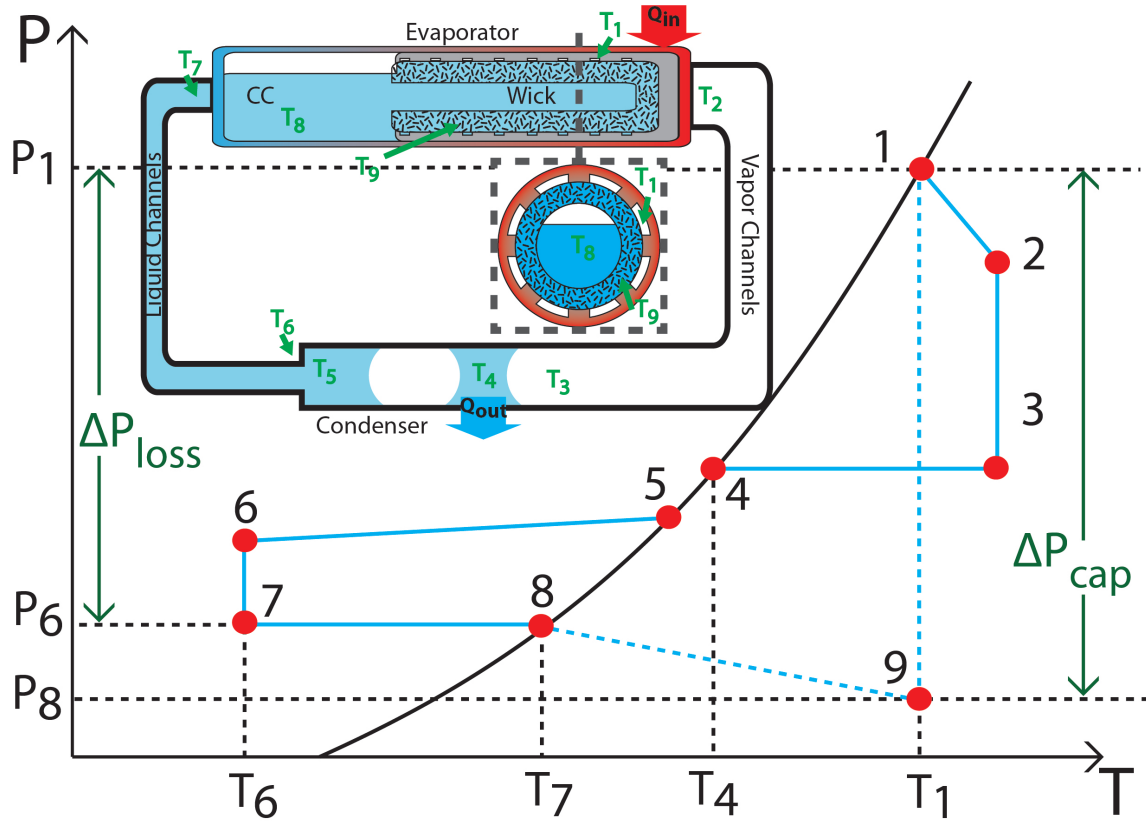


Figure 4.1: Pressure versus temperature graph showing where different regions of a LHP are in relation to there location in the loop.

With the balance of the system in mind a control volume analysis can help determine interacting components of the system. For this analysis the system is broken into three parts, shown in figure 4.2, CV1 is the wick structure where there is both liquid and vapor present. CV2 is the vapor region including the two phase section of the condenser before all of the vapor is converted to liquid. CV3 is the liquid region of the device including any part of the condenser where liquid is subcooled.

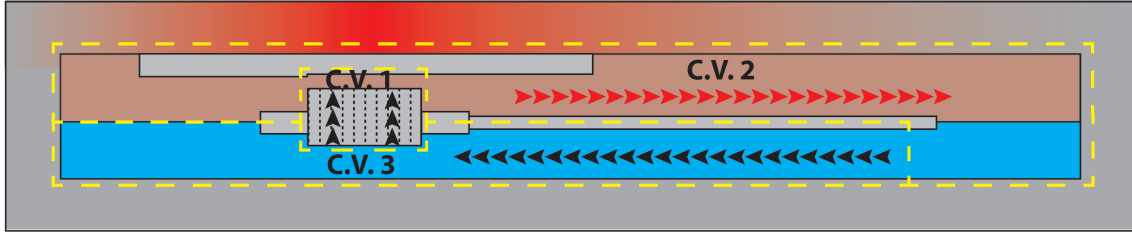


Figure 4.2: Sideview of the device with different control volumes labeled.

The wick section of the control volume CV1 can be described by the following thermodynamic balance:

$$m(h_v - h_l) = Q_{ap} - Q_{e-cc} \quad (4.5)$$

Where m is the mass flow rate of the device, which is equal to the vapor or fluid mass flow rate because the system is closed. h_v and h_l are the enthalpies of the vapor and the liquid. Physically this represents the enthalpy change in the system as the fluid moves from a liquid to a vapor state. Q_{ap} is the applied heat to the device and Q_{e-cc} is heat lost from the evaporator or wick into the compensation chamber. This is usually determined by the temperature difference across the wick and the approximated thermal conductivity of the wick given by this equation:

$$Q_{e-cc} = \frac{k_e A_w}{t_w} \Delta T_w \quad (4.6)$$

There are two key balance requirements in the wick that keep the loop functioning and the thermal and hydraulic lock concepts. A good wick needs to limit the amount of thermal leakage (Q_{e-cc}), known as thermal lock. This can be accomplished by making the wick out on an insulating material or coating the wick in an insulating material. The CPS wick has a layer of silicon and zinc oxide on it, which lowers the thermal conductivity versus a regular metal wick. In addition as the fluid changes from a liquid to a vapor there is a large pressure build up on the evaporator side of the wick. If the pores of the wick are large this pressure difference will cause vapor to push into the wick and the device will dry out. Keeping these pores small so that they can withstand the large pressure build up and keep the wick pumping is called hydraulic lock.

The second control volume (CV2) for the vapor channels of the device can be described by the following equation:

$$m(h_v - h_l) = Q_{lv-a} - Q_{c^{2\phi}} \quad (4.7)$$

Where once again there is a mass flow rate and an enthalpy change, but this time due to the start of condensation. Q_{lv-a} is the heat loss to ambient in the vapor channels this is usually assumed to be zero in modeling, but in an actual device this can be present. $Q_c^{2\phi}$ is the heat removed by the vapor to liquid phase change in the evaporator. The third control volume describing the liquid flow (CV3) is described by the following equation:

$$Q_c^{1\phi} + Q_{ll-a} = Q_{app} + Q_{cc-a} \quad (4.8)$$

Where $Q_c^{1\phi}$ is the heat lost in the portion of the condenser that is all liquid. This loss begins the process of sub-cooling in the liquid return lines. Sub-cooling the liquid that is returning to the compensation chamber prevents boiling from occurring in the liquid return lines. Boiling might occur if the pressure in the liquid lines drops low enough to cause the liquid to boil. The pressure drop is due to friction on the walls and will only cause boiling if the loss is large and the liquid temperature is already close to the boiling point. In addition this sub-cooling also keeps the liquid entering the compensation chamber cooler so that it does not boil when it comes into contact with the heat leaking through the wick. Q_{ll-a} is the heat lost by the liquid lines to ambient usually considered to be zero. Q_{cc-a} is the heat lost in the compensation chamber to ambient also usually considered to be negligible. These control volumes can then be broken down into pressure relations and all of these equations can be solved to determine the maximum possible heat applied based on the wick characteristics and the pressure losses.

4.3 Pressure Loss Modeling in LHP

As shown in the section above and figure 4.1 in the theoretical section, pressure losses and pressure balance is the key in any LHP system. The pressure loss in each part of the device must be calculated and evaluated to make sure that the cumulative loss is not larger than the pressure gain in the wick. The pressure loss can be evaluated in parts by breaking it down into the pressure loss in the vapor regions and the liquid regions. This is assuming that there is no pressure loss due to gravity. The total pressure loss from the device comes from the liquid line, the vapor line and the losses in the wick shown in this equation:

$$\Delta P_{total} = \Delta P_{vapor,vl} + \Delta P_{liquid,ll} + \Delta P_{vapor,c} + \Delta P_{liquid,w} \quad (4.9)$$

Each of the components of this equation are explained below. Starting with the liquid there is pressure loss in the liquid return line and the wick. The liquid line can be calculated using the Darcy-Weisbach frictional pressure loss equation:

$$\Delta P_{liquid,11} = 32 \left(\frac{L_{ll}}{D_{ll}^2 A_{ll}} \right) \frac{\mu_l}{\rho_l h_{lv}} Q_{ap} \quad (4.10)$$

Where L_{ll} is the length of the liquid section, μ_l is the viscosity, ρ_l is the density of the liquid, h_{lv} is the latent heat of vaporization, Q_{ap} is the applies heat flux and D_{ll} is the hydraulic diameter calculated using the following equation:

$$D_{ll} = \frac{4A}{P} \quad (4.11)$$

Where A is the cross sectional area and P is the wetted perimeter. This equation can be used for the hydraulic diameter of any section as long as the correct area and perimeter are used. That same type of pressure loss equation with some modification can also be used for losses in the vapor removal region or condenser line, if they exist. This equation has many limits and is only valid if the flow is laminar and the channels are not thin ($t > 1$ mm) [25]. In addition, this design the vapor does not travel in closed lines, but opens into a large chamber, and the condenser is also a large area so there should be minimal losses in these regions ($\Delta P_{vapor,c} = 0$). There is also little to no loss in the compensation chamber.

The next region of loss is the wick, which can be calculated using Darcy's law, by the following equation:

$$\Delta P_{liquid,w} = \frac{t_w}{A_w} \left(\frac{\mu_l}{\rho_l h_{lv}} \right) \left(\frac{1}{k_{ws}} \right) Q_{ap} \quad (4.12)$$

Where k_{ws} is the permeability of the wick usually provided or calculated as a dependence of the wick's ability to transport liquid when a pressure gradient is applied. The final pressure loss comes from the vapor transport line and this loss is geometrically dependant.

The important thing to know about the pressure loss equations used above is that they come from phenomological research rather than strict derivation. This means that observation, and in some cases modeling, is used to discover those equations or factors in those equations, so there are not solutions for every geometric factor. This becomes a more significant problem when there are thin small flows in strange geometries. To combat this issue modeling on the vapor pressure loss was conducted. This modeling looked at changing the width and thickness of the vapor channels in the μ LHP. By thinning the width and thickness a frictional loss factor can be determined for the geometry that will then allow parameters to be changes quickly without complex modeling. The general form for this type of pressure loss is

$$\Delta P = f \left(\frac{L}{D_h^4} \right) \left(\frac{\mu_v}{\rho_v h_{lv}} \right) Q_{ap} \quad (4.13)$$

Where f is the factor that needs to be determined. The modeling conducted was meant to be specific to this geometry, with width and depth changes, but isolated from changes in length. The goal of the model was to get to a simple correlations that allowed for changes in material, applied power, and geometry within the specified range.

