Abstract

A condenser design was characterized for a multi-condenser loop heat pipe (LHP) capable of dissipating 1000 W. The LHP was designed for integration into a high performance air-cooled heat sink to address thermal management challenges in advanced electronic systems. The multi-layer stack of condensers utilizes a sintered wick design to stabilize the liquid-vapor interface and prevent liquid flooding of the lower condenser layers in the presence of a gravitational head. In addition, a liquid subcooler was incorporated to suppress vapor flashing in the liquid return line. The condensers were fabricated using photo-chemically etched Monel frames with Monel sintered wicks with particle sizes up to $44 \, \mu m$. The performance of the condensers was characterized in a custom experimental flow rig that monitored the pressure and temperatures of the vapor and liquid. Two condensers arranged in parallel were demonstrated to dissipate the required heat load while maintaining a stable liquid-vapor interface with differences in liquid and vapor side pressures in excess of 6.2 kPa. The experimental results defined the stable operating limits of multiple condensers within a LHP given a range of convective heat transfer coefficients and differences in liquid and vapor side pressures. The inclusion of a wicking element in the condenser of the LHP increases the flexibility in design by allowing a modular construction with multiple condensers which can be integrated into air-cooled heat exchangers to cool devices with high power density.

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Acknowledgments

I first thank my adviser, Professor Evelyn Wang, for her persistent support of my research and for stimulating my development as an engineer and researcher. I thank Professor Brisson for teaching me to approach problems critically and creatively as well as for his counseling on my ever-improving technical writing skills. I also received tremendous, selfless support from the PHUMP team of postdocs and peer advisers: I thank Dr. Teresa Peters for helping me get started on the project, reviewing my thesis, and always being available for assistance. I thank Dr. Martin Cleary and Arthur Kariya for being steadfast mentors regarding the multitude of questions, ideas, and debates that arose in my research. I thank Wayne Staats for his help with airflow data and for his inspiring example as an experimentalist, machinist, and innovator. I thank Jay Sircar for his tireless assistance in fabrication and testing of condensers, David Jenicek and Kai Cao for their support with motor controllers, and Nicholas Roche for his appreciation of my defective condensers. I thank my labmates in the DRL for their support and encouragement. Lastly, I thank the Defense Advanced Research Projects Agency (DARPA) for funding the research through the Microtechnologies for Air-Cooled Exchangers (MACE) program with Dr. Thomas Kenny and Dr. Avi Bar-Cohen as program managers.
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Chapter 1

Introduction

1.1 Motivation

Thermal management is a critical limitation for many devices with high power densities including microprocessors, solar cells, fuel cells, radar, and microwave systems. The performance and reliability of computing, for example, is limited by the rate at which thermal energy can be rejected from a microprocessor which can exceed 100 W/cm$^2$ [1]. Additionally, the electricity demand of all data centers in the United States accounts for 5%-10% of national electricity consumption and half of this power is consumed by the cooling systems alone [1]. Thermal management solutions are needed to efficiently dissipate high thermal loads from these power-dense systems.

1.2 Background

With the exponential increase in transistor density, heat flux and total heat generation in microprocessors are increasing dramatically [1]. To effectively reject the increasing heat loads while maintaining the chip at a reliable working temperature, liquid cooling and phase change heat transfer has been considered such as thin-film evaporation, flow boiling, and enhanced pool boiling. Although these technologies promise to effectively dissipate highly localized thermal loads, they suffer from complexities due pumping needed to circulate the fluid and risks of electronic component corrosion due to leakage [2]. In contrast, air-cooling for thermal
management, offers advantages including simplicity, compact packaging, and an abundant, dielectric coolant supply [2]. Because convective heat transfer coefficients for air are limited by its low heat capacity and low thermal conductivity to about 200 W/m²K [3], an effective air-cooled heat exchanger requires fins to increase the surface area available for convective heat transfer. The basic fan and finned heat sink combination is fundamentally limited by both the thermal conductivity of the fin structure, typically copper or aluminum, and by the convected heat transfer, generated by an externally mounted fan. These combined thermal resistances form the overall effective thermal resistance which for state-of-the-art commercial air-cooled heat sinks can be as low as 0.2 °C/W [4, 5]. To improve the fin efficiency, passive phase change heat spreaders such as heat pipes have been integrated into air-cooled heat sinks.

1.3 Description of the System

This study investigated the enhancement of air-cooled heat exchangers by integrating a high performance blower into a loop heat pipe (LHP) with a single evaporator and multiple condensers which serve as fins to increase the surface area available to convection. The low thermal resistance of the LHP leads to a nearly isothermal surface temperature which improves convective heat transfer compared with a solid conducting heat sink. Traditional LHP design does not allow for stacking of components or parallel condensation pathways due to gravity-induced liquid pressure which are prone to flood some condenser layers. The multi-condenser LHP utilizes a sintered wick design in the condenser to stabilize the liquid-vapor interface and prevent liquid flooding of the lower condenser layers in the presence of a gravitational head.

The air-cooled heat exchanger aims to dissipate 1000 W from a heat source at 80 °C to ambient air at 30 °C using 33 W of input electrical power. These requirements correspond to a total temperature drop $\Delta T = 50$ °C, an overall thermal resistance $R_t = 0.05$ °C/W and a coefficient of performance COP = 30. For comparison, the state-of-the-art air-cooled solution for dissipating 1000 W features $\Delta T = 200$ °C, $R_t = 0.2$ °C/W, and COP = 10 [4]. To integrate such a cooling device in a variety of packages, the heat sink needs to be
Figure 1.1: Schematic of integrated heat sink. Cool air is drawn in from the top and blown radially outward by impellers which are interdigitated between fins. The heat sink is a loop heat pipe consisting of a single evaporator and multiple condensers with a low thermal resistance.

contained in a compact space of 10 cm × 10 cm × 10 cm and be capable of operating in various orientations. A schematic of the LHP and integrated blower is shown in Figure 1.1 in which a single evaporator accepts heat and multiple flat-plate condensers are cooled by air convection. The LHP utilizes two vapor and two liquid transport lines, respectively, which also provide structure for the LHP assembly and top-mounted motor. Water was chosen as the working fluid in the LHP due to its high latent heat of vaporization and surface tension. Cool air is drawn in from the top through a central core and blown radially outwards between each condenser. To further enhance convective heat transfer, the blower is integrated into the heat sink with 1.5 mm thick impellers which are interdigitated between the condensers that are spaced 2.5 mm apart, leaving 0.5 mm between the impeller and condenser surface.
This arrangement leads to high convective heat transfer coefficients due to shearing of fluid boundary layers and developing flow in the gaps between condensers [6]. This impeller design combined with a low temperature drop ($\sim 5^\circ C$) between the evaporator and the condensers leads to a near isothermal condenser surface and maximal temperature drop to surrounding ambient air [7].

1.4 Multi-Condenser Loop Heat Pipe

A schematic of the multi-condenser loop heat pipe with the wicking structure in the evaporator and condenser is shown in Figure 1.2. The device accepts heat from the evaporator at the bottom, where liquid is vaporized and escapes via connected horizontal vapor channels to the vertical vapor transport line. Vapor spreads equally into the vapor space of each con-

![Diagram of multi-condenser loop heat pipe]

Figure 1.2: The multi-condenser loop heat pipe incorporates a wick in the condenser to stabilized the liquid-vapor interface to prevent liquid from flooding lower condenser layers.
denser where it condenses onto the wicking structure and flows through the wicking structure toward the liquid return lines. The liquid subsequently passes through a subcooling channel where it is cooled below saturation temperature and finally flows as a subcooled liquid in the wick-free liquid lines towards the evaporator reservoir.

The vertical stacking of multiple condensers introduces new challenges to the loop heat pipe design. When multiple condensers are introduced, the gravitational pressure head resulting from a 10 cm column of water (1.0 kPa) is sufficient to flood the lower condenser layers inhibiting their capacity to condense vapor. Likewise if the heat pipe were inverted, there would be flooding of upper layers. A sintered wick was thus incorporated into the condenser to stabilize the liquid-vapor interface using capillary forces. The stability of the vapor-liquid interface at a wick surface depends on the pressure differences between the liquid and vapor. As long as the capillary forces of the interface can compensate for the pressure difference, the interface will be stable. If the pressure in the liquid is higher than in the vapor, then the liquid menisci between the sintered particles will be convex, referred to here as an advancing meniscus as shown in Figure 1.3. Similarly, if the pressure in the liquid is lower than that in the vapor, then the menisci will be concave, referred to here as a receding meniscus. Measurements performed on candidate sinters have shown that sinters which are able to sustain pressures across the interface without flow (i.e., break-through

![Diagram](image)

Figure 1.3: Maximum capillary pressure at a stable liquid-vapor interface is higher for the receding meniscus than the advancing meniscus.

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pressure or $\Delta P_{cap,max}$) are higher in the receding direction than in the advancing direction [8]. To take advantage of this effect and improve on the safety margin of the design against condenser flooding, the liquid side of the condenser needs to operate at a pressure that is more than 1.0 kPa lower than the vapor side. In the LHP, this requirement is achieved using a compensation chamber to set the liquid side pressure.

1.5 Literature Review

Heat pipes used for air cooling of electronics are typically built in a cylindrical form factor and coupled with solid, conducting fins to spread heat over a large surface area [2]. This traditional design is limited at high heat loads by the fin efficiency and heat flux at the heat pipe-to-fin interface. The current design improves upon fin efficiency by creating multiple pathways for condensation heat transfer. This arrangement, however, increases the system complexity especially when body forces act on the working fluid [9].

The integration of a porous structure in the condenser of a loop heat pipe has been studied to improve start-up stability, however the design did not allow for control of the liquid pressure causing a liquid layer to form over the porous element in the condenser in some cases and render it ineffective for separating vapor from liquid phases [10]. This study explored the stable operation of a loop heat pipe with multiple condensers in the presence of gravity-induced liquid pressures by including a porous wicking structure in the condenser and controlling the liquid side pressure.

1.6 Thesis Overview

The focus of this thesis was to fabricate and experimentally characterize the operation of condensers with sintered wicks and identify the impact of their operation on the overall multi-condenser loop heat pipe design. Chapter 1 introduces the motivation behind a multi-condenser loop heat pipe, the challenges for stable operation, and the proposed solution of including wicking structures in the condensers. Next, Chapter 2 explains the more detailed operation of this particular loop heat pipe, uncovers the detailed requirements for the con-
denser and provides the design for performance and fabrication of the condensers. Chapter 3 describes the experimental analysis carried out to emulate the condenser operation and discusses the implications for integration into the loop heat pipe. Finally, Chapter 4 predicts the improvements for future air-cooled heat exchangers and provides design guidelines for application of wicked condensers in more general LHPs.
Chapter 2

Design & Fabrication of Sintered Condensers

The detailed requirements necessary for stable operation of a multi-condenser LHP are described. A finite element model of the thermal-fluid transport in the condenser is used to validate the design. Fabrication techniques are described including brazing and welding approaches.

2.1 Requirements

The condenser needs to be designed to meet a specific set of thermo-fluidic requirements that allow it to function properly in the multi-condenser loop heat pipe. Furthermore, the design must be manufacturable such that every joint is hermetically sealed to a leak rate of no more than $10^{-9}$ sccm of helium.

2.1.1 Capillary Pressure

In general, a condenser wick with larger break-through or maximum capillary pressure is preferred to form a more stable interface between vapor and liquid. According to the Young-Laplace equation (2.1), the maximum capillary pressure of a wick is

$$\Delta P = \frac{2\sigma \cos \theta}{a}$$  \hspace{1cm} (2.1)
where \( a \) is the pore size, \( \sigma \) is the surface tension, and \( \theta \) is the contact angle between liquid and solid surface. The pressure drop \( \Delta P \) of a fluid flow with viscosity \( \mu \) and volumetric rate \( \dot{V} \) passing a distance \( L \) through a porous structure of cross section \( A \) is described by Darcy’s Law as

\[
\Delta P = \frac{\mu \dot{V} L}{\kappa A},
\]

where \( \kappa \) is the permeability of the wick. The permeability is inversely proportional to the pore size squared [11]. A tradeoff thus exists to govern the optimal wick within a condenser which is best illustrated by a pressure-flow network inside the condenser.

In the pressure-flow network of Figure 2.1, vapor enters the condenser at a pressure \( P_V \) and condenses onto the wick. Because the vapor space is a relatively large cavity (0.5 mm), the viscous pressure drop due to vapor flow is minimal. The condensate flowing through the wick pores with characteristic size 10 \( \mu m \), however, incurs a significant pressure drop of approximately 1 kPa. The pressure difference between the vapor and liquid-filled wick is stabilized by a capillary pressure which is larger near the outlet (\( \Delta P_{cap,2} \)) compared with the inlet (\( \Delta P_{cap,1} \)). If the maximum capillary pressure or break-through pressure of the wick is less than \( \Delta P_{cap,2} \), vapor will penetrate into the LHP liquid channel, blocking liquid flow in the liquid return lines and evaporator reservoir. If the pressure drop due to viscous liquid flow from 1 to 2 is greater than or equal to \( \Delta P_{cap,1} \), an advancing meniscus will form at 1 causing liquid to collect in the condensation space thereby inhibiting condensation.

To avoid an advancing meniscus near the inlet to the condenser, the viscous pressure drop from 1 to 2 must be less than the \( \Delta P_{cap,2} \) established during operation. The total pressure drop between the vapor and the liquid, \( \Delta P \), must carefully avoid the failure conditions of vapor penetration and flooding. The difference in vapor and liquid pressures is set by the evaporator reservoir and depends on the heat load and operating temperatures [12]. At the most strenuous operating point of \( T_V = 100 \, ^\circ \text{C} \) and a total system heat load of 1000 W, the evaporator is designed to produce a pressure drop is \( \Delta P = 6 \) kPa. This requirement sets the minimum break-through pressure for each condenser.
2.1.2 Subcooling

Because liquid flow passages in a LHP are smooth and contain no wick, condensate must be cooled below the liquid-side saturation temperature to prevent it from spontaneously vaporizing [13]. Particular attention must be given to the liquid temperatures of this loop heat pipe because the liquid pressure is reduced and thus the liquid-side saturation temperature is also reduced. To prevent vaporization in the liquid return line, a liquid subcooler is incorporated into each condenser layer as depicted in Fig. 2.2(a). The red lines in Fig. 2.2(a) represent vapor, the dashed blue lines represent liquid being actively subcooled and the blue lines represent subcooled liquid. Figure 2.2(b) is a cross section of two condensers from the dashed line marked AA’ in Figure 2.2(a). Condensate is able to be subcooled because vapor is blocked from contacting the subcooling channels by a solid Monel subcooling plate. A temperature gradient is maintained across the subcooling region because the air-side convection causes the subcooling region to act as a fin.
Figure 2.2: Schematic of vapor and liquid pathways inside condenser: (a) planar cross-section of one condenser and (b) cross section AA' of two condensers (not to scale).
The target liquid temperatures exiting each condenser is 3°C lower than the outlet saturation temperature. If the difference in vapor and liquid pressures is 4 kPa with a vapor temperature of 78°C, the temperature difference between vapor and liquid must be at least 5.5°C.

2.1.3 Material Selection

Several studies have demonstrated that common anti-corrosion materials including stainless steel are chemically disagreeable with water as a working fluid in heat pipes due to hydrogen generation [14, 15]. Hydrogen, a noncondensable gas, can collect in the condenser and block condensation sites, limiting heat transfer because vapor must diffuse through the noncondensable gas to reach the condensation surface. Gold, silver, copper and Monel 400 (an alloy of 66.5% nickel and 31.5% copper) have been proven to generate no hydrogen in heat pipes [11]. Copper and Monel 400 were therefore selected as manufacturing materials for the LHP components of this system.