The modeling was conducted in COMSOL 5.1 using the creeping flow physics because the Reynolds numbers of the flow were hand calculated to be below 1. The model was set up with no slip conditions at the walls and the geometry can be seen in figure 4.3. The inlet conditions were designated as the entrance coming from the small region shown in figure 4.3 and the boundary conditions were a pressure condition. The pressure was set to 30 kPa, 20kPa and 10kPa with the flow set to be in the normal direction with no back flow. In each of these cases the set pressure did not affect the total pressure loss across the device. This condition was chosen to match with the pressure set by the wick in the evaporator region. The outflow was set to be across the entire bottom of the end region. This physically represents the condenser where the vapor would stop moving. The outflow was set as a laminar outflow with a specified flow rate. The flow rate was calculated using the dependant relationship between mass flow rate and input power based on the following two equations:

$$\dot{Q} = \dot{m}h_{lv} \quad (4.14)$$

$$\dot{m} = \rho V A = \rho \dot{V} \quad (4.15)$$

These equations can be simplified and solved for $\dot{V}$ and this is the input into model. This was altered for 3 different values of $\dot{Q}$ (5W, 6W and 7W) and was found to linearly effect the pressure loss. The mesh used was a tetrahedral physics based mesh, this was run on course and normal with no variation in the pressure loss demonstrating convergence in the model. After the model was verified to not change unexpectedly with pressure input, volumetric flow rate output or mesh the model was considered sufficiently verified and the geometric changes could be made. The geometric model altered the width and thickness of the device from 2.54cm to 0.159cm and 0.5mm to 0.1mm, respectively.

Creeping flow ($Re < 1$), Stationary ($dt = 0$)

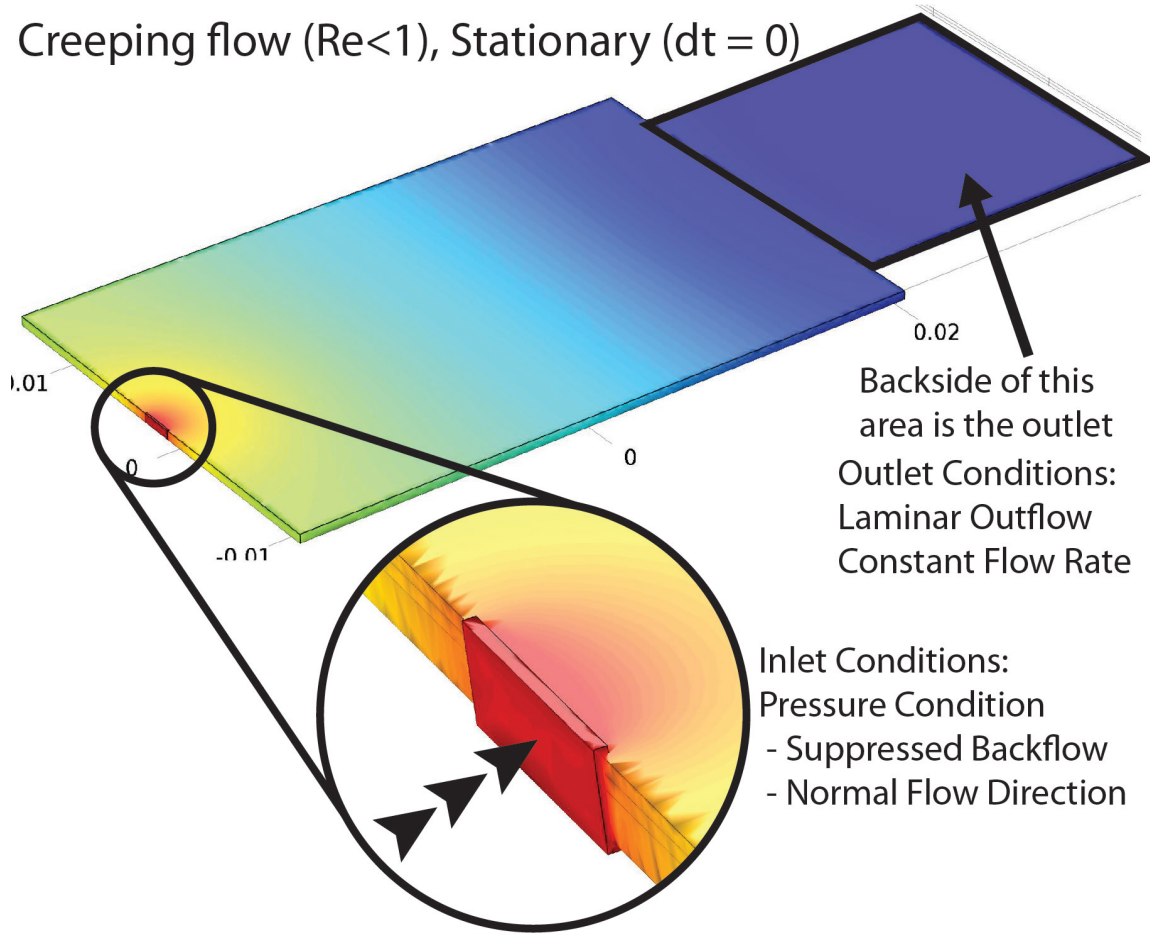


Figure 4.3: Comsol model overview with inlet and outlet conditions explained.

The results for the modeling show that the data has a non linear relationship with both width and thickness. This makes finding a simplified friction factor fairly complex. The first step to finding this relationship is to non-dimensionalize the width and thickness terms. This is important for the equation to match in terms of units and also keeps consistent with the non-dimensional nature of the friction factor. In many cases the frictional factor is determined in terms of the Reynolds number, the version of the equation shown above can be simplified to be in terms of Reynolds number. The non-dimensional thickness and width are given by the following equations:

$$t^* = \frac{t}{L} \quad w^* = \frac{w}{L} \quad (4.16)$$

Another simplification is to isolate the known terms in the equation by setting them to one symbol. In this case this is done by the following equation:

$$\zeta = \left(\frac{L}{D_h^4} \right) \left(\frac{\mu_v}{\rho_v h_{lv}} \right) Q_a p \quad (4.17)$$

This allows the friction factor to be isolated as shown in the following equation:

$$f = \frac{\Delta P}{\zeta} \quad (4.18)$$

From here a constant power relationship for the friction factor was determined from the data. The friction factor was determined to be:

$$f = \frac{14w^{*(-0.675)}}{t^*} \quad (4.19)$$

Figure 4.4 and 4.5 show the results of the modeling (shown as dots), a line fit on those models (solid line) and then the results using the theoretical friction factor (dotted line). The percent error on the theoretical model ends up being 10%.

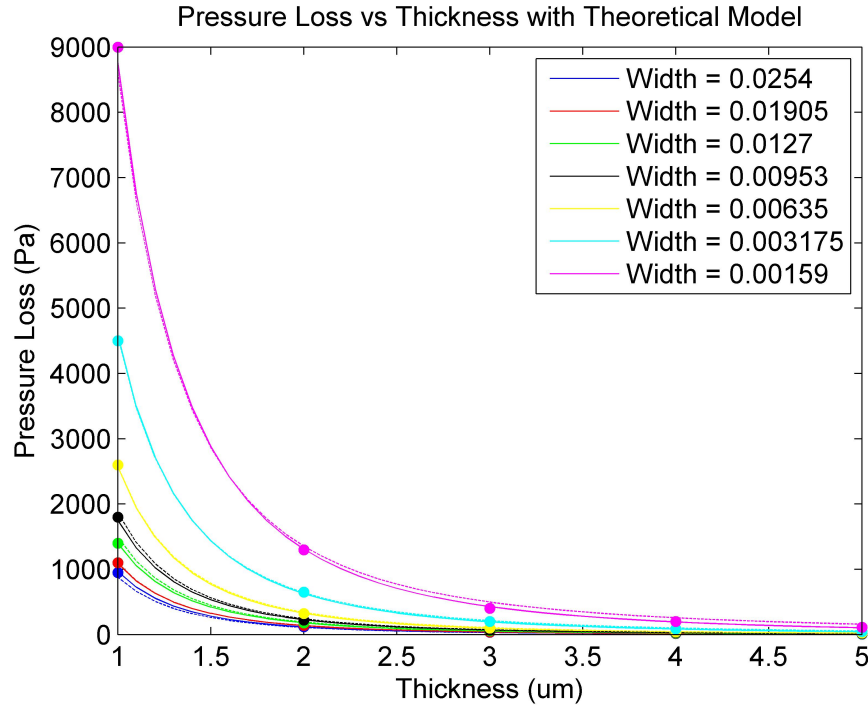


Figure 4.4: Pressure loss graph with experimental data as solid dots, a fit line as a solid line and the theoretical model marked with a dotted line.

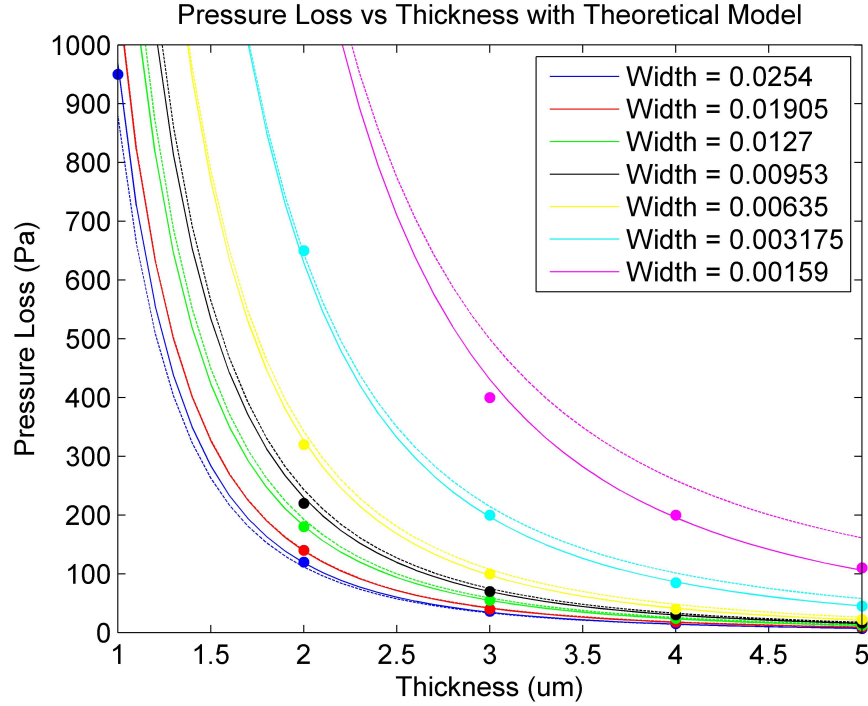


Figure 4.5: A close up view of figure 4.4 where the solid dots are the modeled data, the solid line is a fit line and the dotted line is the theoretical model.

From the theoretical model shown above further calculations can be done for specific scenarios, but the close fit of the equations shows some important trends in the pressure loss behavior of the device. First the relationship between power increase and pressure loss in linear the pressure loss increases as more power is applied, which is to be expected. Another interesting point is that from this equation a comparison between pressure loss with water vapor (the material modeled in COMSOL) versus other materials can be approximated. Table 4.1 shows this comparison by calculating a frictional pressure loss constant that is comprised of the fluid properties that do not change as long as temperature stays relatively constant. This is calculated using the following equation:

$$constant = \frac{\mu}{\rho h_{lv}} \quad (4.20)$$

The interesting results is that ethanol would theoretically, with this model, have the least pressure loss with a 21% reduction in vapor pressure loss as compared to water vapor and the liquid losses in all of the materials are almost 200% less than the water vapor losses. This shows that the vapor pressure losses represent the most significant losses in the system. The other important factors in the fluid selection process are also the latent heat, which dictates

the amount of heat it absorbs and the boiling temperate as this will be the stabilization temperate of the device if it is kept internally at atmospheric pressure.

Table 4.1: Possible Fluids for a LHP

	Vapor		Liquid	
Fluid	Constant	Percent Loss	Constant	Percent Loss
Water	8.3E-12	-	1.3E-13	195%
Methanol	8.3E-12	12%	3.9E-13	185%
Ethanol	7.5.6E-12	21%	5.9E-13	176%
Acetone	8.3E-12	11%	6.2E-13	175%

	General	
Fluid	Latent Heat (kJ/kg)	Boiling Point (°C)
Water	2256	100
Methanol	1100	65
Ethanol	9630	78
Acetone	506	56

Table showing the fluid properties and there effect of the frictional loss constant.

Overall this model is a window into the pressure loss in the LHP and can be used to predict the effects of device thinning and width changes on applied power ranges. An example of this analysis would be for ways in which the current μ LHP could be thinned. For this case the first thing to review is the maximum amount of pumping possible in the wick. Figure 4.6 is the capillary pressure in the wick for water and methanol, figure 4.7. The close up for under 10,000 pressure gain shows that in the methanol case a wick with pores $4.5 \mu\text{m}$ or smaller would be needed and for water pores $13 \mu\text{m}$ or smaller would be needed. Theoretically it is advantageous to have the wick pressure difference to be more than the maximum system pressure loss with. This is not infinitely true because there is a ratio of refill time that creates a delay in the fill of the wick [26]. The conclusion of that work in that once the pore size shrinks below $1 \mu\text{m}$ there is a delay in refill of the wick due to frictional losses that will stop the function of the device. With this in mind as well as concerns about fabrication limits $4.5 \mu\text{m}$ is the smallest reasonable pore size achieved in the μ LHP design.

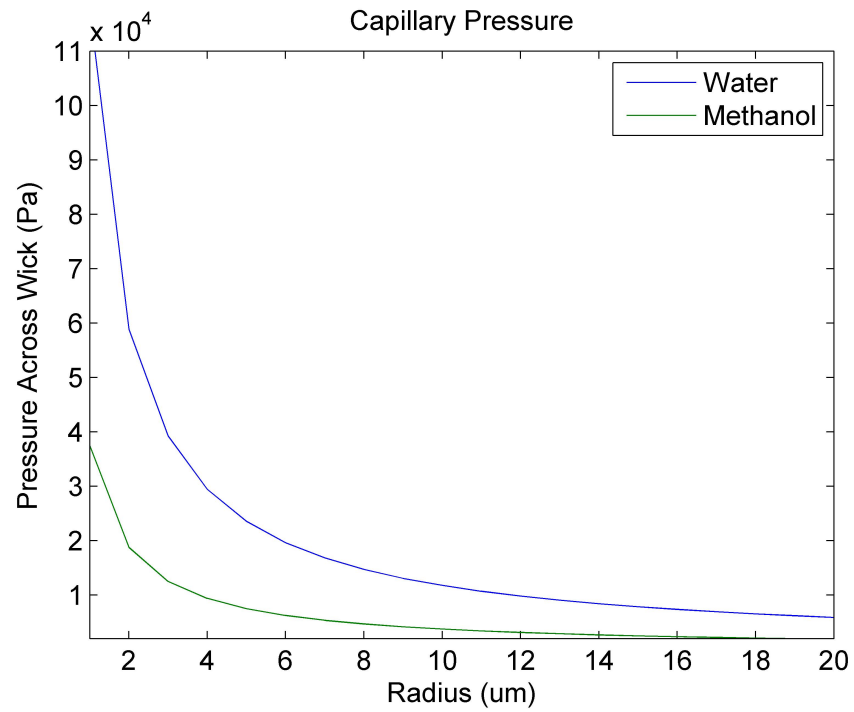


Figure 4.6: Capillary pressure graph for a wick with methanol or water in it.

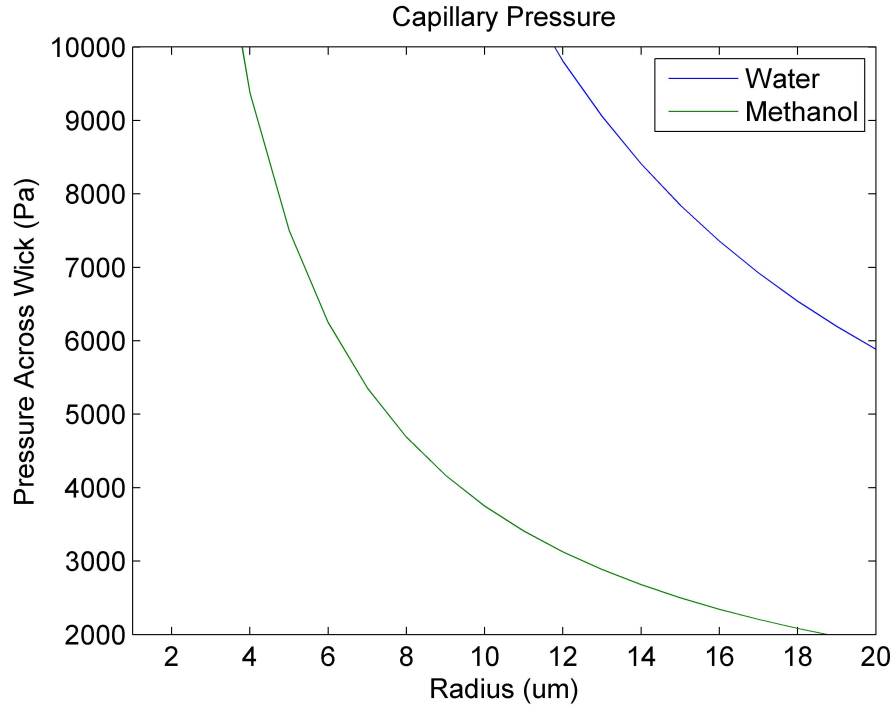


Figure 4.7: Close up image of figure 4.6

As a specific example using the derived equation, if the device needed to absorb between 10 and 20 W with a width of 9.5 mm the thickness could go down to 100 μm , with a wick that produces at least 6 kPa. The cumulative loss with water would be about 5.8 kPa and for methanol about 5.4 kPa. This would require a wick with pores smaller than 18 μm for water and under 6 μm for methanol. Even though methanol has less pressure loss it has much lower surface tensions so the wick produces 1/3 of the pressure.

4.4 Loop Heat Pipe Start up Conditions

A loop heat pipe device is considered a self starting two phase heat transport device. This means that the device starts solely from heat input without any additional external input. Although this is a known and important attribute of a loop heat pipe, a key thing to note is that just because it is self starting does not mean that the starting process will be quick or immediate. There are several necessary attributes of the device and physical requirements in order for circulation to begin. The main input for starting is heat flux into the evaporator region, but under certain conditions the loop may not start. These limitations and requirements make it critically important to understand the start up dynamics of a loop heat pipe. It is also important to know what loop start-up looks like so the data can be reviewed

and start-up can be observed. These and other start up related issues are discussed in this section.

Over the years there have been many studies into the operating conditions surrounding LHP start up [27] [7] [28] [29]. Most of the studies use the same condition to observe start up, which be seen by a temperature increase in the vapor channels and the inlet of the condenser. The physical theory behind this behavior is that when the circulation begins there is a shift of vapor from the evaporator into the vapor lines and the condenser and this shift shows the beginning of circulation. The shift in the vapor could be recognized by a temperature increase in those regions, when that increase is observed it is usually considered the onset of circulation. Before this temperature increase is witnessed there is usually a delay from the time the heat flux is applied and some overshoot of the evaporator wall temperature above the boiling point. This delay of circulation has also been specifically studied and a variety of factors can cause the delay or increase the time it takes for the circulation to begin. One of the factors that can delay circulation is the distribution of the liquid and vapor in the device. There are ideal conditions where, both the quantity and position of the liquid versus the vapor before start up, are important. Figure 4.8, shows different distributions of liquid and vapor in the evaporator and how they affect the time to start up and the evaporator wall temperature rise [30]. From the image it is clear that when there is liquid in the vapor channels (see situation 2 and 4) the evaporator wall temperature increases drastically, but the case where the wick is not entirely wetted (situation 4) takes much longer for start up to occur. The other key observation is that the more liquid contact that occurs with the wick the lower the settling temperature of the evaporator wall is once looping has begun. From a design standpoint any evaporator configuration, should try to keep liquid from the vapor lines and keep the wick completely wetted at all times. This is usually accomplished through specific vapor groove design and wick characteristics. Another important start-up delay metric includes the amount of material that the evaporator is made out of, or its thermal mass. High thermal mass at the evaporator will cause this region to store more heat causing it to get hotter than is needed to be for evaporation to occur. This can result in higher than usual temperatures at the evaporator wall or the device can stop circulating because the surface becomes too hot to form stable thin film evaporation or boiling. These are the most important start-up characteristics because they center around controllable geometric aspects of the LHP other behaviors that increase delay are more focused on external conditions.