2.1.4 Wick Selection

The condenser and evaporator wicks in the loop heat pipe of this study were made from sintered metal particles which are commonly used in heat pipes due their high conductivity, wicking ability, and flexibility in fabrication. The sinter particle size, material and temperature for sintering were chosen based on an experimental study of different types of sinter [11]. The 44 μm non-spherical Monel powder yielded the most favorable properties including a bulk permeability of $\kappa = 1.5\pm0.2 \times 10^{-12} \text{m}^2$, a maximum break-through pressure of $\Delta P_{max} = 18.6\pm3.0 \text{kPa}$, and a bulk thermal conductivity of $k = 2.3\pm0.2 \text{W/mK}$ [11].

2.2 Modeling

Given the basic condenser structure, wick properties, and liquid subcooling requirements, a finite element model was developed to optimized length ($l_{SC}$) and width ($w_{SC}$) of the subcooling channel to meet the subcooling requirements while providing maximum surface
area for condensation [16].

By symmetry, only one eighth of a condenser was analyzed in COMSOL Multiphysics (Burlington, MA) with a mirrored boundary condition applied to each sectional slice. The boundary condition on the vapor inlet was set with a reservoir of saturated vapor at 78 °C ($P_{sat}=43.7$ kPa), corresponding to the saturation temperature of the vapor, and the boundary condition at the liquid outlet was 39.7 kPa, corresponding to the design liquid pressure of the evaporator reservoir. A liquid mass flux proportional to the local conductive heat flux was applied across the condensation surface. The air-side boundary condition was based on a heat flux experimentally determined [6] with a convective heat transfer coefficient of $h=211$ W/m$^2$K. The external air temperature increases with radial position according to an energy balance based on experimental air flow rates with the air entering at 25°C. The wick properties, specifically the thermal conductivity and permeability, are based on experimentally measured bulk values [11]. The edges were conservatively assumed to be adiabatic. The COMSOL modeling was carried out for a range of sub-cooling geometries. The results are shown for the geometry that best meets the design goals in Table 2.1 and contour plots for pressure and temperature appear in Figures 2.3(a) and 2.3(b).

<table>
<thead>
<tr>
<th>Boundary Conditions</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_v$ $78^\circ$C</td>
<td>$T_L$ 63.22 $^\circ$C</td>
</tr>
<tr>
<td>$T_{air}$ 25$^\circ$C</td>
<td>$\dot{Q}$ 116 W</td>
</tr>
<tr>
<td>$\omega$ 5000 RPM</td>
<td>$l_{sc}$ 9 mm</td>
</tr>
<tr>
<td>$h$ 211 W/m$^2$K</td>
<td>$w_{sc}$ 6 cm</td>
</tr>
<tr>
<td>$\Delta P$ 4.0 kPa</td>
<td></td>
</tr>
</tbody>
</table>

Table 2.1: COMSOL modeling summary

Several factors which are difficult to incorporate into COMSOL modeling are important to identify for comparison to experimental results. First, the impellers create high heat transfer coefficients by shearing and mixing the flow in a circular footprint with diameter 10 cm. Since the condenser footprint is a square of 10 cm × 10 cm, some of the heat transfer surface area does not benefit from the impeller shearing motion. Secondly, the presence of liquid and vapor tubes cause a small area to be "shadowed" by circulating air, leading to an adiabatic region behind the tubes. Next, although most of the air flow enters the channel
from the central inlet, some ambient air is ingested into the low-pressure wake of each impeller blade, improving heat transfer at the condenser edges. Lastly, the wick properties measured from 44 μm samples represent idealized bulk measurements, whereas the permeability of the actual condenser wick is much larger due to weak bonding of the sintered wick with solid Monel walls as explained in the next section.
Figure 2.3: COMSOL modeling results: (a) Temperature profile for brazed condenser. The subcooling section is sized just large enough to sustain a temperature gradient between the vapor and liquid regions. (b) Pressure contour plot for a brazed condenser taken from a slice through the sintered wick. The white regions represent rigid boundaries including the white quarter circle indicating the slotted liquid transport tube.
2.3 Fabrication

The fabrication of condensers is subdivided into the assembly of the condenser package, the sintering process for making the wicking structure, and the tube joint connections to the remainder of the LHP.

2.3.1 Assembly

A metallic, hermetic seal of all joints in the heat pipe is necessary to guarantee a robust product that can withstand a lifetime of thermal and mechanical stresses. The most effective method of testing for hermetic seals is with a helium leak detector (Adixen ASM 142) for its sensitivity and high spatial resolution for locating microscopic leaks. The sinter and solid components of each condenser are fabricated from Monel 400 alloy which has high resistance to corrosion. To manufacture thin (0.5-1.25 mm) Monel parts with high precision, such as the part in Figure 2.4, sheets of Monel were photo-chemically etched (Photofabrication Engineering Inc., Milford, MA) to varying depths.

Brazing Approach

The first approach to condenser assembly was by a series of furnace brazing and sintering steps at successively lower temperatures. Each condenser was made from four photo-chemically etched plates of two types: a base plate (0.5 mm thick) and frame plate (0.75 mm thick) shown in Figure 2.4. Note that the subcooling plate is a sub-component of the frame plate. The subcooling plate is elevated by 0.5 mm to create the subcooling channel as shown in Figure 2.2(b). First two half-condensers were assembled. The base plate was bonded to the frame plate with silver at 970 °C. The base plate and frame plate provide structure for the condenser and a cavity to contain the sintered wick. Next 44 μm non-spherical Monel powder was poured into the frame and centrifugally packed under the subcooling plate by spinning the structure to 200 RPM. To allow for a vapor space, 0.25 mm of powder was scraped off the surface. A 1 mm wide "fillet" was left surrounding the perimeter of the subcooling plate as visible in Figure 2.5(b). The powder was sintered at 820 °C for 12 minutes. To compensate for linear shrinkage of the sinter (10 ~ 15%) which degrades capillary
pressure by opening relatively large gaps, an extra layer of 44 μm powder was layered over the "fillet" and sintered again at 820 ºC. The two halves were joined with a eutectic AgCu (72% Ag, 28% Cu) braze alloy at 810 ºC. A cross section of a condenser appears in Figure 2.6. Next, the condenser was brazed onto 9.5 mm (0.375") diameter slotted Monel tubes for transporting liquid and vapor with 60% Ag, 30% Cu, 10% Sn braze alloy at 750 ºC. To complete the heat pipe assembly, the tubes leading from the stack of condensers were attached to the evaporator with 80% Au, 20% Sn braze alloy at 330 ºC. Silver brazing, AgCu brazing, and all sintering processes were done in a tube furnace with constant flow of 5% H₂ + 95% N₂ at 3 liters/min while AgCuSn and AuSn brazing were done in a vacuum furnace of 1 × 10⁻⁵ Torr.

Many difficulties were encountered with the Brazing Approach, particularly with AgCu, AgCuSn, and AuSn braze alloys. Critical factors for successful braze joints include avoiding...
Figure 2.5: (a) The Monel base plate and frame plate were silver brazed together at 970 °C. (b) Monel powder was packed under the subcooling plates and sintered at 800 °C. Extra powder sintered along the perimeter of the subcooling plate increases maximum capillary pressure.
proximity to sinter, an appropriate temperature profile, cleanliness, and joint contact. The use of braze alloys as a metallic bond in the presence of sinter introduces critical problems for forming hermetically sealed joints. Ironically, the water-wicking sinter also induces liquid braze to flow out of the joint and into the sinter yielding an unsealed joint and clogged wick. If the parts are held too long at the brazing temperature, the braze will flow out of the joint. If the parts do not reach the brazing temperature, the braze may not melt completely. The braze and bonding surfaces are sensitive to contamination especially from organic compounds. All parts and brazes were cleaned in acetone, trichloroethylene, and ethanol. Lastly, the brazing process is driven by capillary action which requires that the parts be in close contact (25 μm) to ensure a hermetic seal of the entire joint.

For purposes of testing the internal fluidic structure of LHP components, a vacuum grade epoxy (Torr Seal, Varian Vacuum Technologies) was used as a temporary solution to seal leaking braze joints. Although the vacuum epoxy is strong and has low gas permeability, it is also brittle and cannot reliably withstand long-term thermal stresses.
Welding Approach

An alternative to brazing condensers is to weld the joints together. The weldable condenser approach necessitates a new fabrication process free of braze alloys to avoid welding dissimilar materials.