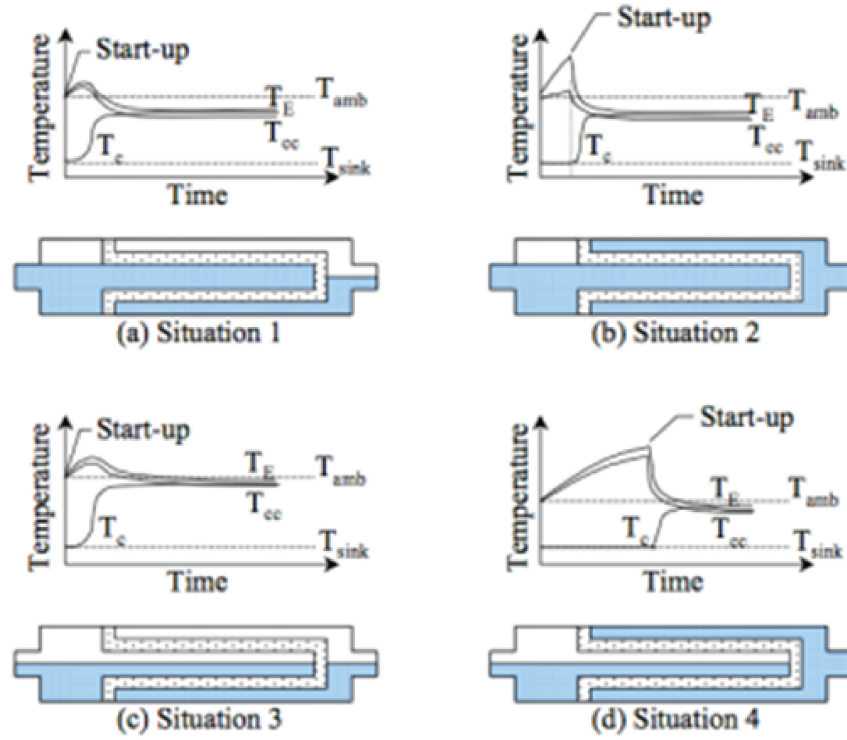


Figure 4.8: Schematic of main physical mechanism of liquid pooling in a LHP with start-up temperature effects.

In addition to the geometric concerns that can hinder start-up there are important external factors based around the way heat is applied and the external conditions around the LHP. One of these conditions has to do with the applied power supplied to the device to start circulation. For most large heat pipes start-up is more reliable with less temperature over shoot if input power is high. This is most likely due to the larger heat supplied causing the entire device to heat up faster limiting thermal gradients and decreasing start-up time. Ambient conditions being too cold or warm can also delay start-up. If a heat sink is applied to the condenser, the temperature of the heat sink being too low can cause the vapor to condense in the vapor channels. This would slow start-up or stop fluid circulation all together. When this happens it is physically similar to having lots of extra fluid in the vapor channels, which, as previously mentioned, increases peak temperatures and delays start-up. These are the external conditions that can delay start-up. All other conditions that affect start-up are based on the fluid and any non-condensable gasses in the system [31]. These issues are fluid specific and should be reviewed, but are not discussed here.

One additional reason why start-up is such an important factor is that the initial heating up of the device can actually effect the entire system behavior. Research into this area has shown four possible modes that can result from different start-up heat applications.

These modes include: under heat failure mode, overheat failure mode, oscillation mode, and normal mode [7]. Under heat failure is when the device never gets enough heat to begin circulation. Overheat failure is when the device has too much heat applied and this causes failure. Oscillation mode is when the applied heat is right above or right below the right amount of heat. In the case where there is not enough heat the device will begin to circulate for a moment and then absorb so much heat that the evaporator surface cools, stopping the flow. On the high end this can happen where fluid reaches the surface starting the loop and circulation begins, but the surface dries out in places, there is then less evaporation that is needed to get the loop going, but eventually some vapor builds starting the loop temporarily and then it stops. An alternate theory is that the wick has such small pores that there is a fluid refill time that starts the flow, waits for the fluid to refill and then it begins again, but no studies of this theory have been conducted. The reason that start-up affects the stable operating conditions is that during start-up there is heat leaking from the evaporator into the compensation chamber and the fluid lines. If start-up takes too long these areas can get hot and then evaporation can occur in areas other than the evaporator ruining the pressure distribution in the device [30].

The last physical interaction that is also important to understand LHP start-up is the creation of a pressure differential in the wick that initiates and sustains circulation. As heat is applied to the evaporator wall the top and bottom of the wick become different temperatures and as Clausius-Clapeyron (see section 4.2 equation 4.4) shows that temperature difference leads to a pressure difference. The pressure difference begins the pump. For low heat input the pressure difference is too small to start circulation, so the heat is not dissipated or transported. In this case the conduction through the wick becomes the main heat flow path. This can cause a vicious cycle where the bottom of the wick is too hot to begin circulation and as heat is steadily applied this condition never changes. In practical terms this means that any LHP has a minimum heat flux input requirement. With this requirement in mind it can be concluded that certain LHP evaporator designs should allow for lower minimum input heat flux.

Chapter 5

System Assembly and Testing

5.1 System Assembly

The hybrid μ LHP (HLHP) was composed of two silicon components the CPS wick and the interline evaporator and an aluminum shell that acted as the piping to move the vapor and liquid between the condenser and these points. The wick was fabricated in the process described in the following paper [1]. The MEMS fabrication process for the interline evaporator follows standard silicon deep reactive ion etching process [22]. The aluminum was machined using a CNC mills from two 1.5 mm plates making the total thickness of the assembly 3 mm. All of the components can be seen in figure 5.1. The window, aluminum barrier and the roughened interline evaporator were attached using a methanol resistance epoxy from Master Bond called, EP41S-1ND. This was a 2 part epoxy that was mixed and attached to the back side of the silicon and the edges of the window. It is then left for 24 hours then baked at 90°C. Once all of the components were attached and hardened the CPS wick, see figure 5.2, was placed in the bottom plate and a silicon gel gasket was added. The top layer could then be attached and bolts were fastened with nuts to hold the assembly together. This was the first step in the experimental process.

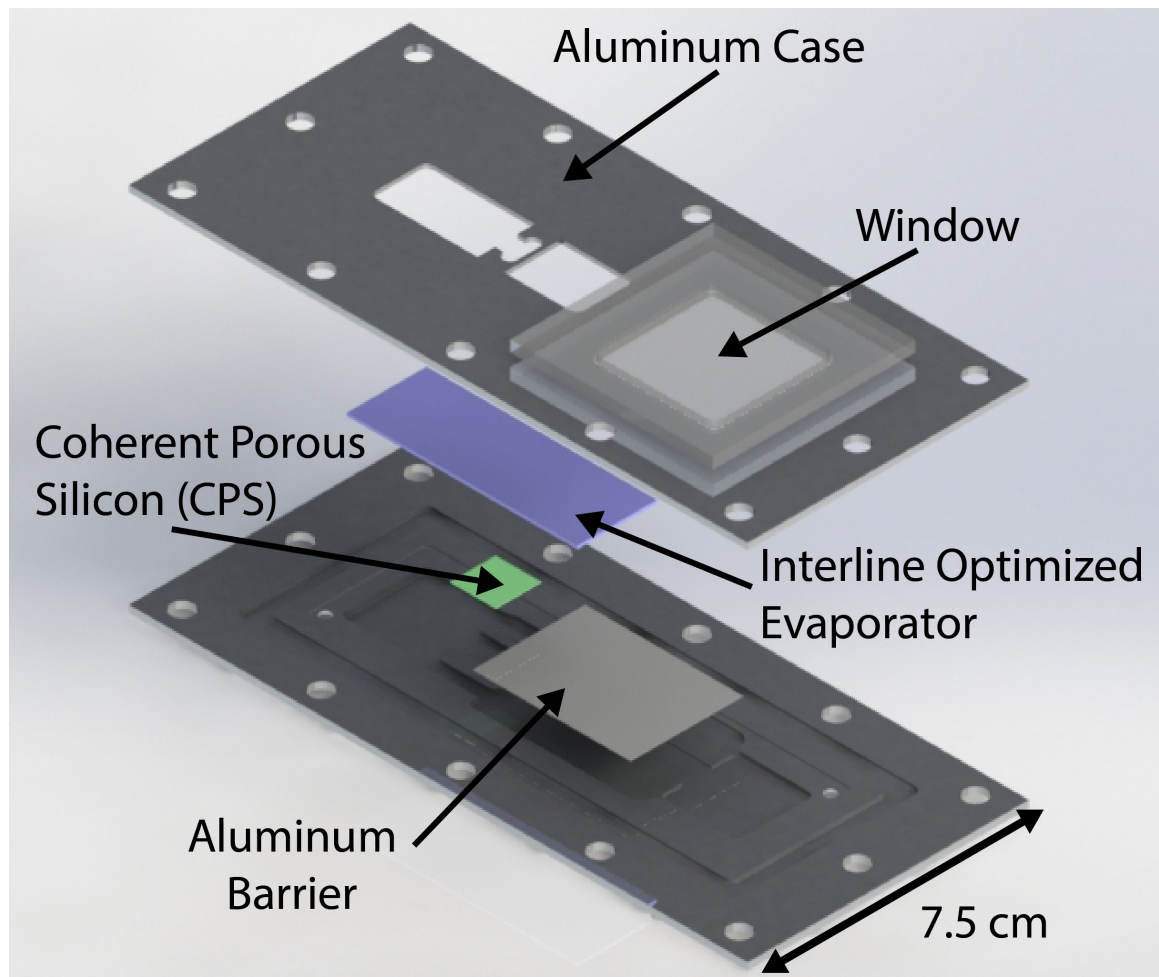


Figure 5.1: Exploded view including all the parts of the hybrid loop heat pipe.

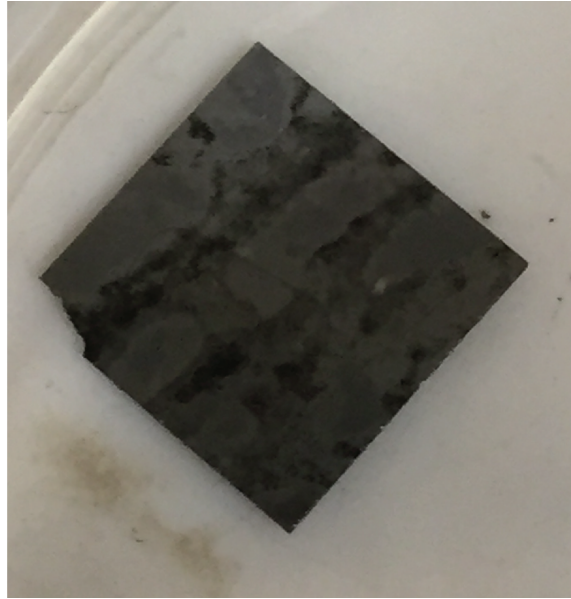


Figure 5.2: Image of the CPS wick used in the hybrid loop heat pipe.

After assembly thermocouples were arranged on the device to measure the temperature at various locations. The thermocouple placement can be seen in figure 5.3. There are 14 thermocouples (TC) in total with TC 2 being the thermocouple under the heater in between the heater and the silicon. TC 3-5 are on the evaporator and are used to determine if the fluid is condensing in the vapor lines. TC 6-8 are in the condenser region and can be averaged to get an idea of the condenser surface temperature. TC 9 and 10 are in the liquid line and can indicate whether there is boiling in the liquid return lines. TC 11-13 can be used to tell the average temperature in the compensation chamber. Finally TC 14 indicates the temperature under the wick and because the metal is in direct contact with the wick this can also be used to infer how much heat is leaking back through the wick. Once all of the thermocouples are on the device filling can begin.

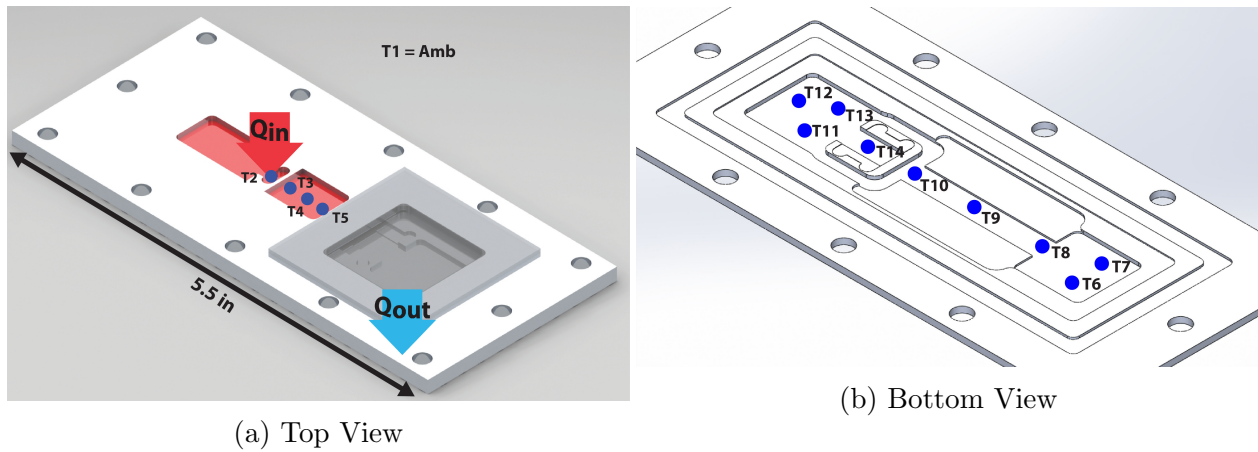


Figure 5.3: Schematic of thermocouple locations on the HLHP.

5.2 System Filling

The filling process had two steps first there was the initial step of checking that the bottom plate filled with liquid in an expected manner and then there was the filling process before a system test. The initial testing was also in part to determine what the compensation chamber looked like when filling in comparison to the condenser. This was important because the only view point into the device was the condenser. Figure 5.4 shows the device filling without the top plate and as the device filled once the system has a certain amount of liquid inside pooling on top of the CPS wick could be viewed. This was tested several times to determine the approximate condenser fill level that resulted in liquid pooling at the wick. This was a practical step that allowed for an empirical approximation of the needed fill level of the device.

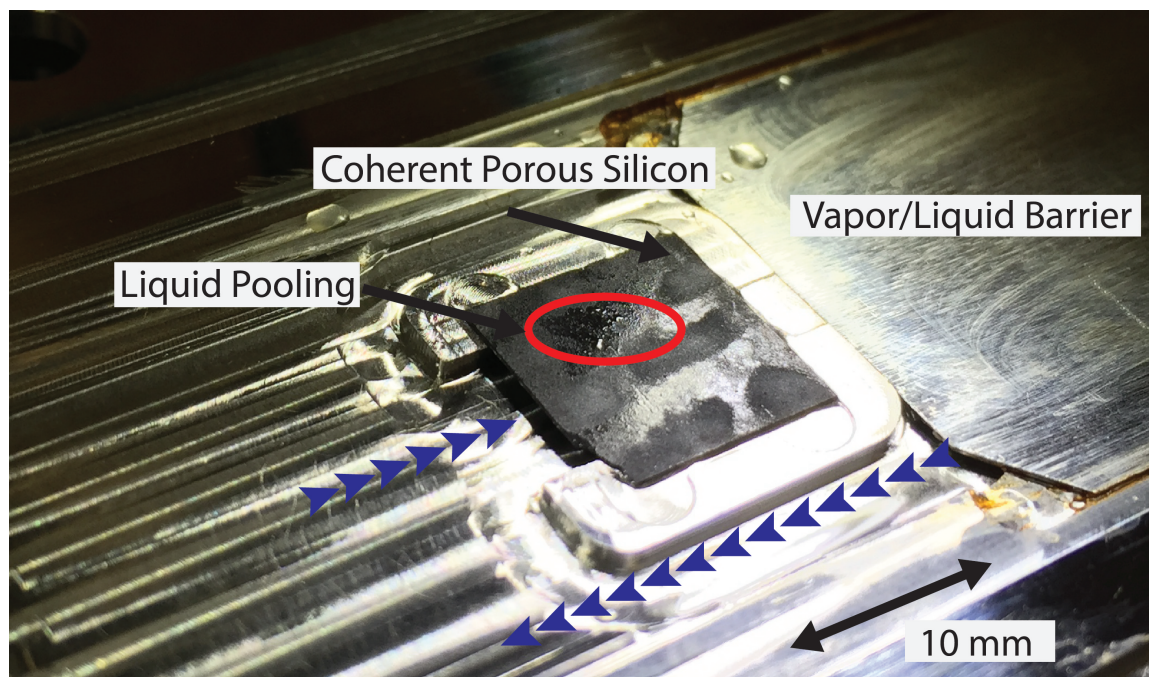


Figure 5.4: Image of liquid pooling on top of the CPS wick during initial filling tests.

Once the approximate fill level was determined the LHP filling process for testing was created. Filling the LHP is a process that is specifically designed to remove all non-condensable gasses and leave the system filled with only methanol vapor and liquid. The first step in the filling process is to place an Erlenmeyer flask on a hot plate set at 100-150°C. This temperature range is above the boiling point, so it will remove any dissolved gasses. It is also high enough that it will create some pressure to move the vapor into the device, but not so high that it will over pressure the device above atmospheric pressure. Once the methanol is boiling a rubber cap is added with a piece of chemical resistance clear Tygon tubing. This tubing connects to a fitting that is then attached to an open valve that is screwed in to the bottom aluminum plate, see figure 5.5. The valve has a silicone gasket keeping it sealed to the plate. Once the pressure inside the flask builds vapor and slugs of liquid will begin traveling into the device. Once the device is full liquid will begin to drop out of the fluid outlet valve. To minimize the chance of having non-condensable gas in the device the device is filled for 5 minutes starting from when liquid begins to drip out of the outlet valve. Then the outlet valve and inlet valve are closed and screw in valve plugs are added for maximum hermeticity. The device is closely monitored during filling to see if any liquid is leaking from undesirable locations like the edges, the window or the silicon aluminum connection. Once the device is filled testing can begin.

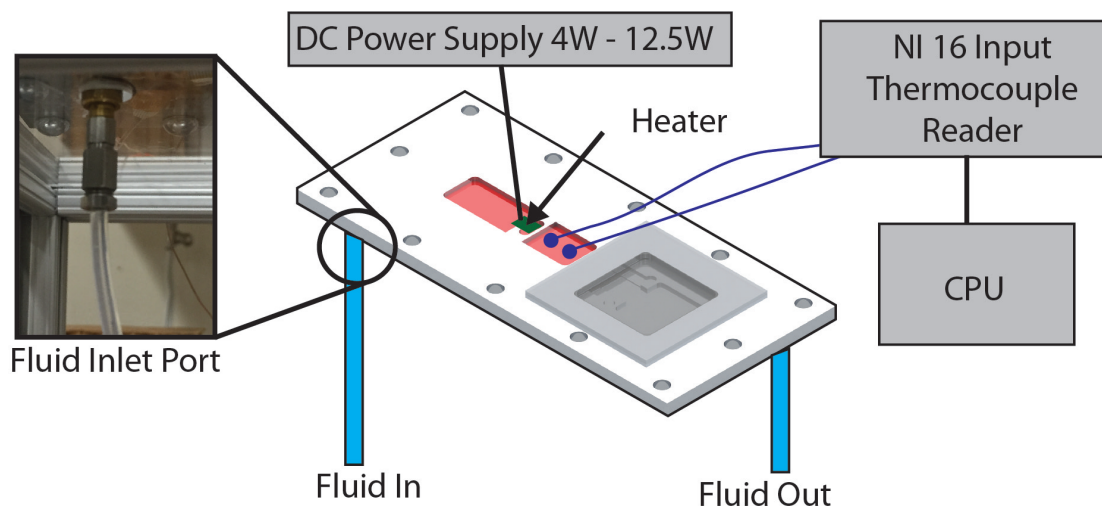


Figure 5.5: A schematic of the test set up for th HLHP.

5.3 System Testing Procedure

After filling device testing began. A Labview program was written to read the thermocouple data and once this was turned on data was recorded. From here the DC power supply was turned on and cycled through a range of input heats. The first tests were at low input power to verify the system wouldn't melt. These inputs ranged from 0.4W - 7W in increments of around 0.2W. The DC power supply was manual so the input powers are approximations. Once the system was determined to be safe, testing that cycled the power from 0.5W to 12.5W in 0.5W increments, and increasing the power every 30 seconds were conducted. Once the overall device temperature began to approach 90°C the device was shut down for safety. An important thing to note about these tests is that these were ambient air tests with no chiller on the condenser, so all heat removal from the condenser is natural convection. This was the testing of the device, figure 5.6 shows the device during testing.

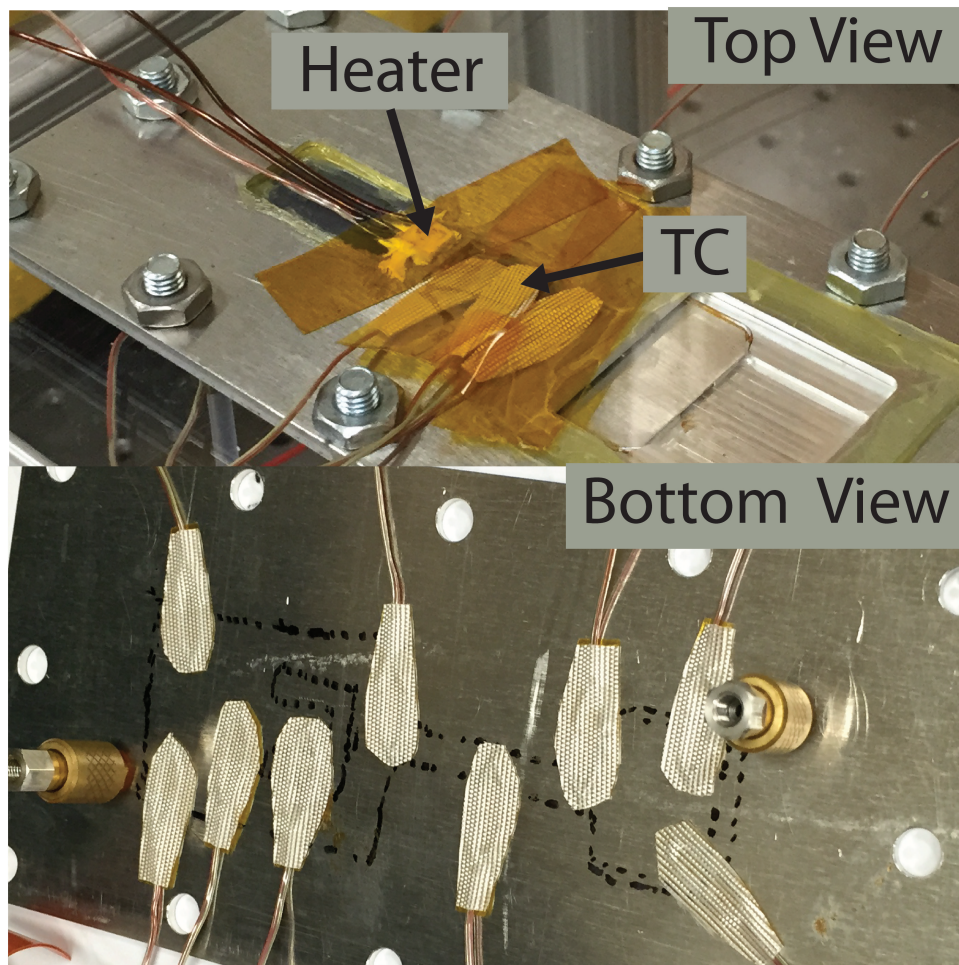


Figure 5.6: Image of the experiment with thermocouples and heater placement.