The weldable condenser approach eliminates the subcooling plate from Figure 2.2(b). Instead, the entire subcooling region is formed by sinter as shown in the schematic of Figure 2.7. First two Monel half condensers were photo-chemically etched from 1.25 mm sheets to form two trays each with a 0.75 mm deep cavity and 0.5 mm thick welding flanges. Next, a layer of 44 μm Monel powder was sintered into the Monel trays. The sinter is uniformly 0.5 mm thick in the condensation area and an "L" shaped section surrounding the liquid holes rises an additional 0.25 mm, flush with the top surface, to form the subcooling channels as shown in Figure 2.8. This first sintering process was done at 800 °C. A liquid channel 2 mm wide was carved out of the subcooling "L" and then a 0.35 mm layer of 22 μm Monel powder was added to the top surface of the subcooling channel of the first half while the second half was pressed face down on the first. The second sintering process, also done at 800 °C, resulted in two halves bonded by a 22 μm monel sinter "bridge" which has shrunk

![Figure 2.7: Schematic showing cross section AA' of two welded condensers (not to scale)](image-url)
Figure 2.8: Weldable half-condenser after the first two sinter steps (not to scale). The dark region represents the bridge for bonding the two halves and is made from 22μm Monel particles.

to 0.2 mm in thickness. The gap between the flanges, also 0.2 mm thick, was welded to hermetically seal the condenser and provide structure. The weld flange appearing in Figure 2.9 is included to supply material to make the joint and is made thin to prevent too much heat from spreading into the condenser during welding which could cause warping and destroy the subcooling structure. The completed condenser was subjected to a capillary pressure test to ensure that the capillary pressure across the sintered "bridge" was acceptable for operation in the LHP.

A COMSOL model for the welded condenser pressure and temperature profiles is shown in Figures 2.11(a) and 2.11(b). The presence of sinter in the weld joint prevented successful hermetic welds due to contamination of the weld puddle, therefore all sinter is spaced at least 2 mm apart from all weld joints in the design. Because of this spacing, the welded condenser can have vapor in the weld joint surrounding the liquid subcooling region.

The welding approach was adopted late in the development process because brazing processes are typically better suited for mass production compared to welding. The ability
Figure 2.9: Sectional view of weldable condenser (not to scale)

to achieve consistent, hermetic seals of condensers with sufficiently high capillary pressure and subcooling compelled the adoption of the welding approach for this LHP.
Figure 2.10: (a) Photo-chemically etched condenser half before sintering and (b) after sintering showing raised subcooling sections and carved-out liquid channels.
Figure 2.11: COMSOL modeling results for a welded condenser: (a) Pressure contour plot for a welded condenser taken from a slice through the sintered wick. The white regions represent rigid boundaries including the white quarter circle indicating the slotted liquid transport tube. (b) Temperature profile for a welded condenser. In contrast to the brazed subcooling geometry, a vapor temperature boundary condition was applied to the outer edge. This causes the edge of the subcooling region to be much warmer.
2.3.2 Sintering

The most significant challenge in achieving high capillary pressure during the fabrication was in forming a reliable bond between the sintered wick and solid Monel. During the heating process, sintered wicks tend to shrink and even detach from adjacent walls. When the sintered wick detaches from the solid Monel around the subcooling channel, the resulting gaps can lead to a reduced break-through pressure. A capillary pressure test was performed on each condenser by flooding it with water and pressurizing air on one side of the subcooling channel with a syringe until bubbles burst through and into the liquid channel as shown in Figure 2.12 in more detail. Although the measurements from sintered tube samples yielded $\Delta P_{\text{max}} = 18.6 \pm 3.0$ kPa, the test on an actual condenser demonstrated that such defects were indeed present due to the wick detaching from the subcooling plate and limited the break-through pressure to 3 to 4 kPa for the brazing approach. To compensate for the detachment, more sinter was added around the perimeter of the subcooling plate in a second sintering step which increased the break-through pressure to 7.1 - 8.4 kPa. The presence of defects also led to a slight increase in permeability of the porous wick.

![Figure 2.12: Air is pressurized into the vapor space of the condenser. The maximum pressure sustained before bubbles emerge from the liquid channel is the break-through pressure.](image)

Figure 2.12: Air is pressurized into the vapor space of the condenser. The maximum pressure sustained before bubbles emerge from the liquid channel is the break-through pressure.
2.3.3 Tube Joints

In order to seal the condenser to the monel vapor and liquid supply lines, a AgCuSn braze alloy was initially proposed which melts at 760 °C, however the same brazing problems in the vicinity of sinter were encountered as described in Section 2.3.1. A temporary solution to sealing tube joints for use of experimental testing was developed using soft Buna-N A50 AS568A size 012 o-rings. Figure 2.13 shows a cross-sectional schematic of the adopted o-ring assembly. By compressing vertically as shown, the o-rings deform and expand horizontally into the tube and retainer sleeve. The critical factors for achieving successful o-ring seals include having a linear compression of 25%, polishing the mating surface, and applying vacuum grease to the mating surface. The o-ring seals are simple and fast to set up, allow for removable connections, and are effective for common high vacuum applications [17]. However, the non-negligible permeability of o-rings to gases limits their use to a few days for heat pipes.

![Figure 2.13: Custom o-ring joint. The retainer sleeve is sized small enough to provide enough compression to the mating surface but large enough for the o-ring to fit in the gap without crushing the condenser. Two small o-rings are used instead of one large one to prevent interference with the impeller. (Not to scale).](image)
## Fabrication Summary

### Brazing Approach

<table>
<thead>
<tr>
<th>Step</th>
<th>Process</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Photo-chemical etch</td>
<td>plates are etched from 0.5 mm and 0.75 mm Monel 400 to make subcooling channel</td>
</tr>
<tr>
<td>2</td>
<td>Silver brazing</td>
<td>frame plate is bonded to base plate at 970 °C for 12 minutes in jig for compression and alignment</td>
</tr>
<tr>
<td>3</td>
<td>Sintering</td>
<td>half-condenser is filled with 44 μm Monel 400 particles (centrifugally packed), scraped to 0.25 mm thin to form vapor space, and sintered at 830 °C</td>
</tr>
<tr>
<td>4</td>
<td>Sintering</td>
<td>44 μm Monel 400 particles are added to the perimeter of the subcooling section and sintered at 830 °C</td>
</tr>
<tr>
<td>5</td>
<td>Quality check</td>
<td>test capillary pressure for each half</td>
</tr>
<tr>
<td>6</td>
<td>AgCu Brazing</td>
<td>two condenser halves are bonded together with eutectic AgCu braze at 810 °C</td>
</tr>
<tr>
<td>7</td>
<td>Quality check</td>
<td>test sinter and brazing joints for capillary pressure</td>
</tr>
</tbody>
</table>

### Welding Approach

<table>
<thead>
<tr>
<th>Step</th>
<th>Process</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Photo-chemical etch</td>
<td>plates are etched from 1.25 mm Monel 400 to form monel tray and welding flanges</td>
</tr>
<tr>
<td>2</td>
<td>Sintering</td>
<td>a 0.25 mm layer of 44 μm Monel 400 particles and raised subcooling is sintered at 800 °C</td>
</tr>
<tr>
<td>3</td>
<td>Sintering</td>
<td>a thin layer is sintered to compensate for shrinkage of sinter in step 2</td>
</tr>
<tr>
<td>4</td>
<td>Sintering</td>
<td>two halves are sintered together at 800 °C with a 0.35 mm layer of 22 μm Monel 400 particles added between them</td>
</tr>
<tr>
<td>5</td>
<td>Quality check</td>
<td>test capillary pressure</td>
</tr>
<tr>
<td>6</td>
<td>Weld</td>
<td>seal outer edges and central air inlet with tungsten inert gas (TIG) welder</td>
</tr>
</tbody>
</table>

*Table 2.2: Step-by-step fabrication procedure for condensers*
2.4 Improvements for Future Work

As discussed in section 2.1.2, there exists a compromise in the selection of sinter forming the condenser wick. If the sinter particles are too small, the viscous pressure loss due to liquid flow in the wick leads to flooding in the condenser whereas if the sinter particles are too large, the wick will not enable a large enough capillary pressure to separate the phases. This dilemma will always exist for a wick with bulk properties, however, a multi-layered wick could exploit the advantages of both wick properties. The first wick, which lines the condenser walls and is responsible for carrying liquid flow, could be made from larger sintered particles. The second can be sintered on top of the first to form a "mono-layer" of smaller sintered particles, yielding a high capillary pressure. This two step process is feasible because larger particles sinter at higher temperature than smaller ones because the surface energy driving sintering is lower with lower curvature particles. If the wick permeability is high enough, the size and form of the "L" shaped subcooling section can be reduced, allowing more surface area for condensation heat transfer.