Chapter 6

Results and Discussion

6.1 Metrics

Once the measurements were taken the data was analyzed and metrics were calculated to gain a sense of the efficiency and overall function of the device. The metrics of operation used to characterize the operation of the μ LHP device are the max heat input, the overall thickness of the device, the distance the heat traveled (sometimes called heat throw) and the thermal resistance ($R_{\mu LHP}$). The thermal resistance is calculated using the following equation:

$$R_{\mu LHP} = \frac{T_{evap} - T_{cond}}{\dot{Q}} \quad (6.1)$$

Where T_{evap} can be the thermocouple under the heater or the first thermocouple in the vapor line. T_{cond} is the average of the thermocouples on the condenser TC6-8. The thermal resistance can be thought of as the inverse of thermal conductivity so a low thermal resistance means that the device moves heat effectively.

6.2 Results

The testing described in chapter 5 resulted in the following temperature profile data. In this data the thermocouple under the heater has the highest temperature and the second highest is the first thermocouple in the vapor line. Due to the large number of thermocouples that are used to track unexpected behavior there is a lot of data in this graph, see figure 6.1. The notable trends are that the thermocouple at the end of the vapor line is the coldest. On the other side TC14, which is under the wick is warmest thermocouple. Finally there are three TC's right around T9 on the graph that represent the temperature of the condenser. These temperatures were averaged for the thermal resistance calculations. The trends of this data

show that some regions reach stable temperatures, while others continue to increase. The testing increased the temperature from 4W to 13W. The data (figure 6.1) shows that at 4W the device is heating up, but does not begin to cycle. Once, 8W is applied boiling begins, this is evident from the stabilization in the bulk of the thermocouple measurements. This is stable until around 12.5W when the device begins to overheat and thermal runaway begins. This data is promising, stable boiling and looping is evident.

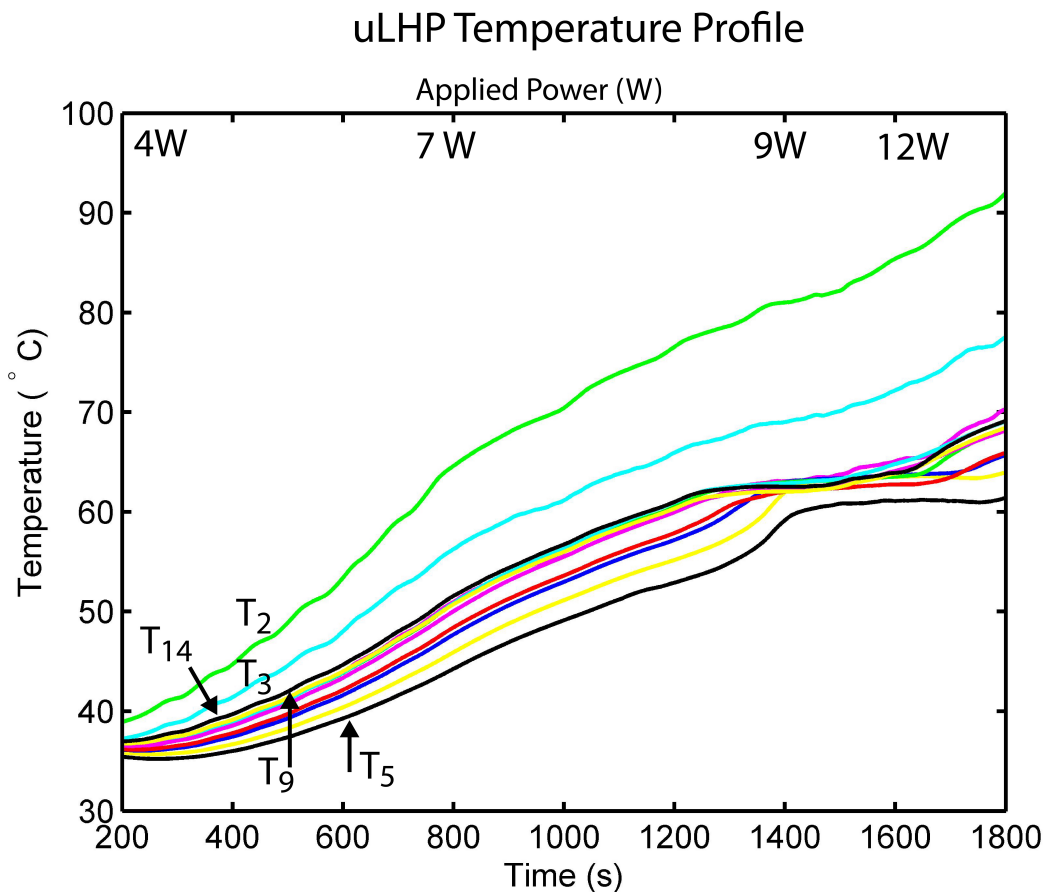


Figure 6.1: Temperature profile of the HLHP when the power is cycled from 4W to 13W.

Additional tests at single applied powers show start-up behavior and more data on the device function. The stable temperature tests were run at 5W and 6W, other tests at 7W were also run, but the device never started circulating so the tests were stopped. In figure 6.2 the trend of the device during the 5W tests can be seen. Start up of the device begins at around 200s and the device bulk to the device remains at a stable temperature for several minutes. Towards the end of the test the device does start to heat up. It is also clear from this data that the heater area never stabilizes and is continually heating, which is not ideal. Figure 6.3 for the 6W tests also shows the device operation. This data is much less clear

in its trends as the device seems to have some start-up transients. It does appear that the device begins to loop cycle at around 150s, so slightly before the 5W tests, but it does not operate as long as the device begins to dry out at around 600s. This is the operation of the device under constant input power.

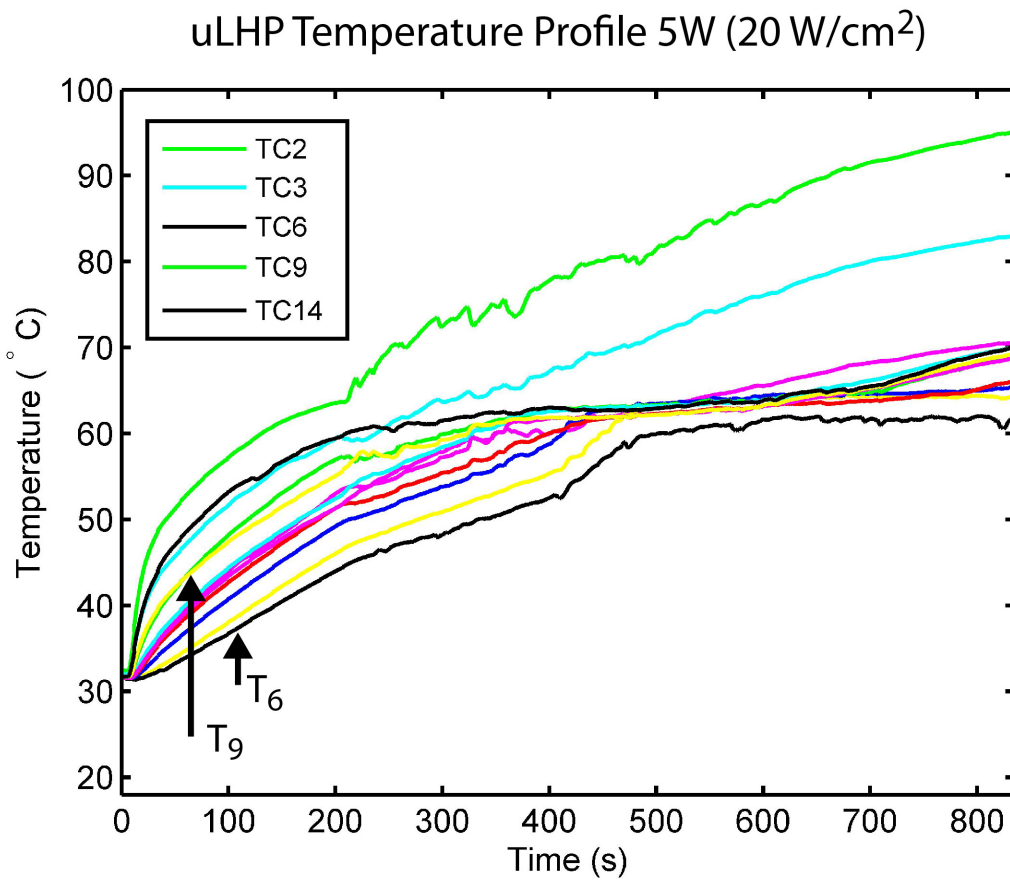


Figure 6.2: Temperature profile of the HLHP when the power is kept constant at 5W.

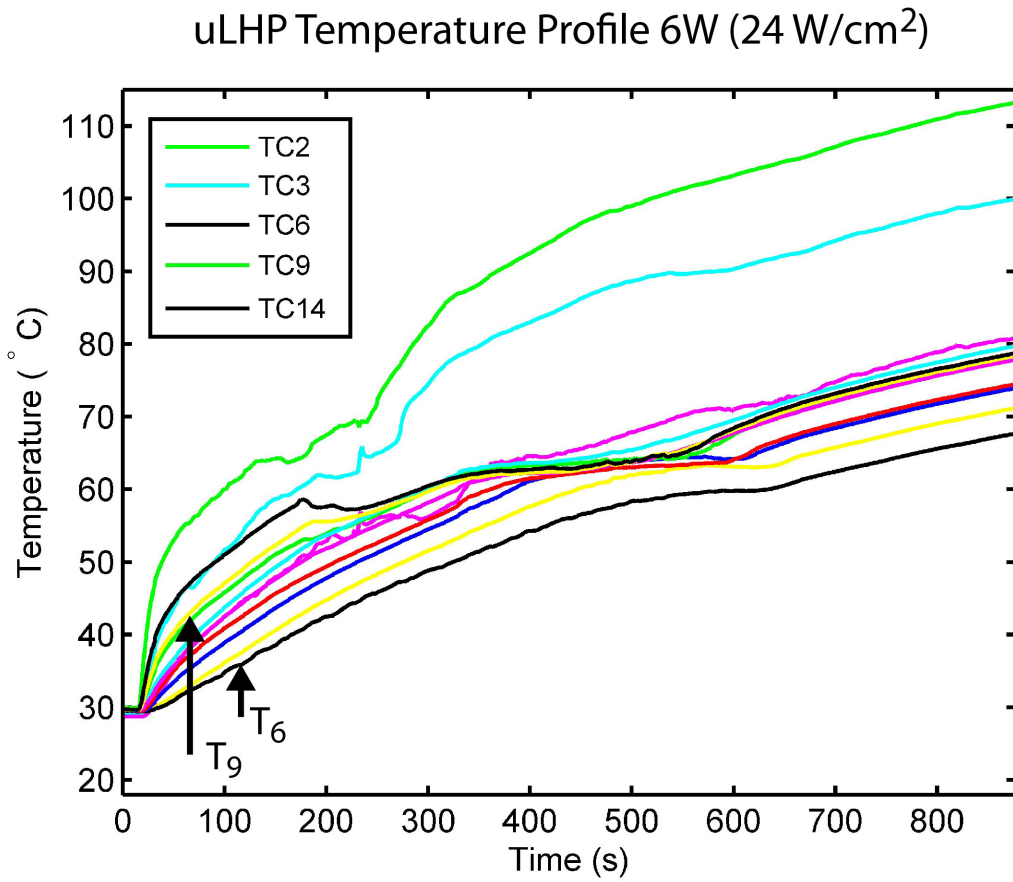


Figure 6.3: Temperature profile of the HLHP when the power is kept constant at 6W.

From the temperature profile data resistance data was calculated using equation 6.1. Most loop heat pipe data looks like an exponential curve where at the low fluxes the resistance is high because the device takes a certain amount of time to heat up. The μ LHP does not seem to have this problem, this is most likely due to the device thickness, see figure 6.4. A large heat pipe has a lot of material that has to heat up, but this system is so thin that it heats up quickly, so the resistance remains fairly constant. The trend in the data appears that there is an increase in the resistance as more heat is applied.

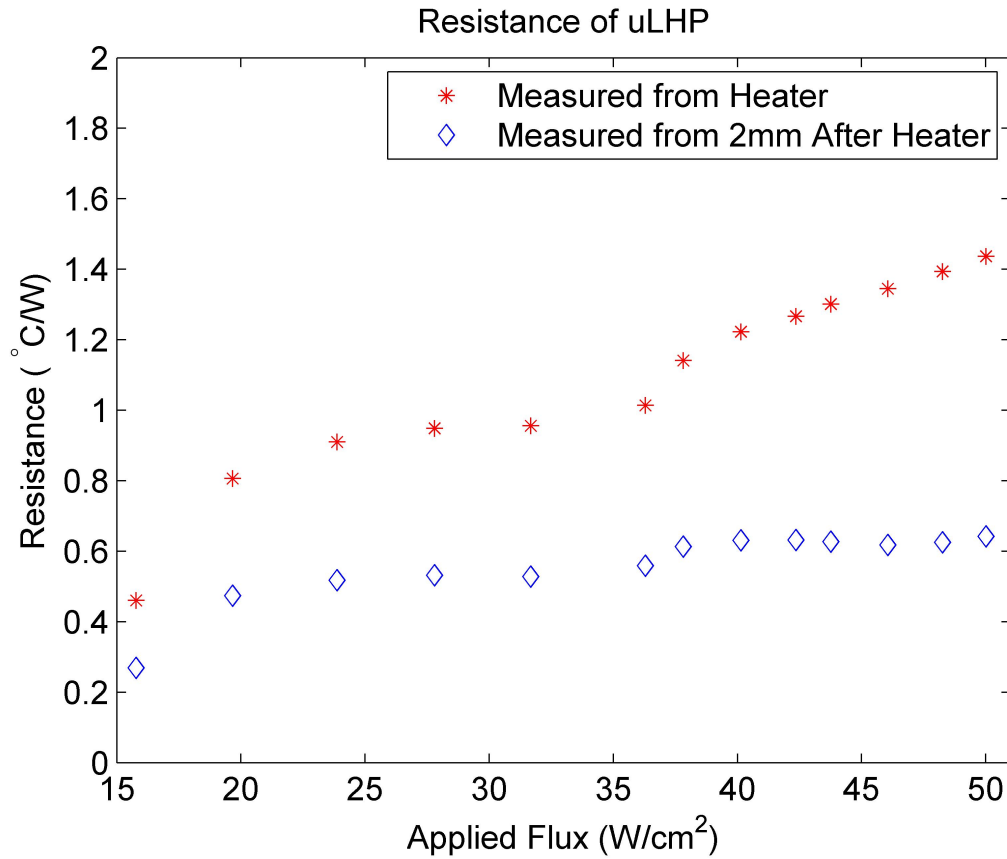


Figure 6.4: Graph detailing the thermal resistance of the HLHP as flux is increased.

6.3 Discussion and Conclusions

The results measured from the hybrid loop heat pipe device confirm the start-up cycle, looping and general function of this design. With that in mind there are some trends in the data that are not favorable and show room for improvement. The first of these trend is that start-up time. This device took a long time to get to stable function. This is something that can be improved by further thinning of the device as well as making the device all one material so that there is no insulating epoxy between components. In addition if the data is viewed with fewer thermocouples like in figure 6.5 and 6.6 some other trends become more evident. In figure 6.5 it is easy to see that the thermocouple under the wick heats up quicker than the thermocouples in the vapor line and condenser. This trend may indicate that the stable circulation of the device does not actually start until these other regions heat up at around 500s, were those points are seen to reach stable temperatures. There is also a clear point at 700s where the TC14, under the wick, starts to increase in temperature and actually increases above the boiling temperate of methanol. This indicates the failure mechanism of

the device. It seems like from this data that the device fails when there is boiling in the wick and once this began there was no recovery in the device. Figure 6.6 also seems to have this trend and arguably may have never reached stable function. The data never really reaches a stable temperature, but if it is approximated at around 500s then the wick is seen increasing temperature into boiling at 550s so the circulation may have only occurred for under 1 minute. These results indicate important mechanisms about the device and will help with future improvement to the design.

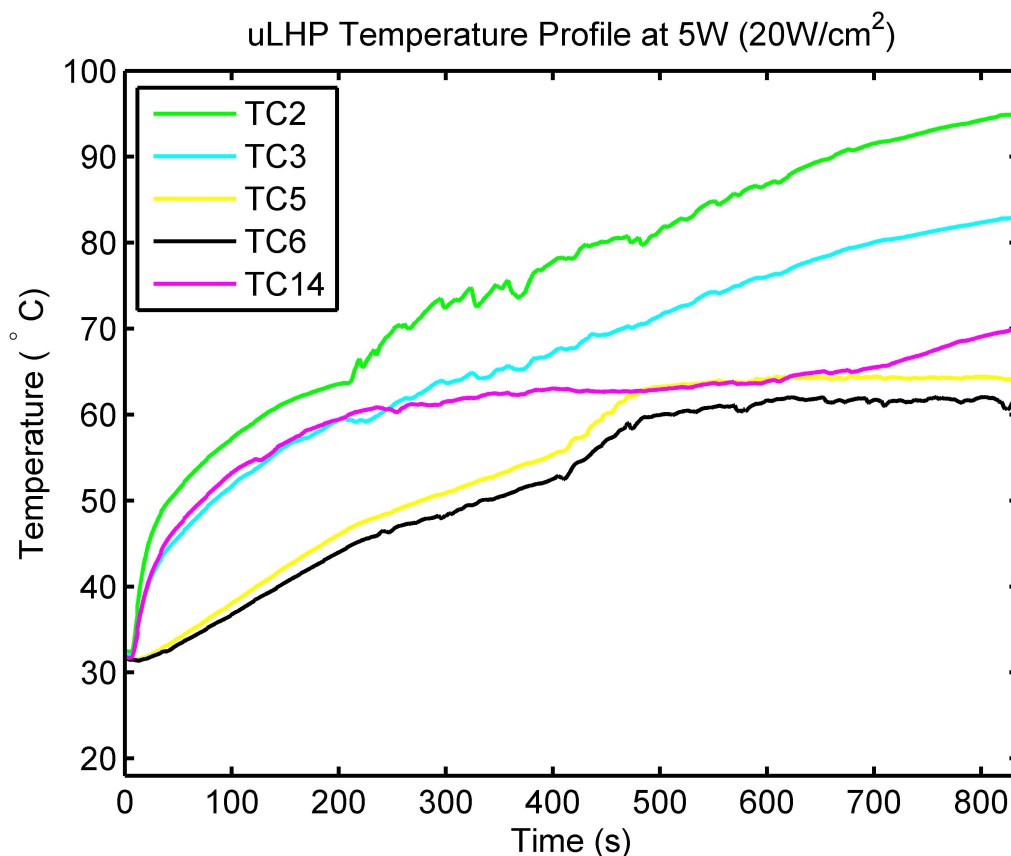


Figure 6.5: Temperature profile of the HLHP when the power is kept constant at 5W with fewer TC data lines.