A significant limitation to the heat transfer performance of the condenser is the insulating layer of sinter \( (k = 2.3\pm0.2 \text{ W/m-K}) \) which lines the condenser walls. With a heat output of 100 W per condenser and a total temperature drop of 50 °C between the base and ambient air, the temperature drop across the condenser wick is approximately 2.5 °C. If the condenser wick were fabricated from a higher conductivity wick (e.g., copper) or the condenser were lined with a thinner wick, a 4% reduction in the overall thermal resistance of the air-cooled heat sink could be immediately realized.

The thickness of the condenser was set at 2.5 mm for feasibility of manufacturing. Reducing the thickness of each condenser is favorable because more condensers can be packed into the integrated device and the thermal resistance of the condenser walls will also be reduced. The estimated pressure drop in the vapor space is 54 Pa compared with 1000 Pa in the wick [18], therefore the vapor space thickness can be reduced without deleterious effects to the thermal performance. If manufacturing methods were improved, a reduction in condenser thickness could lead to significantly increased fin packing density.
2.5 Chapter Summary

For the multi-condenser LHP to operate stably, the condenser must meet a stringent set of operational requirements. A flow model for pressure and critical maximum for liquid outlet temperatures provided design constraints for the evaluation of a COMSOL model of the temperature and pressure distribution in the condenser. Two fabrication processes were described in detail and the use of welding for making hermetic seals was identified as the better approach.
Chapter 3

Experimental Characterization & Analysis

An experimental setup to characterize the performance of a single condenser and a pair of condensers is described in Chapter 3. The results are compared to the modeling presented in Section 2.2 and are used to determine the range of pressure differences between the vapor and liquid side within which condensers in the loop heat pipe can operate stably.

3.1 Objective

The emphasis of this study is to characterize the operation of condensers with sintered wicks and define the stable operating range of the LHP. To characterize the condenser behavior, a custom flow rig was designed to regulate the condenser operation independent of the evaporator. The key operating conditions imposed by the integrated LHP and fan are the impeller rotational speed and the difference between liquid and vapor pressure. The condenser was thus characterized at a fixed vapor temperature over a range of liquid-side pressures and impeller speeds to identify the best operating point for the LHP.
3.2 Single Condenser Experiment

The condenser for this study contains a sintered wick to stabilize the liquid-vapor interface and thus permit multiple condensers to operate in parallel under a gravitational liquid pressure head. To simplify the experiment and provide a control, a single brazed condenser was first characterized followed by a comparison to a multi-condenser experiment.

3.2.1 Setup

The setup emulated operating conditions of the complete LHP. As shown in Figure 3.1, a vapor supply tank with an attached band resistive heater (not shown) was used to generate vapor at 80 °C which corresponds to a saturation pressure of 47.4 kPa. The temperature was chosen to match the design vapor temperature of the LHP. The vapor flows into the condenser via two vapor inlets on the bottom side. Condensate flows out of the condenser via two liquid outlets and returns to a condensate tank with an attached resistive heater used to adjust the condenser outlet pressure by varying the temperature in the tank. The difference in pressure between the vapor entering the condenser and the liquid flowing out of the condenser, \( \Delta P \), was measured using a differential pressure gauge (FDW 060-M838-02, Honeywell) with an uncertainty of \( \pm 0.1 \) kPa. The vapor and liquid temperatures at all four tube connections were measured by thermocouples (TMQSS-062G-3, Omega) positioned in the fluid flow channel, each with an uncertainty of \( \pm 0.5 \) °C and are indicated by \( T \) in Figure 3.1. To emulate the cooling mechanism and air temperature profile of the system, an impeller was setup to rotate at 5000 rpm on each side of the condenser. Adjacent to each impeller was a heated aluminum square plate, which is cut away in Figure 3.1 to reveal the underlying impeller. The heated aluminum plates emulated the air flow constriction and heat input of adjacent active condensers on each side of the experimental one. Each plate was heated with 50 W by kapton film heaters and insulated on the side opposite to that of the spinning impeller.

Non-condensable gases impede heat transfer in a condenser because the transport of vapor to the condensation surface is limited by diffusion [3]. Therefore, the use of de-gassed water as the working fluid and vacuum tight seals are essential. The most effective method
for removal of non-condensable gases is the freeze-pump-thaw cycle [18]. Because gases have low solubility in ice, freezing the water causes the release of non-condensable gases. The vapor supply tank was filled with water, frozen, and the tank was then pumped to vacuum to evacuate released non-condensable gases. Since some non-condensables can be trapped in the crystalline structure of ice (though not actually dissolved), the ice was thawed and the cycle was performed two more times or until the tank pressure and temperature matched by saturation conditions.

Figure 3.1: Top view of experimental setup. Vapor flows into the condenser, which is cooled by an impeller-driven air flow. Liquid exits the condenser to a condensate tank which is maintained at a pressure lower than the pressure in the vapor supply.
3.2.2 Procedure

After all freeze-pump-thaw cycles were completed, the condenser and all connected pipes were pumped down to vacuum with the vapor supply valve closed. The de-gassed water in the vapor supply tank was heated to 80 °C and the empty condensate tank was also heated to 80 °C. The vapor supply valve was opened and steady state was reached after water filled the liquid pipes and enough water entered the evacuated condensate tank to satisfy saturation conditions. While the vapor supply was maintained at 80 °C and 47.4 kPa, the temperature and thus the pressure in the condensate tank was gradually lowered to impose a pressure drop, $\Delta P$, across the condenser. The pressure drop was increased until vapor penetrated through the sintered subcooling channel and into the liquid pipe, destabilizing the liquid-vapor interface and indicating a failure of the condenser. Measurements from each sensor were recorded in LabView by a DAQ card (PCI-6289, National Instruments).
3.2.3 Results

Figure 3.3 shows characteristic experimental results of temperature and pressure data as a function of time. Figure 3.3 (a) shows the measured inlet vapor temperature ($T_{vap}$), each of the measured outlet temperatures ($T_{iq}$), and the calculated saturation temperature at the liquid outlet ($T_{sat,out}$) as functions of time. The liquid outlet saturation temperature was found by converting the saturated vapor inlet temperature to its corresponding saturation pressure, subtracting the measured $\Delta P$ from this value, and calculating the saturation temperature corresponding to that pressure, or

$$T_{sat,out} = T_{sat} \left( P_{sat}(T_{vap}) - \Delta P \right)$$  \hspace{1cm} (3.1)

Figure 3.3 shows that the temperature of vapor entering each inlet of the condenser has a constant value of $T_{vap} = T_{vap,1} = T_{vap,2} = 79.1 \pm 0.3 ^\circ C$. Figure 3.3 (b) shows the imposed pressure drop, $\Delta P$, between the vapor inlet and liquid outlet as measured by the differential pressure gauge as a function of time. The periodic oscillations in $T_{vap}$ and $\Delta P$ are due to the band heater on the vapor supply tank turning on and off. From Equation 3.1, since the average vapor temperature and pressure remain constant, the change in $\Delta P$ varies inversely to the change in $T_{sat,out}$, the local liquid outlet saturation temperature. The differences in the two liquid outlet temperatures is likely due to variability in the two porous subcooling channels caused by defects in sinter bonding to solid Monel walls.

The trends in temperature and pressure over time clearly demonstrate the functionality of the condenser. Vapor always enters the condenser in each vapor line while only liquid exits the liquid lines for the first 20 minutes of the experiment as the pressure drop $\Delta P$ increased.