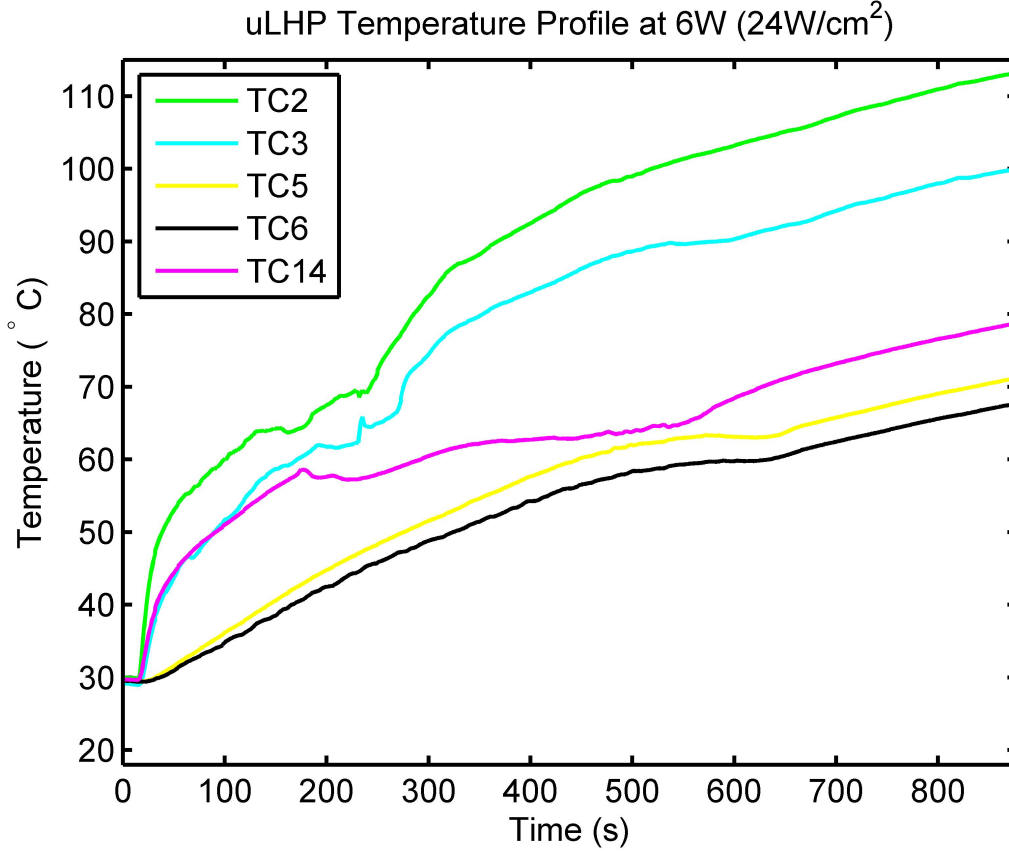


Figure 6.6: Temperature profile of the HLHP when the power is kept constant at 6W with fewer TC data

The thermal resistance data shows a consistent device that could still use some improvement. The trend in the data where the thermal resistance increases as flux increases shows that the device is not absorbing all of the applied flux. This is also evident in the temperature profile data. The heater temperatures never reach a constant stabilization temperature it is always rising. If the device was working as expected the thermal resistance would decrease over time because the power would increase, but the device would stay at a constant temperature.

The dry out conditions seen in figure 6.5 and 6.6 as well as the resistance data seen in figure 6.4 are most likely a result of two weaknesses in the current device: leaking and insufficient wick etching. Leaking has always been a huge issue in the TGP devices and the hybrid device despite many efforts to combat this issue was no different. All of the work done to determine methods for sealing TGP device (see section 3.4) ended up not helping in the aluminum silicon hybrid device. This was in part because there is a large thermal mismatch between these materials and in part because the epoxy would eventually fail after a certain number of tests, making repairs necessary. During testing the devices could be

run once and then afterward they needed to be refilled this implies that there were losses somewhere in the system. In future testing the aluminum will either need to be welded shut or the system should move to the all silicon device.

The wick was also a weakness of this design figure 5.4 and 5.2 in chapter 5 are clear evidence of the issues with the wick. In these images the dark patches in figure 5.2 show the only regions where the pores were etched all the way through the device and in figure 5.4 the liquid pooling is seen only in a few regions. The reason this was a problem was that etching the pores was difficult and the samples were prone to breaking, so a consistent method of getting all the pores though the silicon was never developed. More research into this problem will help the final μ LHP device, although on the wafer scale there will be huge advantages to making pores all the way through the wafer because the pores will only be etched in a small region while the rest of the wafer would be solid silicon making it less fragile and more manageable. Overall the results seem to indicate that the evaporator design works in a rudimentary sense and that the next steps of full device integration using more of the models shown in Chapter 4 will see the end goal of this research met.

Chapter 7

Conclusions and Future Work

The final hybrid loop heat pipe system tested here is just the beginning of the system integration work that can be done. The majority of the research done on this loop heat pipe system was component level development. These components were the heart of the focus of this work. As explained earlier, the approach for this project and the loop heat pipe development was to build a system meant for the micro scale with specific advancements meant to maximize the efficiency by utilizing micro scale physics. The wick is a great example of this, using micro pores to increase capillary pressure by using surface tension to create an advantage. The evaporator, where the efficiency of the device is really determined, uses capillary forces and pinning to create a thin film of liquid on the surface. This thin film then maximized the heat transfer coefficient improving the operation of the system. Once the components were developed and their function was well understood they could be used in a variety of ways to make the next generations of cooling technology.

Phase change systems have a clear functionality that requires flow and circulation to be maximized in order for the device to function properly. Previous research on maximizing efficiency in phase change systems focus on components with static fluid not dynamic fluid flowing through them. Convective pool boiling is a classic example of this. Some surfaces could create high heat transfer coefficients, but they had such high frictional losses that they cannot be used in loop heat pipes. The work done here converted the previous technology to something that could be useful in these dynamic phase change systems.

The system presented here showed stabilization and low thermal resistance. This device handled heat input up to 12W or about 50W/cm². The mechanism of wick heating that causes device failure was identified as well as the slow start-up time. The device also currently has slow response times to heat input. Both of these issues can be fixed by further thinning of the device, which will lower the thermal storage of the device. The thermal resistance was found to always remain under 2°C/W. All of this data as well as the high heat transfer coefficient, determined from the interline evaporator and flow improvements from the CPS wick, show that this system is on the way to becoming the next big thermal management solution. Overall the μ HLHP system showed great promise, but because this is a component system with two plates versus a fully integrated and bonded surface from one material there

are unsolvable issues. These are mostly due to leaking that is difficult to prevent when the system is gasket sealed.

The future of this work is to integrate the system into an all silicon device with three layers made of silicon, see figure 7.1. The best part about this integration is that the work done here on each component and on the basic system can be tuned to make a device designed for a variety of industry needs.

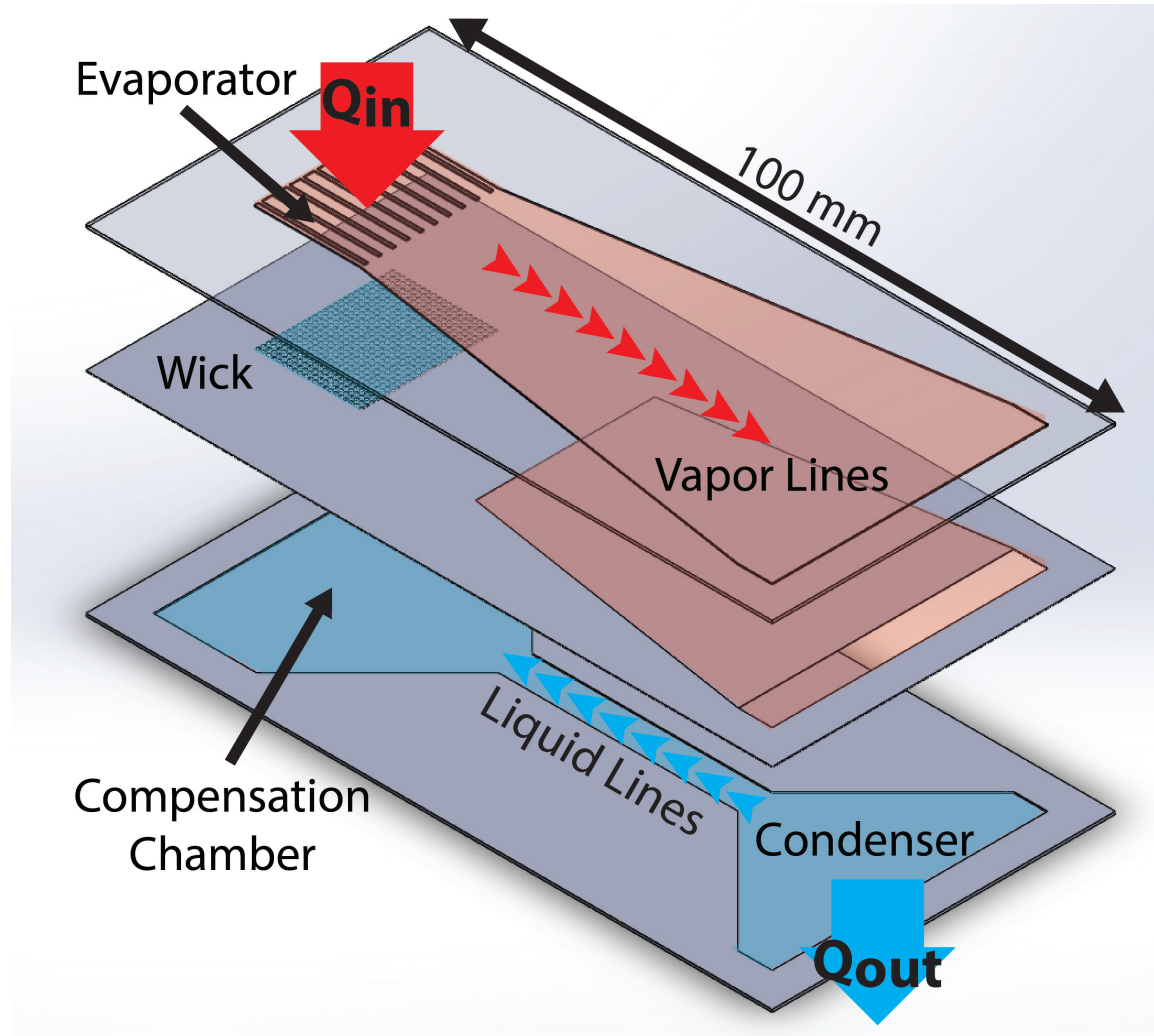


Figure 7.1: Three layer stack of possible future silicon μ LHP.

The vision for future devices is to continue the wick development, use the evaporator surface knowledge, and the modeling for the piping pressure losses to make devices for many different applications. In future iterations; the μ LHP may be made of silicon or some other material, but the overall device should be made and hermetically bonded. With that in mind based on the results shown in Chapter 4, the channel thickness could be shrunk to as

thin as $100\text{ }\mu\text{m}$ if the wick has sufficient capillary pressure; although slightly thicker like $200\text{ }\mu\text{m}$ will theoretically be a much more stable device, because the pressure losses will not be as sensitive to external fluctuations. In addition once the entire device is made from one compatible material and bonded so that all leaking issues are avoided higher power testing with increased internal pressures can be performed. There is also some work in testing the device with fans or heat sinks on the condenser to see if the device performance can be improved further. This all seems possible based on the 12 W dissipation with only natural convection on the condenser. Theoretically with a larger temperature gradient in the device this can be significantly improved.

The future design should be created with the idea of the failure modes in mind. The first failure mode is the risk that droplets will form in the vapor lines and stop circulation. This is a higher risk if the channels are thinner because this will decrease the vapor region and increase the pressure losses. The other failure mode is the wick heating up and evaporation occurring in the wick instead of on the evaporator surface. Future designs will need to be a balance of thinning the device and robust function. There is also a lot of future work in isolating the wick from the hot spot, which may utilize a filling method, where the wick is placed further from the hot spot so that there is a larger liquid buffer between the wick and the heater. This may prevent dry out, but will need to be thoroughly explored and could be a huge advantage to protect against thermal runaway.

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