An important indicator for stable operation of the condenser is the liquid subcooling, $\Delta T_{sc}$, which is the difference between the liquid outlet saturation temperature and actual liquid outlet temperature:

$$\Delta T_{sc} = T_{sat,out} - T_{liq}$$  \hspace{1cm} (3.2)

The average of the two liquid outlet temperatures was used for $T_{liq}$ in calculating $\Delta T_{sc}$. Although the liquid subcooling may be expected to reduce to zero with vapor appearing in the liquid line after the total pressure drop surpasses the break-through pressure of 6.2 kPa, the
Figure 3.3: (a) The temperature of the liquid exiting the condenser (blue lines) rises to meet the saturation temperature (green line). The vapor temperature (red) is nearly constant. (b) The corresponding differential pressure also rises.
Figure 3.4: The change in liquid subcooling ($\Delta T_{SC}$) as a function of the differential pressure ($\Delta P$). After the breakthrough pressure (6.2 kPa) is exceeded, the liquid subcooling decreases.

liquid remained subcooled. The level of subcooling in Figure 3.4 was observed to gradually decrease with increasing pressure drop until $\Delta P = 13$ kPa. This gradual failure can be explained by two phase flow through the sintered wick as shown in Figure 3.5. At pressures below 6.2 kPa, the liquid-vapor interface is stable at the designed location, with liquid filling the entirety of the sintered subcooling channel. When the pressure drop rises above the break-through pressure of 6.2 kPa, some vapor is able to penetrate into the condenser wick and low density vapor begins to flow through the low permeability sinter yielding a large viscous pressure drop. Due to non-uniformities in the sintered wick, vapor is able to penetrate the larger gaps, leading to vapor cavities and vapor tendrils. The largest defects are exploited at 6.2 kPa and as the differential pressure increases, smaller gaps with higher capillary pressure are exploited leading to a gradual decay in liquid subcooling. The vapor is prevented from reaching the liquid outlet as it is cooled by the sinter and mixes with subcooled liquid but the temperature of the exiting liquid increases, i.e., $\Delta T_{SC}$ decreases. Eventually $\Delta T_{SC}$ is reduced to zero as vapor and liquid coexist in the liquid return line which
would cause the LHP to fail.

![Schematic](image)

**Figure 3.5:** Schematic showing vapor penetration through sinter. As the pressure drop increases, the interface recedes non-uniformly into the sintered subcooling channels. As a result, the amount of subcooling decreases.

The liquid mass flow rate was measured using a calorimetric flow meter (TURCK FCI-TCD04A4P-LIX-H1141) which is limited by its sampling rate of approximately 3 s. Pressure oscillations of the same time scale, which were likely caused by capillary effects from liquid slugs in the vapor tubes, induced oscillations in the liquid flow which the flow sensor could not accurately measure, however, the mass flow meter provided useful insights to the condenser operating conditions.

By conservation of energy, the heat dissipated by the condenser can be calculated using the measured mass flow of condensate. Vapor entering the condenser at saturation transfers its latent heat of vaporization to the condenser as it changes to the liquid phase. The liquid transfers more heat to the condenser as it is subcooled. Equation 3.3 captures the transfer of heat from the working fluid to the condenser:

\[
\dot{Q} = \dot{m}(h_{fg} + c_p(T_{vap} - T_{iq}))
\]  

(3.3)

where \( \dot{Q} \) is the rate of heat transfer, \( \dot{m} \) is the mass flow rate, \( h_{fg} \) is the latent heat of vaporization, and \( c_p \) is the specific heat of the liquid at constant pressure. The heat loss associated with subcooling is small compared to the latent heat loss. The heat rejection
calculated by this method includes heat loss through the vapor lines connecting the vapor tank to the condenser, however all pipes were well-insulated during the experiment so the contribution from the insulated lines should be small relative to the heat transferred to the air at the condenser. Because heat rejection for each condenser is governed by air convection factors such as geometry and impeller speed, the flow of condensate should be constant during the entire experiment. The data for condensate mass flow was consistent with this assumption during the experiment and averaged 0.043±0.015 g/s which corresponds to a heat loss of 101±36 W for the condenser. A more accurate method of measuring heat transfer was developed for the dual condenser experiment.
3.3 Dual Condenser Experiment

In addition to sustaining a positive $\Delta P$, the dual layer experiment was designed to test the heat dissipation of each of two condensers arranged in parallel, specifically whether the lower condenser will flood due to hydrostatic liquid pressure. In addition to manipulating the liquid side pressure, the influence of impeller speed on condenser performance was also characterized.

3.3.1 Setup

The vapor tank, liquid tank, fluid connections, and instrumentation for the dual condenser experiment were identical to the single condenser setup as shown in Figure 3.1 except that two brazed condensers were attached to common vapor and liquid tubes and separated by 6.5 cm as shown in Figure 3.6. Four slotted monel tubes were soldered to a brass base plate which houses the lower bearing of the shaft. Each condenser was slid onto the four tubes from above with o-rings used to seal the tube connections as shown in Figure 2.13. The plates adjacent to each condenser were not electrically heated because the soldered connections caused heat to be conducted to the liquid tube perturbing the liquid temperature measurement.

3.3.2 Procedure

The procedure for characterizing the two condensers for given a range of $\Delta P$ was identical to the procedure for the single layer experiment. The effect of impeller speed on heat transfer performance was also investigated. An improved method to measure heat dissipated by the condenser was developed based on the electrical power applied to the vapor tank. Whereas the heater control system used for single layer experiments was a simple on/off feedback loop based on the vapor tank temperature, the improved control system consists of a proportional feedback with constant offset. The constant offset was tuned to supply the correct amount of heat to hold the temperature at its set value while the proportional feedback accounted for error in the offset. The power supplied to the heater, which was in excess of 500 W, was modulated by the duty cycle of a square wave signal sent to a solid state relay which connects the band heater to a VariAC voltage supply. The instantaneous power delivered
to the vapor tank was recorded based on the voltage of the VariAC, resistance of the band heater, and duty cycle of the signal. At steady state, the heat dissipated by the condensers is equal to the electrical power provided to the vapor tank less the heat lost by the vapor tank and vapor supply lines. Since the vapor set point temperature was constant even as the heat flux to the tank increased, the heat loss was also constant with total heat input. Thus as the impellers rotated faster and increased the convective cooling, vapor was condensed at a higher rate and more power was required to maintain the vapor tank at its set temperature.
3.3.3 Results

The dependence of liquid subcooling, $\Delta T_{SC}$, on differential pressure, $\Delta P$, at 8000 RPM and $h_{ITD} = 198.9 \text{ W/m}^2\text{K}$ [6] is plotted in Figure 3.7 for the liquid temperature at the top and bottom condenser. The convective heat transfer coefficient $h_{ITD}$ is defined in Equation 3.5. The liquid subcooling is an important indicator to condenser performance. When the differential pressure (measured at the bottom condenser) is too low, the subcooling rises suddenly because the condenser begins to flood. The top condenser flooded at a lower indicated $\Delta P$ because the local differential pressure was 0.64 kPa lower due to hydrostatic forces. Likewise, vapor burst through the top condenser at a lower indicated $\Delta P$ because the local differential pressure was 0.64 kPa higher. Between 2 kPa and 8 kPa, both condensers were operating as designed with the liquid-vapor interface stabilized by the condenser wick.

![Figure 3.7: The plot shows changes in liquid subcooling due to the change in differential pressure with impellers spinning at 8000 RPM. The subcooling trend in the top condenser is shifted to the left because the hydrostatic liquid pressure is 0.64 kPa lower.](image)
Figure 3.8: The change in liquid subcooling ($\Delta T_{SC}$) as a function of the differential pressure ($\Delta P$) at the lower condenser. After the breakthrough pressure (6.2 kPa) is exceeded, the liquid subcooling decreases dramatically.

The data for subcooling compared to differential pressure are plotted for three different impeller speeds in Figure 3.8. The curves represent time-averaged data points. The data shown is from the thermocouple measuring liquid temperatures exiting the bottom condenser. The subcooling measured at 8000 RPM is shifted up and left from the data at 2000 RPM. The subcooling trend is an effective indicator of the state of condenser operation and each curve has three characteristic slopes indicated by Zone 1-3. The subcooling behavior of Zone 1 at 8000 RPM is large because the condenser was in the flooded state at these low differential pressures. The slope corresponding to Zone 2 is related to the slope of the saturation curve. In this range, the liquid temperature does not change significantly, but the local saturation temperature decreases yielding an apparent decrease in liquid subcooling, $\Delta T_{SC} = T_{sat, out} - T_{liquid}$. At some value of differential pressure, the liquid subcooling transitions from
Zone 2 to Zone 3 with a steeper decreasing trend. At this transition, the capillary limit of the meniscus is exceeded and vapor penetrates into the liquid line forming vapor cavities and vapor tendrils. The differential pressure at the transition point occurs at higher $\Delta P$ for higher RPM because the measured differential pressure is

$$\Delta P_{\text{transition}} = \Delta P_{\text{cap, max}} + \Delta P_{\text{viscous}}$$

(3.4)

according to the pressure network in Figure 2.1. The break-through pressure, $\Delta P_{\text{cap, max}}$, is not a function of the impeller speed. Instead, the viscous pressure drop in the sinter is proportional to the mass flow through the wick which is proportional to the rate of heat transfer. The relationship between heat transfer and impeller speed determined experimentally [6] is summarized in Table 3.1 for a 2.5 mm thick channel with a 1.5 mm thick impeller. As a system designer, it is useful to note that the electrical pumping power $W_e$ of the impellers scale with the square of impeller speed $\omega$ while the convective heat transfer coefficient $h_{IDT}$ scales linearly with the impeller speed. The convective heat transfer coefficient based on the surface area and inlet temperature difference $h_{IDT}$ is defined as

$$h_{IDT} = \frac{\dot{Q}}{A \cdot (T_{\text{surface}} - T_{\text{inlet}})}.$$  

(3.5)
The data and linear fit in Figure 3.9 represent the electrical power applied to the vapor tank band heater to maintain it at a constant temperature and pressure. The heat loss to the surroundings by the insulated vapor tank and vapor tubes can be extracted by the y-intercept of the linear fit, which is 112 watts. The plot of impeller speed and heat transfer for a condenser, shown in Figure 3.9, matches the air flow experimental data in Table 3.1 within 4.2%. The data presented in Table 3.1 over-estimates the heat dissipation from an active condenser because the surface temperature above the subcooling sections of the condenser surface was not as high as the vapor temperature. The linear relationship between 2000 and 5000 RPM confirms that the heat dissipation by the condenser is air-convection-limited in this range. At larger heat loads, the data deviates from the linear trend because the thermal resistance of the sinter has a more significant impact on the heat transfer. Since vapor condenses on the surface of Monel sinter which has thickness $L = 0.5$ mm and bulk conductivity of $k = 2.3$ W/mK, the heat must be conducted through an additional thermal resistance:

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{sinter}}$$  \hspace{1cm} (3.6)

$$R_{\text{total}} = \frac{1}{h_{\text{IDT}}A} + \frac{L}{kA}$$  \hspace{1cm} (3.7)

Table 3.1: Air side heat transfer data for two heated plates spaced 2.5 mm apart with a 1.5 mm thick impeller rotating between them.

<table>
<thead>
<tr>
<th>$\omega$ [RPM]</th>
<th>$\dot{Q}$ [W]</th>
<th>$h_{\text{IDT}}$ [W/m²K]</th>
<th>$\dot{V}_{\text{air}}$ [liters/s]</th>
<th>$\dot{W}_{\text{e}}$ [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>84.4</td>
<td>49.7</td>
<td>1.06</td>
<td>0.39</td>
</tr>
<tr>
<td>3000</td>
<td>126.6</td>
<td>74.6</td>
<td>1.59</td>
<td>0.98</td>
</tr>
<tr>
<td>4000</td>
<td>168.8</td>
<td>99.5</td>
<td>2.12</td>
<td>2.01</td>
</tr>
<tr>
<td>5000</td>
<td>211.0</td>
<td>124.3</td>
<td>2.65</td>
<td>3.63</td>
</tr>
<tr>
<td>6000</td>
<td>253.2</td>
<td>149.2</td>
<td>3.18</td>
<td>5.96</td>
</tr>
<tr>
<td>7000</td>
<td>295.4</td>
<td>174.1</td>
<td>3.71</td>
<td>9.15</td>
</tr>
<tr>
<td>8000</td>
<td>377.6</td>
<td>198.9</td>
<td>4.24</td>
<td>13.33</td>
</tr>
</tbody>
</table>
Figure 3.9: Electrical band heater power supplied to vapor generation tank over a range of impeller speeds. The condensation rate can be extracted from the band heater power by subtracting the heat lost by the (insulated) tank and vapor supply lines due to natural convection.
A plot of heat dissipation as a function of differential pressure is another useful indicator of condenser performance and appears in Figure 3.10. When the differential pressure is too low, flooding occurs in the lower condenser which is indicated by the drop in heat dissipation. An accurate measurement in condenser heat dissipation requires steady state heating of the vapor tank and so a spike is sometimes evident below 2 kPa due to a sudden rise in heat supplied to the vapor tank. When the liquid-vapor interface is stabilized by the condenser wick, the heat dissipated is constant over a range of differential pressure as the condenser was designed for. When $\Delta P$ increases beyond the maximum capillary pressure or breakthrough pressure, the heat dissipation increases slightly as more sites become accessible for condensation. Although it seems favorable to performance, this regime approaches the limit where liquid subcooling rapidly decays leading to failure of the LHP.

![Figure 3.10: Heat dissipation as a function of the differential pressure for two condensers in parallel. Below 1.5 kPa, liquid floods the lower condenser yielding a drop in heat dissipation.](image)
3.4 Discussion

To ensure that the condensers operate stably between the limits of flooding and vapor penetration, the loop heat pipe must set an appropriate ΔP across the condenser. Figures 3.8 and 3.10 provide regimes for the optimal liquid pressure in which to operate a multi-condenser LHP over a range of impeller speeds. For example, the stable operating range of differential pressure at 8000 RPM is 2.6 kPa < ΔP < 9.0 kPa as shown in Figure 3.11.

The condenser was not designed to operate with a differential pressure exceeding its maximum capillary pressure. However, the gradual rather than abrupt decrease of subcooling due to vapor cavities and vapor tendrils in the sintered wick extend the allowable pressure drop from 9.0 kPa to 10.6 kPa for the case of 8000 RPM. Although this operating zone should be avoided, it serves as a buffer zone before complete failure of the loop heat pipe.

![Figure 3.11: The liquid subcooling data identifies the stable operating zones of the condenser as shown above for 8000 RPM.](image-url)
3.4.1 Comparison to Model

Table 3.2 compares the experimental data for a single brazed condenser to those obtained by the COMSOL modeling of Section 2.2. With the boundary conditions nearly identical, the model over predicts the heat dissipation $\dot{Q}$ by 16% and under predicts the liquid subcooling $\Delta T_{SC}$ by 42%. The reasons for the discrepancy may be accounted for by dissimilarities in boundary conditions and material properties. Defects in the sintered wick allow larger permeability than the model assumed based on bulk sinter properties. The convective heat transfer coefficient is very sensitive to the gap between impeller and condenser surface [6] which is 0.5±0.2 mm. The "shadowing" effect of the tubes blocking air flow is difficult to model. Furthermore, in the dual condenser experiment the vapor and liquid tubes were connected using o-ring assemblies similar to Figure 2.13 which cover an additional 7.2% of the available heat transfer area from airflow. Boundary conditions for the COMSOL model assume that all incoming air is at 25 °C and enters only from the central inlet. Small amounts of cool air flowing into the low pressure wake of the impeller blades near the edge of the condenser could lead to a dramatic increase in liquid subcooling.

<table>
<thead>
<tr>
<th></th>
<th>COMSOL</th>
<th>Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boundary Conditions</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T_v$</td>
<td>78 °C</td>
<td>77.2 °C</td>
</tr>
<tr>
<td>$T_{air}$</td>
<td>25 °C</td>
<td>25.2 °C</td>
</tr>
<tr>
<td>RPM</td>
<td>5000</td>
<td>5140</td>
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<tr>
<td>$\Delta P$</td>
<td>4.0 kPa</td>
<td>4.0 kPa</td>
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<tr>
<td>Results</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>113 W</td>
<td>97.5 W</td>
</tr>
<tr>
<td>$T_{liquid}$</td>
<td>63.2 °C</td>
<td>53.3 °C</td>
</tr>
<tr>
<td>$\Delta T_{SC}$</td>
<td>12.6 °C</td>
<td>21.7 °C</td>
</tr>
</tbody>
</table>

Table 3.2: Comparison of experimental results to COMSOL model
3.5 Chapter Summary

A single brazed condenser with a sintered wick was experimentally characterized in a custom flow apparatus and determined to successfully stabilize the liquid-vapor interface. Next, two brazed condensers arranged in parallel were tested to verify the equal operation in the presence of a gravitational pressure head. The trends in liquid subcooling and heat dissipated over a range of differential pressures served as indicators for the window of stable operation which is dependent on the impeller speed.
Chapter 4

Conclusion and Future Work

The loop heat pipe is a passive heat exchanger which exhibits low thermal resistance due to phase change heat transfer using carefully designed channels and wicks. This work applied the fundamentals of the LHP to a novel air-cooled heat sink with multiple condensers to address increasing difficulties in thermal management. A multi-condenser loop heat pipe would typically suffer from performance instabilities due to parallel condensation pathways, however to stabilize the liquid and vapor regions during operation, a porous wick was incorporated into each condenser. A pair of condensers with sintered wicks were experimentally characterized to determine the limits of stable operation for integration into a LHP. The design demonstrated the potential for producing versatile, robust, efficient heat exchangers for a variety of applications extending beyond the scope of air-cooled heat exchangers.

4.1 Conclusion

The stability of the LHP was achieved by modulating the difference in vapor and liquid side pressures to avoid failure states inside the condenser. A positive pressure drop $\Delta P = P_{vapor} - P_{liquid}$ is necessary to drive the flow of liquid condensate through the sintered wick without the condenser flooding, which would degrade heat transfer performance. If the pressure drop is too large, vapor will penetrate through the sintered wick and collect in the liquid return lines, depriving the evaporator of liquid supply. These two failure mechanisms were carefully characterized to define a map of stable LHP operation based on the cooling...
rate provided by the impellers. A finite element model was used to validate the design while a dedicated experimental rig was used to emulate the operation of condensers in the LHP and provided confirmation for the build and design of the condenser.

LHPs are capable of transferring heat over large distances with low thermal resistances. The LHP with a wick in the condenser improves this flexibility by allowing for multiple condensers even when attached to separate vapor and liquid transport lines. Because the interface is stabilized in all condensers independent of the flow, not all condensers must be cooled equally, thus the system has more flexibility depending on the cooling supply available. For example, for moderate thermal management needs, some condensers can be cooled by a high efficiency blower while other condensers are only activated when cooled by a less efficient, reserve refrigeration or liquid cooling system for higher heat loads. Additionally, the condenser of this study allows for a modular design of the LHP. A common evaporator can be coupled with one, two, or twenty condensers depending on the application cooling requirements.

4.2 Operational Limits

Although there is in theory no limit to how many condensers such a device can accommodate, there are limits to the scalability and total size of the LHP which are dependent on the wick properties. For example, the condenser of this study demonstrated a stable operating window of up to 6 kPa, depending on the heat flux and operating point on the saturation curve. This range will allow for an array of vertically stacked condensers spaced up to 63 cm apart to function while resisting gravitationally induced liquid pressure differences. The convective cooling surface area of a condenser similar to the one presented here is limited by the pressure drop due to viscous flow through the wick, which is summarized by Figure 2.1. If the width and length of the condenser were doubled while the thickness remained the same, the minimum pressure drop needed to avoid flooding would be eight times higher because the pressure drop due to viscous flow is proportional to the length and proportional to the mass flow rate. The mass flow rate is proportional to the heat dissipation which is proportional to the surface area, hence the cubic dependence of the viscous pressure drop on
the length of the condenser:

\[
\Delta P_{\text{viscous}} \propto l \cdot \dot{m} = l \cdot \frac{\dot{Q}}{h_{fg}} = l \cdot \frac{h_{ITD} A \Delta T}{h_{fg}} 
\]  

(4.1)

\[
\Delta P_{\text{viscous}} \propto l^2 \cdot \frac{h_{ITD} \Delta T}{h_{fg}} 
\]  

(4.2)

where \( l \) is the length scale of the condenser, \( \dot{m} \) is the mass flow rate, \( \dot{Q} \) is the heat dissipation, \( \Delta T \) is the inlet-to-surface temperature difference, \( h_{ITD} \) is the convective heat transfer coefficient, and \( h_{fg} \) is the latent heat of vaporization. The capillary pressure must be greater than the viscous pressure drop to avoid flooding in the condenser. Alternatively, the critical restriction to scaling the heat pipe smaller would be maintaining a temperature gradient between the vapor and liquid phases with minimal heat conduction through the subcooling channel.

### 4.3 Future Work

The most significant obstacle remaining for integrating condensers into a complete loop heat pipe is the development of hermetically sealed tube joints. Braze alloys for the joint such as the proposed 60% Ag, 30% Cu, 10% Sn were unsuccessful in consistently sealing tubes to the condensers partially due to the thin (0.5 mm) joint geometry. Welding the tubes in place is not possible from the outside because there is not enough space between the layers (2.5 mm) for a TIG welding torch to fit. Instead, a welding approach to tube attachment is in development in which the seal is made from inside the joint as shown in Figure 4.1.

The condenser is sintered and the external joint is welded before tubes are attached. First, the bottom surface of a sealed condenser is welded to four tube stubs which extend from the evaporator. Next, four tube stubs are welded to the top surface of the condenser which allows attachment of the next condenser. Welded joints can be done visually yielding a higher success rate compared to brazed joints and can achieve long-lasting, hermetic seals compared to o-ring joints.

Improvements in manufacturing techniques and further optimization of the condensers can lead to significant improvements in the overall effective thermal resistance. The single and dual layer experimental results indicate that the subcooling region was oversized leading
to more liquid subcooling than was needed at the expense of reduced condensation surface area. A reduction in the subcooling width and length would lead to more heat dissipation for each condenser. New shapes should also be explored such as a diagonally or radially shaped subcooling sections rather than the L-shaped geometry. Lastly, improvements in the condenser wick capillary pressure, permeability, and thermal conductivity should be explored as suggested in Section 2.4.

In addition to the continuing development of repeatable manufacturing techniques for durable, hermetically sealed components, the performance of multiple condensers will be characterized while coupled with an evaporator. The construction of a multi-condenser loop heat pipe will validate the design and functionality of the condenser. Particular attention will be addressed to passive control of liquid-side pressure, which may be achieved using a two phase reservoir built into the evaporator structure. The functionality of inverted startup and steady operation of a multi-condenser LHP will also be investigated.
This study demonstrated that the inclusion of a wicking element in the condenser of the loop heat pipe increases the flexibility in design by allowing a modular construction with multiple condensers which has the potential to produce high performance heat exchangers.
Appendix A

Experimental Results for a Welded Condenser

A welded condenser was experimentally characterized by the same procedure as described in Chapter 3 to validate its performance in comparison to a brazed condenser. Plots of subcooling and heat dissipation as functions of differential pressure are shown in Figures A.1 and A.2. The liquid subcooling for a welded condenser was better than for a brazed condenser because the bond formed by the sinter 'bridge' for a welded condenser is better than the bond of sinter to the solid, smooth subcooling plate. A plot indicating the heat dissipation as a function of RPM is shown in Figure A.3. The heat dissipation for a welded condenser is the same as for a brazed condenser.
Figure A.1: Liquid subcooling as a function of the differential pressure for a welded condenser

Figure A.2: Heat dissipation as a function of the differential pressure for a welded condenser
Figure A.3: Heat dissipation as a function of impeller speed for a welded condenser
Bibliography


